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Practical Approach for Solving Vibrations of Large Turbine and Generator Rotors - Reconciling the Discord between Theory and Practice

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The purpose of this paper is to illustrate a different perspective in viewing and solving vibration problems in large rotating machines regarding the commonly seen discord between theoretical predictions of dynamic behavior, especially the standard predicted and expected "fixes" to many vibration problems, versus observed operation in practice when unexpected vibration problems still remain or arise anew. The paper will also discuss the key root causes behind this discord with regard to large turbine and generators rotors, and behind unexpected or unexplainable vibration in operation, usually after a major outage. In short, the primary cause in a substantial portion of such cases is the presence of "significant", axially distributed mass eccentricities inherent to individual rotors, or compound eccentricities from misaligned rotors or bearings. These cases require a different approach versus the methodology traditionally utilized for diagnosing and resolving "unbalance responses" in general, on a variety of rotating machines of different sizes and operating speeds. The paper also presents and describes an improved rotor balancing approach when dealing with such cases. These problems should be ideally resolved in service shops, and when balancing significantly eccentric rotors in balancing facilities, it is necessary to apply a new balancing method using 2N+1 balancing planes, where "N" is the highest mode reached in operation.

1 Introduction

Over many years of consulting work, the authors have recognized that at any number of power plants, turbine-generator rotor vibration problems continue sometimes for years without an effective resolution, despite the best efforts of OEMs and plant engineers and other consultants to solve them. It is likely that most practicing engineers working in this area of rotordynamics and vibration have encountered special cases where the machine appears unaware of the theoretical behavior it is supposed to adhere to. Likewise, the suspected root causes and proposed solutions predicted by standard theory do not always reliably resolve the turbine-generator vibration problems at hand. In a lot of these cases, any excessive discord between predicted and actual measured behavior is written off as "nonlinearity" or as due to unknowable external variables. Subsequently, an educated series of what become essentially trial-and-error solutions (particularly field balancing) are usually attempted until finally a combination arises that creates a tolerably running machine.

However, through many years of focusing on troubleshooting these kinds of special cases, it is evident that a good majority of these situations actually do tend to follow a regular and predictable pattern of root cause, and can be treated with a reliable framework of solutions - if the observed symptoms are properly recognized and understood. The root cause of "unexplainable" vibrations is almost always the presence of distributed mass eccentricity in the rotor train. This includes distributed eccentricity on individual rotors (such as a rotor bow, or skewed generator retaining rings, etc.), and includes induced eccentricity within an overall rotor train (such as from bearing or coupling misalignment, off-square couplings, etc.).

The largest point of discord arises when interpreting high measured rotor vibration as resonant modal excitation (or "unbalance response"), and likewise attempting to resolve it by balancing, when the real root cause is actually distributed mass eccentricity. Distributed mass eccentricity is commonly and often erroneously considered as a particular subset of unbalance. However, there are subtle but crucial differences in rotor behavior and in the approach and viewpoint needed to recognize and address distributed eccentricity versus typical unbalance. First, it is important to distinguish between the definitions of "unbalance" and "eccentricity".

"Unbalance" as addressed in this paper can be considered as any axially-localized mass that produces net-asymmetric centrifugal force when under rotation, but that doesn't otherwise statically shift the mean mass center axis of the rotor (such as the effect of a chipped blade, or variance in blade static moment weight). "Unbalance" forces can act as a source of excitation of resonant responses at system critical speeds, creating whirling (an enlarged shaft orbit) when passing through the critical(s), but it doesn't otherwise affect the rotation or orientation of the rotor in its bearings. It is important to remember that any resonant modal response requires a sufficient force to act as a source of excitation, or otherwise the rotor will pass through its system critical speed regions without a visible increase in the shaft orbit amplitude (or so-called high vibration).

"Distributed mass eccentricity" produces the same excitation effects as unbalance, but with the important added effect that it also statically shifts the mean mass center axis of the rotor relative to the rotor's geometric axis. This means that there is a net parallel offset and/or skew (depending on axial eccentricity distribution) between the rotor's geometric centerline axis and the actual mean mass axis (the "amount" of eccentricity), which itself is constant regardless of the rotor’s modal deformation. The geometric axis refers to the line connecting the concentric journal centers, based on the rigid shape of the rotor, and is the line through which torque is applied. If the rotor dynamically and elastically bends at high speed or under resonance, both the geometric axis and the mass axis would equally distort along with the elastic bent shape of the rotor. The threshold of relevance (by practical experience) when the amount of mass eccentricity can be considered "significant" enough to have an effect is generally around ~0.05 mm (~0.002 inches). (Note, by the author’s interpretation, ISO 1940-1, for class G2.5 rotors, gives a more conservative value at 0.025 mm (0.001 inches) based on modal mass at the system fundamental harmonic frequency.)

The rotor's mass axis itself can also be thought of in two separate ways. One way would represent the intrinsic rotor mass eccentricity of the stationary rotor. This is essentially the curving line connecting the radial mass center of every "slice" of the rotor, and would never change with speed. The second way would be to additionally incorporate any induced modal bending of the rotor (which would be speed dependent), and create a straight line through the centers of the modal masses of the purely rigid segments (or elements) of the rotor. The latter can be referred to as the "modal mass axis".

The key consideration when dealing with significant mass eccentricity is the recognition that every object if free and unconstrained has a natural tendency to rotate about its center of mass (by conservation of angular momentum). Any object rotating in this manner is perfectly balanced, and would produce no forces against any constraint holding it in space. Likewise, any object not rotating about its center of mass must be forced into maintaining this "unnatural" state by an imposed constraint, and will produce cyclic forces against its constraint with each revolution. For horizontal rotors, the applicable constraints are the rotor bearings and the force of gravity holding the journals in the bearings at the points of contact. In multi-rotor trains, adjacent rotors/couplings also act as constraints.

When dealing with a rotor with significant mass eccentricity, the rotor's natural tendency is to rotate about its mean mass axis, which incorporates any eccentricity. However, the rotor is constrained by gravity in its bearings and is forced to spin about its principal rotation axis defined by the line connecting its journal centers, assuming the journals are concentric. Likewise, torque is applied concentrically to the rotor about this same principal rotation axis, either transferred concentrically through an adjacent coupling or via turbine blades. The net result is the rotor being maintained in a forced unnatural state of non-centroidal rotation. This requires a perpetual force to be applied by and against the constraints (bearings), as a force-pair to the reactive centrifugal force being generated by the mass eccentricity of the rotating rotor. Unlike "unbalance", which produces enough centrifugal force only to excite the resonant modal responses at the critical speed regions, the reactive centrifugal force from distributed mass eccentricity is sufficient to affect the rotor motion through much of the speed range, in a manner proportional to increasing speed. These responses would be considered the "rigid modes" of the rotor (lateral translational and conical or "rocking"). These rigid modes are entirely dependent on the presence of mass eccentricity and its axial distribution, and cannot and should not be treated in the same manner as resonant modal excitation from "unbalance".

Generally speaking, the traditional theoretical-based approaches to resolving rotor vibration tend to focus on unbalance as fundamental, and on mass eccentricity as secondary or as simply another form of the same thing. However, by experience, whenever distributed mass eccentricity is present on a rotor or within a rotor train, it must be recognized and resolved as the fundamental problem before dealing with any modal excitation from unbalance. This applies equally to rotor trains installed in the field and to individual rotors on a balancing machine in the shop. It could be argued that recognizing these differences (and evaluating the presence of eccentricity) is actually the most relevant and critical when balancing a rotor with significant mass eccentricity in a high-speed balancing facility, prior to installation in the field. Furthermore, it is also crucial to distinguish that a rotor with significant mass eccentricity behaves differently at speeds below the fundamental system critical speed (first critical) and at speeds above it.

2 Common Points of Discord between Theory and Practice

In a wider sense, the differences between the classical interpretation of unbalance and the focus on eccentricity presented here originate in the application of rotor condition **assumptions** used within theoretical rotordynamics for predictive modeling purposes and to simplify mathematical modeling. This is not to point out a problem with modern classical rotordynamics theory, which is well proven in its general predictive ability. Rather the discord arises with the practical means of application of the theory to troubleshooting and diagnosing problems within operating machines, and in creating practical solutions to remedy these problems. This can occur when applying assumptions regarding often-unverified real-life rotor conditions, when the rotors do not adhere to the boundary conditions and assumptions required within the theory (namely rotor concentricity/symmetry). This applies especially to the measurement and evaluation of 1x and 2x rotor eccentricity, as well as coupling perpendicularity, journal concentricity and issues arising from bearing misalignment.

2.1 Rotor Reference Frames and Axes

Generally speaking, the standard theoretical models upon which rotor vibration behavior (and subsequently rotor balancing) is understood assume a concentric rotor, with added static and dynamic "unbalances" that are reduced to point masses. For an otherwise concentric rotor, this approach is accurate, and creates no issues. For eccentric rotors, it can lead to problems. Distinguishing unbalance and eccentricity becomes important when recognizing rotor behavior within the speed region of the fundamental system harmonic frequency (first critical speed); namely the rotor's center of rotation switches from a forced centering around the journal (geometric) axis to a natural re-centering and re-orientation around its mass axis. As previously described, when significant eccentricity is present, there are two initial reference axes of note within the rotor itself: the journal (geometric) axis about which the rotor is constrained within its bearings, and the true mean mass axis that governs the rotor's natural tendency of rotation.

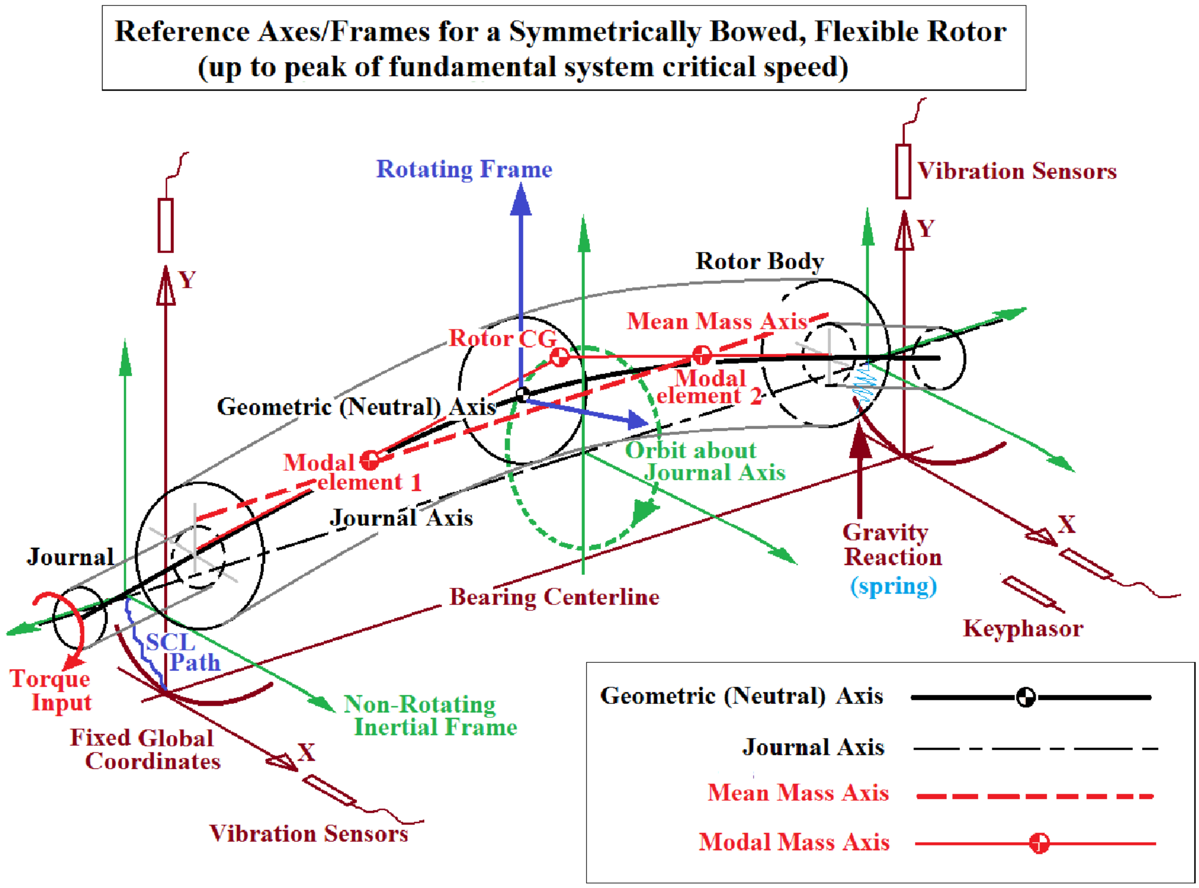

Figure 1. Reference coordinates and axes within an eccentric/bowed rotor

Note that when in dynamic motion, there are five relevant axes or frames of reference: 1) the stationary global coordinates that represent the standstill rotor shaft centerline position within each bearing (the proximity sensors remain fixed to this reference), 2) the non-rotating but slowly shifting "inertial frame" representing the "static" axis line connecting the rotor shaft centerline at each bearing through all speeds (the "virtual" center points of the shaft orbit), 3) the "geometric axis" or neutral axis representing the concentric geometric centerline of each radial "slice" of the rotor through which torque is applied and transferred (including dynamic bending), 4) the "journal axis" which represents the line connecting the journal centers not including any dynamic bending (this can also be called the principal rotational axis), and 5) the mass axis as previously described. When the rotor exhibits only rigid-mode behavior (no dynamic or resonant bending), the "geometric axis" and "journal axis" are coincident. When the rotor is in motion, the geometric axis (transmitting torque), journal axis and the mass axis (and the resulting forces in reference to these axes) are most easily recognized and described within a rotating reference frame (Figure 1).

Below the first system critical speed peak, the rotor maintains its forced constrained rotation about its journal axis. This leads to the increasing rotor forces and whirling seen while accelerating toward the critical speed peak, as net centrifugal force arises from any eccentric mass. Upon passing the critical speed peak and at higher speeds, the rotor achieves a new mode of rotation self-centered about its mean mass axis, to the extent that bearing clearances allow, in accordance with its natural tendency to rotate about its center of mass, based on its eccentricity distribution.

2.2 Points of Discord within Applied Rotor Balancing

Balancers often carry and apply a number of common theoretical modeling assumptions to practical shop and field balancing practices that can lead to unrecognized errors and create problems in the field. One of the points of discord with traditional theory (or modeling) relative to real-life rotor balancing is the consistent focus on only a single axis of rotation for all speeds and conditions, generally based on the stationary global coordinates. A second is the assumption that all balancing performed both below or above the first critical speed is equivalent. This overlooks the switch in rotation axes through the first critical speed region, and therefore implicitly assumes that the journal axis remains the sole reference frame for all speeds, and that all measured dynamic rotor vibration amplitudes at all speeds are caused by unbalance relative to this reference axis. The balancers then focus their efforts on resolving only dynamic modal displacement responses observed and measured at the journals (at the bearings as points of constraint). The rotor is assumed to bend or deflect at any speed by centrifugal force(s) acting radially out from the "inertial frame" axis line. Therefore, this approach diminishes the relevance the shaft centerline path and recognition of speed-dependent rigid mode responses as indicators of unresolved mass eccentricity.

A similar understanding can be applied within assembled rotor trains as a whole, where bearing misalignment can lead to induced bows in flexible rotors when the couplings of adjacent non-parallel rotors are pulled and bolted together. This situation is quite commonly seen, particularly when rotor coupling faces are unknowingly off-square, but still used to set the bearing alignment via standard 16-point coupling gap and rim measurements. The off-square couplings combined with resulting misaligned bearings lead to induced "static" bows within the rotor train and unexpected vibration in operation, often seen as shaft whirling at outboard bearings or as seal rubs as the rotors attempt to "naturally" self-align under increasing inertia and applied torque in accordance with the overall mass axis of the rotor train. Attempts to field-balance such a condition believing it to be rooted in "unbalance" are not often successful, and can actually create larger long-term problems or damages within the rotor train due to imposing additional internal cyclic bending moments and corresponding axial forces on the rotors.

Shop balancers also often neglect the relevance of the self-orientation of the "inertial frame" which represents the natural shift or skew in space (within bearing clearances) of the principal rotation axis, dependent on the distribution of mass eccentricity (observable in the shaft centerline path). When significant eccentricity is present, balancers will inadvertently balance an eccentric rotor about its mass axis when at speeds above the first critical speed region. This can still lead to a very well-balanced rotor when it is spinning unconstrained and solo in the balancing facility. The problem arises upon installation in the field when this rotor is installed within a rotor train and constrained by adjacent rotors to maintain rotation about its journal axis even above its individual fundamental critical speed.

The balance weights that were placed in the shop during the balancing process above the critical speed region (relative the self-orientation governed by the rotor's mass axis) now create an unbalanced condition in the field. The rotor will still naturally tend to self-align to its mass axis, but if constrained against doing so, it will require increasing forces proportional to speed to maintain that "unnatural" state of rotation. Upon the assembly and restart of a unit in a power plant following a major outage, this can frequently lead to unexpected or stubborn vibration with a rotor that was supposedly “balanced” in the shop. These forces will be reflected as either high dynamic vibration amplitudes where the rotor has clearance to whirl, high bearing seismic forces or high bearing metal operating temperatures where the bearings constrain such motion, or will be absorbed in more flexible rotors as internal cyclic bending. Such internal bending can create hidden damage over time, especially in generator rotor materials and other assembly components.

The primary focus in balancing should first be the evaluation of mass eccentricity on every rotor. If 1x evaluated eccentricity near 0.050mm (~0.002 inches) or greater is seen, then the balancer's focus needs to be placed on the often-overlooked overall static asymmetry of the mass axis of the rotor relative to its journal axis, and on rigid-mode behavior at speeds below the fundamental system critical speed range up to the critical speed peak amplitude. The rigid modes of the rotor must be addressed and "balanced" first at “pseudo high speed” (Ehrich, 1993) to avoid the source of unexpected dynamic behavior (the change in rotational axis and self-centering of the rotor) when the rotor is accelerated to speed at and above the fundamental system resonance velocity (1st critical speed).

Another important area of consideration in balancing is the axial distribution of correction weights. In standard balancing methods, the first critical speed response is addressed first, and is usually compensated with a single weight placed at the midplane (N-method) (Bishop, 1972). In flexible rotors with significant eccentricity, this type of correction for the first critical response results in a too-concentrated weight, and acts to bow and distort the rotor around the midplane. In effect, the balancer is inadvertently attempting to push and bend the rotor's geometric axis to align the journal centers to coincide with the rotor’s mass axis. This results in deforming the rotor and introduces internal cyclic bending moments within the rotor body. This can be particularly problematic for longer term operation of generator rotors, which can experience premature fatigue in internal insulation and windings, leading to electrical shorts and other damage. Also, corresponding axial forces generated from this cyclic bending may create forced or resonant excitation of free standing turbine blades.

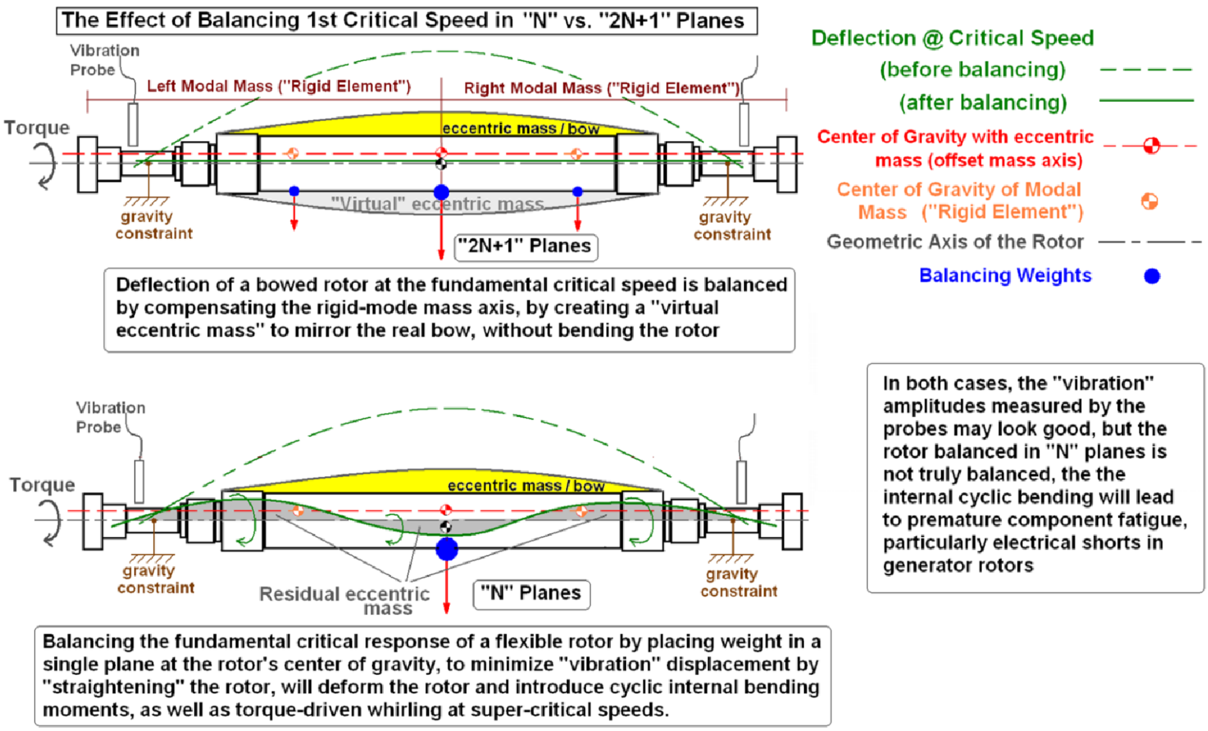

Figure 2. Comparison of post-balancing condition between balancing in N (bottom) and 2N+1 (top) planes

When balancing rotors with mass eccentricity distributed across a sizeable portion of the rotor body (any permanent bow or distributed asymmetry on the rotor between the bearings), the rigid-mode eccentricities need to be statically compensated around the principal rotational axis by mirroring the eccentricity distribution without otherwise bending or distorting the rotor (Figure 2). In a balancing facility, this must be performed at rotational speeds at or below the fundamental critical speed of the rotor-bearing system, restoring the symmetry of the rotor about its journal axis and eliminating the rotor's rigid-mode responses. This is necessary to prevent the rotor from switching its center of rotation to the mean mass axis once above the critical speed. To achieve this goal for significantly eccentric rotors, it is necessary to follow a new balancing method using 2N+1 balancing planes (similar to a patented method applied in practice by GE (Ehrich, 1993), where "N" is the highest mode of the rotor within its designed operating speed range). This also means that the rigid modes of a bowed rotor must be balanced at minimum in three balancing planes (Racic, 2014a).

2.3 A Look at Real Rotor Behavior

To better conceptually define the idea meant by “rigid mode” behavior in this paper, consider the motion of a bowed flexible rotor at higher speed which is undergoing some amount of induced elastic bending while in some orbital lateral translation or pivotal rocking motion. We can conceptually dissociate this total rotor motion into two parts, a rigid component and a flexible component. The “rigid-mode” component of the motion is representative of the dynamic motion of the rotor in its innate bowed shape. This is generally limited to motion of the journals within the oil film of hydrodynamic bearings. The flexible or elastic-bending component is representative of any superimposed bending deviation of the rotor from its innate shape. The rigid-mode rotor motion is tied purely to any distributed mass eccentricity on the rotor and to its effect on the position and skew of the rotor’s mean mass axis relative to its inertial frame.

The elastic-bending component is tied to the centrifugal force generated at any given speed by any points of “unbalance” on the rotor. This can additionally include the centrifugal force generated by distributed mass eccentricity, and also includes any amplified harmonic modal responses from resonant excitation. There is some conceptual overlap in this visualization, in that mass eccentricity both affects the static symmetry of the rotor, and can act as unbalance generating a centrifugal force to induce bending deflection of a constrained rotor (Racic, 2014b). In diagnosing the cause of observed vibration and in identifying an ideal balancing solution, the rigid-mode behavior should be viewed and treated as fundamental, and the elastic-bending component should be secondary.

It has been experimentally shown (Zhyvotov, 2011) that in a torque-driven rotor at speeds below its first system critical speed peak, its mass center axis (intrinsic rotor mass eccentricity) is synchronously rotating around its geometric axis. Above the first system critical speed, the rotor's geometric axis is rotating around its mass axis. These rotations are referenced within the rotor body itself, best viewed in a rotating reference frame centered within the rotor body. At the same time, the rotor’s geometric axis is laterally translating (essentially synchronously) in an orbit around the rotor’s principal rotational axis (the straight line connecting the journal centers). This is best referenced from the non-rotating "inertial frame" represented by the shaft centerline position at each bearing (the "virtual" center of each orbit).

When an eccentric rotor is rotating at speeds below its system critical speed range, it behaves as in the upper image shown in Figure 3, since the rotor journals are held and constrained by gravity in their bearings. Above the peak of the first critical speed range, the rotor's natural tendency is to rotate around its mass axis, as shown in the lower image in Figure 3. When rotors are unconstrained, rotated in free space, or rotated vertically, (e.g. like a spinning top, or vertical hydro-turbine-generators or other vertical machines), the natural tendency of rotation would resemble the lower figure at any speed and always naturally self-center around its mass axis.

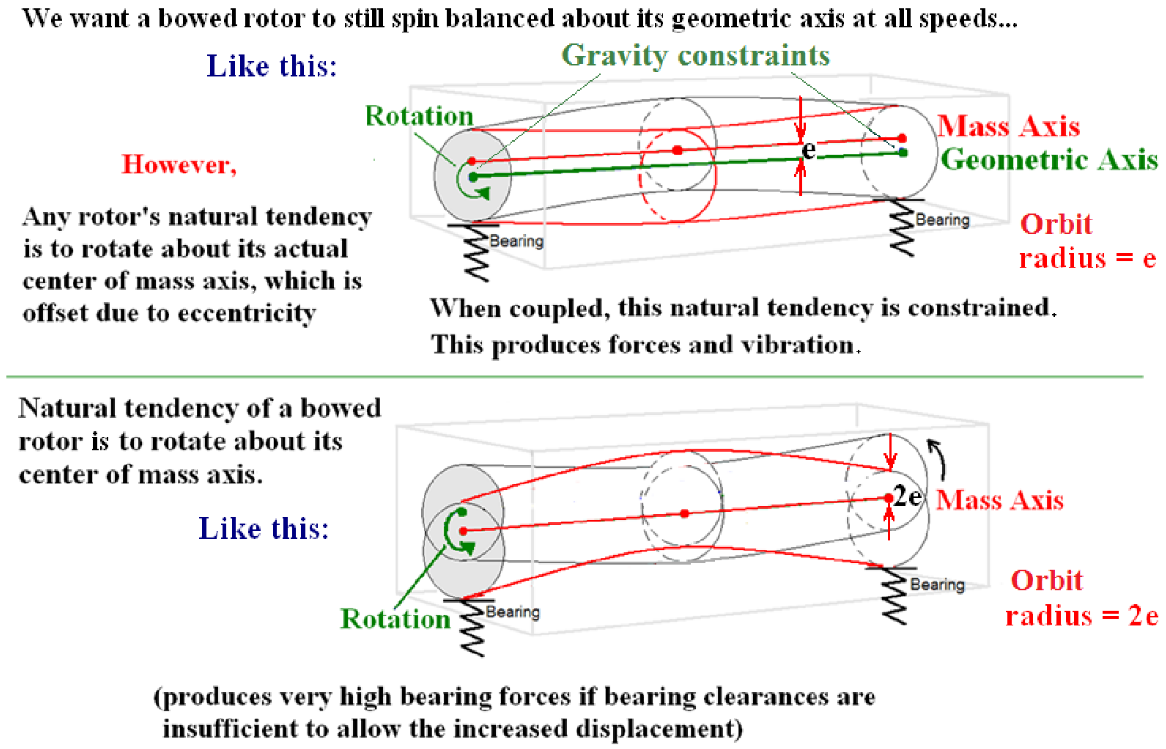

Figure 3. The natural behavior tendency of bowed rotors (top) and (bottom) the first critical speed

For a better understanding of real life vibration problems the combined turbine-generator rotor-bearing system can be thought of as comprising two interconnected equilibriums, one being the non-rotating "static" equilibrium between the shaft and the bearing (best viewed in an inertial reference frame), and the other being the rotating "quasi-dynamic" equilibrium of the forces in the spinning rotor itself (best viewed in a rotating, non-inertial reference frame). These equilibriums are "cross coupled" (Figure 4) and affect each other, and instability in one can produce instability in the other. The non-rotating, "static" equilibrium of the rotor position in the bearings remains generally stable (only following the shaft centerline path). It is held by gravity load which is constant, and oil hydrodynamic forces which govern the rotor's elevation in the bearings. The rotating "dynamic" equilibrium remains referenced to the geometric center of the shaft or its neutral centerline.

The dynamic forces originating from rotor rotation/orbit are vectorially summed with the static forces from the static equilibrium in the bearings. The net summation then governs the position and orientation and stability of the rotor in its bearings. In an eccentric rotor, the internal moments and forces dynamically generated by any eccentric mass (tangential forces in particular) will at some sufficient speed govern the rotor's orientation in "search" of its static bearing equilibrium. If gravity loading on a bearing is low, these forces can potentially create subsynchronous rotor instabilities.

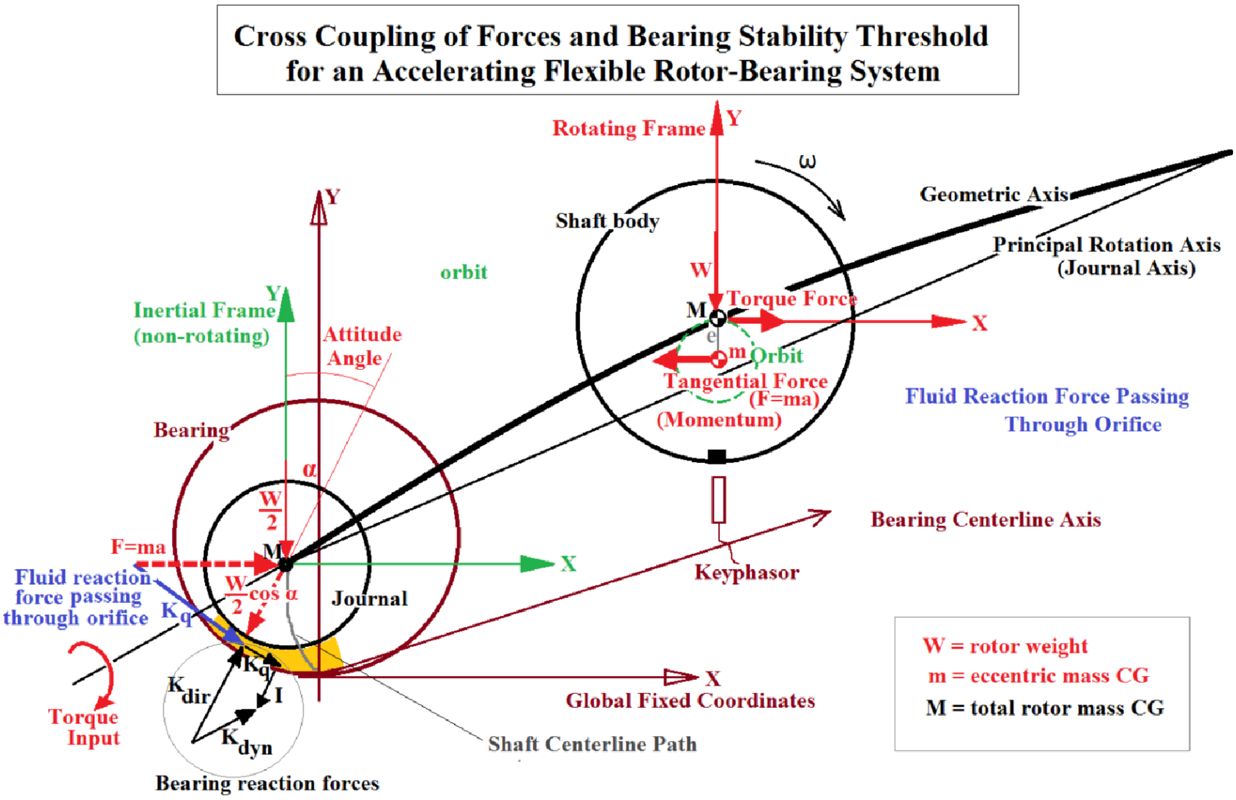

Figure 4. Cross coupling between "static" and "dynamic" forces during rotation under torque, the static best viewed in an inertial reference frame (left) and the dynamic in a rotating reference frame (right)

The real generated vibration of a flexible rotor is not a single simple motion and is not truly as modeled by a linear mass-spring-damper analogue, since the "vibration" of the rotor itself is really a predominantly synchronous translational motion in an orbit, without any real cyclic bending oscillation of the shaft itself (if the orbit is circular). That said, there still is a standard radial oscillatory component within the total rotor-bearing system response at the bearings (points of gravity constraint) "seen" by vibration sensors located in a global (fixed) reference frame. Although a "vibrating" rotor is really undergoing a combination of synchronous rotations, its bearing support (the oil film primarily, but also the shell, housing and pedestal) can be viewed as a linear, non-rotating spring under a "static" rotor gravity load. This load is in a line of action matching the attitude angle of the rotor in the bearing, and is summed with the cyclic dynamic load from the translation orbit.

3. Rotor Balancing - Modified Approach for Bowed and Eccentric Rotors

The key requirement of the rotor itself for balancing when any significant eccentricity is present is that the rotor must have at least three balancing planes, two endplanes and a midplane. If a rotor has more "significant" body runout (>0.050mm, or > 0.002 inches, 0-peak), but only has two balancing planes on the ends, then a third plane, either a machined grove or bolt-holes must be added at the midplane. If attempting balancing in the endplanes only, it can be practically guaranteed that the rotor will cause problems in the field after shop balancing, even if the vibration amplitudes in a balancing facility look successful. If a third, center balancing plane cannot be added due to material conditions or potential thermal stresses, then the only chance of good operation in the field is to remove the eccentricity in the shop either by machining the full rotor to a new centerline, by "truing" the centers, or by thermal straightening, to bring the rotor body runout under a threshold of 0.025mm (0.001 inches), as referenced to the journals (Hidalgo and Racic, 2009).

Historically, there have been numerous proposals and experiments of balancing in three planes, but they have not yet been raised to a level of general acceptance by industry or academia to become scientifically approved as necessary for balancing large turbine-generator rotors, at least as far as the service industry is concerned (Federn, 1957; Kellenberger, 1967; Zorzi et al, 1979). To have a proper and effective way of balancing and resolving the system responses of a flexible, bowed or eccentric rotor, there was a need to develop a new balancing method (Racic, 2014a). This balancing method is a hybrid of rigid-mode balancing combined with the standard influence coefficient method, and is based on displacement readings, though it could also be applied based on measuring and vanishing bearing reaction forces (Ehrich, 1993). To account for rotor flexibility, this balancing method is based on the inverse of a particular Finite Element modeling requirement. In the FE modeling of rotors, the minimum number of nodes of modal elements required in the model to ensure that no rotor modes are missed in calculation works out to 2N+1 (Chen and Gunter, 2005).

This correlates to the minimum number of axial divisions (or balancing planes) required to divide a rotor by to ensure that each resulting division or modal element will then behave sufficiently as a rigid beam through the full speed range. A well known idea in balancing is that any purely rigid rotor can be fully balanced in any two balancing planes. By taking a flexible rotor and dividing it using 2N+1 divisions (nodal points) as in the FE model, each resulting division or element can be assured to behave as a fully rigid beam, with a defined number of constraints. As a result, each element can be properly balanced as a rigid beam in two planes. The inner planes shared by two neighboring elements are essentially used twice (Figure 5).

This means that any correction weights for a flexible rotor passing through the first system critical speed must initially be placed simultaneously in three balancing planes, symmetrically axially distributed (often 50% in the midplane, and 25% at each end-plane, at least as a starting point). This can be considered as solving the so-called "static" component of the rotor response, effectively compensating the translational rigid mode. When dealing with a more flexible rotor operating above its second mode (N=2), five planes may be used to compensate for the mode seen at the second harmonic frequency. This second mode is a result of an axially asymmetric position of the net center of mass of the rotor. It is usually a combination of rigid-mode rocking (which increases in amplitude proportional to speed once above the first critical speed range) plus the resonant bending response at the second harmonic frequency of the rotor. This second-mode response can be potentially resolved in three planes by axially redistributing the initial static correction used for the first (lateral) rigid-mode response without adding any additional correction masses. In cases of a second mode for more slender, flexible rotors, the proper axial redistribution may have to be spread among five balancing planes, and determined with additional trial runs with pairs of weights to obtain additional influence coefficients for proper redistribution. The weights given by a "single" influence coefficient calculation (from a pair of equivalent forces) are placed simultaneously as a pair within each end of a single modal element.

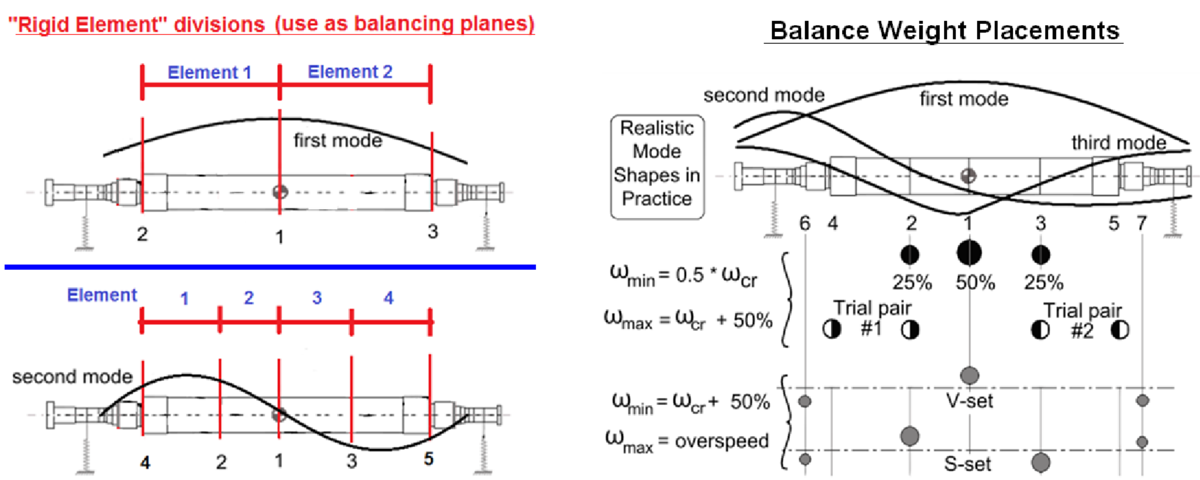

Figure 5. Representation of rotor element divisions and weight placements in the 2N+1 plane balancing method

This axial redistribution is optimized through trial runs as necessary until all dynamic cyclic reaction forces in the bearings are vanished (Ehrich, 1993), or displacements are minimized to the magnitude of the initial indicated runout. This latter procedure is not performed at the actual speed of the second critical resonant response, but rather at a “quasi-high speed” not more than up to 50% above the first critical speed (Racic, 2014a).

If a very flexible rotor (such as newer types of large generator rotors) operates at or above its first flexural mode, or its so-called third critical, then the final trim balancing to achieve desired amplitude limit values in operation is determined using modal influence coefficients, but in this stage is run and measured at operating speed. It is even better to balance at rotor overspeed to assure a well-balanced condition when placed in the field, where different bearings and support stiffness may alter the relative response and speed of the third critical. The trial weight sets must be placed following the well known modal balancing weight distributions, where the sum of the forces and sum of the moments of the weight distribution is zero, so as not to disturb the already corrected rigid-mode solution by inadvertently altering the corrected net mass axis of the rotor (Kellenberger, 1972).

Utilizing the 2N+1 plane balancing method is beneficial for balancing any rotor, but it is absolutely necessary when balancing any large flexible rotor with "significant" residual eccentricity or bowing. Overall, it is essential that the rigid mode responses are resolved first at lower speeds, before even considering balancing at higher critical speeds (Schneider, 2006). Because this balancing procedure is done mostly at lower speeds, the author of the initial iteration of this approach, (Ehrich, 1993) named it “Pseudo-High Speed Balancing.” The modified and improved balancing method described in this paper has therefore been named similarly as the “Quasi-High Speed Balancing Method” (Racic, 2014a).

As a result of using this axial distribution of balancing weights, any rotor will maintain its inherent shape, even if it is bowed, without any distortion during operation, which can be termed as running “dynamically straight”, i.e. with minimum residual error (Giordano and Zorzi, 1985). This prevents any internal cyclic bending moments, which can be a problem with generator rotors especially, and prevents high cyclic forces being transmitted into the bearings, which often happens when standard balancing methods are applied to flexible and eccentric rotors in the balancing facility, or when balancing an assembled rotor train in the field. The rotor also remains balanced about and naturally spinning and orbiting about its journal axis at all speeds, and behaves as if it were concentric, since now the journal axis is coincident with the resultant of the inherent mass axis and dynamic mass axis from balancing weights. The most important factor is that this method restores symmetry to the rotor about its journal axis connecting the coupling and journal centers, which is the line that the rotor is constrained to run about in the field when coupled to adjacent rotors. The final balance weight distribution (and their generated force vectors) will approximately mirror the distribution of mass eccentricity on the rotor.

4 Conclusion

It is important to review the assumptions used to create the common theoretical understanding and predictions of general rotordynamics behavior, and to recognize the rotor conditions and areas in practical rotor behavior and balancing that diverge from those assumptions in order to more accurately identify root causes and effective solutions to vibration problems with rotors in operation. Of course, the effects of resonances, damping and stiffness in the design of the system are still of great importance when dealing with external forces acting on rotors in operation, and these are generally well-studied and optimized by design engineers. However, many “unexpected” vibration problems have little to do with the common areas of focus (resonances and typical unbalance, damping and "instability") but rather are rooted in the presence of mass eccentricities and the rigid-mode behavior of the rotor, which is the true cause of the majority of synchronous vibration problems.

The recognition of the change of rotational axes during the balancing process as a rotor is accelerated through the fundamental system critical speed explains why the rigid-mode eccentricities of a rotor body need to be fully corrected in a balancing facility at rotational speeds at or below the fundamental critical speed of the rotor-bearing system. This means that all eccentricity is statically compensated around the principal rotational axis, restoring rigid mode symmetry of the rotor. It also shows why certain defects that can introduce eccentricity in operation (e.g. static runouts of couplings and journals) must be fully resolved by machining prior to balancing.

It has been found by practical experience on multiple occasions that verification and correction of journal and coupling runouts (eccentricities), and utilizing the Quasi-High Speed Balancing Method using 2N+1 balancing planes in the shop prevents and solves the majority of potential vibration problems at start up after a planned outage, and practically eliminates the need for field balancing of turbine-generators at start-ups after a major outage.

Based on discussions with plant managers and service shops, the estimated prevalence of flexible turbine and generator rotors with significant distributed mass eccentricities in the rotor service industry has statistically risen to over 20 percent of serviced rotors. This fact should require that current shop and balancing methods be amended to first include mandatory specific measurement and mathematical 1x and 2x evaluation of rotor and coupling runouts, including the correction of coupling deviations by machining prior to balancing, and that rotors should be balanced by the new 2N+1 plane balancing method (Racic, 2014a). Additionally, the allowable residual eccentricities stated in ISO-1940-1 for rigid rotors should be further applied to the concept of "modal elements" described in this paper, referenced to the rotor/bearing system fundamental critical speed.

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