PROACTIVE SHOP STRATEGY FOR A SUCCESSFUL TURBINE-GENERATOR ROTOR OUTAGE

Zlatan Racic and Marin Racic
Z-R Consulting
1370 S 74th St
Suite 103
West Allis, WI 53214
Telephone: (941) 527-6196
mzracic@gmail.com

Abstract: Standard power generation industry practices of inspection, machining, balancing, assembly and alignment have been developed and streamlined by OEMs for newly manufactured rotors, and implicitly assume that rotors are within ideal dimensional specifications. However, applying these standard practices in the service industry can fall short for rotors with eccentricities outside such specifications, and by design do not catch the errors that cause true dynamic problems, because they are assumed not to exist or their effects are fully unrecognized by “traditional industry practice” with regard to real rotordynamic behavior (not the rotordynamics theory based on a Jeffcot rotor). This misunderstanding leads to various problems, like reinstalling a “balanced” rotor from the shop only to be unable to run due to high vibrations upon installation. This can be followed by weeks of field balancing in an effort to salvage the situation.

A “successful” start up following a planned outage means that there will be no need for field balancing following the restart of the unit. There are two key processes that must be incorporated into an outage scope to ensure success. The first is to collect and evaluate (1x and 2x) total indicator runout (TIR) readings to identify any excessive bow or distributed eccentricity in the rotor body between the journals, and to verify and machine all rotor couplings and journals to within the criteria of ISO 1940. Any excessive off-squareness of the coupling faces or any taper or runout of the journals *must* be corrected by machining. The second key process is utilizing a new balancing method if the rotor body exceeds runout limits of ~0.002" eccentricity. Any such bowed or eccentric rotor *must* be balanced in a minimum of 3 balancing planes (or more accurately, 2N+1planes, where N is the highest mode reached by the rotor within its operating speed range).

Since most causes of dynamic problems on turbine-generator rotors are present even prior to starting the unit, almost any such problem can also be identified and fully prevented ahead of time during an outage if only a few improved steps of measurement and analysis and the proper balancing method are included into an outage scope. By proactively incorporating rotor TIR evaluation, making provisions for any necessary machining, balancing any bowed or eccentric rotor in 2N+1 balancing planes, and following OEM alignment procedures, a successful restart of the unit without field balancing can be assured.

Key words: Balancing; eccentricity; outage; rotordynamics; runout; TIR; turbine-generator; 2N+1 balancing method

Introduction: In any planned outage, the primary goals are to minimize its duration, stay on schedule, and not run into unexpected shop work and related logistics issues. Additionally, a "successful" outage can be defined as when there is no need for field balancing after the first startup of the unit, with the unit ready to run and produce power as needed without restriction.

Unfortunately in many outages, even when all work on the turbine-generator rotors was performed to "industry standards" by all initially involved parties, including shop repairs, balancing, and realignment and reinstallation, the unit is too often found unable to successfully operate upon restart, or can only operate on the edge of allowable margins, due to high vibrations, rubs or high bearing temperatures. The need for subsequent field balancing (which is not true balancing) and additional work can easily cost a plant hundreds of thousands or even millions of dollars in lost production time, labor and fuel costs. Furthermore, the emergency patch-ups like field balancing generally do not truly resolve the underlying dynamic forces, and often act only to transfer their effects onto a less-observable location or component, which can lead to larger problems in the long run.

The standard industry practices of inspection, total indicator runout (TIR) assessment, machining, balancing, assembly and alignment have been developed and streamlined by OEMs, and are designed for and applicable to newly manufactured rotors. These "industry standard procedures", in particular regarding rotor balancing and field installation/alignment, contain assumptions on expected rotor condition, individually or in an assembly, in order to expect a properly running machine with predictable rotordynamic behavior. The assumed expectations generally correspond to the guidelines set within ISO-1940 for residual eccentricity and unbalance. In a new-rotor manufacturing environment, if on odd occasion a rotor exceeds the prescribed manufacturing tolerances, corrections and exceptions to standard procedures, such as during balancing, are made individually as a "special case" condition. However, when dealing with planned outage work in the service industry, "special cases" are increasingly prevalent, where unexpected operational excursions or simply many years of rotor operation have created excessive rotor bows, eccentricities or coupling defects outside of ISO 1940 tolerances. In these special cases, "standard industry practices" are often insufficient to properly identify or remediate these problems, and by design do not catch some common faults that cause true dynamic problems, either because they are ignored or not expected to exist, or are fully unrecognized by traditional rotordynamics theory and practice.

The issue then in assuring a successful outage outcome is not necessarily whether a certain service shop or group is properly following standard industry practices (of course, they must as a minimum), but to examine the assumptions behind the standard practices and address all areas that allow problems to still slip through. Typically, in most other areas of shop work, standard industry practices are suitable. However, the two key areas in particular needing the most attention and revision are TIR assessment and rotor balancing in the shop. Incorporating small changes in shop procedures in these two areas can prevent a great majority of unsuccessful restarts that would otherwise require post-outage field balancing or other work, which passes on unnecessary and avoidable cost to plants and utilities.

Most service shop procedures and processes, which are derived from OEM procedures for new rotors, are more or less carved in stone, and reflect the basis of all terms and conditions imposed by the service industry in general. Such terms and procedures are followed in the shop to properly satisfy the pure letter of the contract, but without regard to consistently satisfying the laws of physics, and without necessarily asking "why". For a shop to incorporate necessary and different procedures to address the unique behavior of eccentric rotors, the shop needs to understand why, and recognize the assumptions hiding within the current standard procedures.

The major issues with regard TIR assessment is in how runout data is physically taken and recorded in the shop, how it is mathematically evaluated to be in compliance with shop standards, and what measures are taken to remedy runouts exceeding allowable limits. The second major issue is the shop balancing method used if significant eccentricities are present on a rotor body between the journals.

The Main Problem: As mentioned, the root of the problem is taking OEM-based shop procedures designed for new, concentric rotors, and applying them equally in service shops to rotors with significant eccentricities or coupling defects. ISO-1940 (class G2.5 for large turbine-generator rotors) provides rather strict eccentricity limits, though in practice, a slightly more lenient standard is acceptable. These limits define what can be considered a "concentric" rotor, and can be used to set the rotor condition limits to which standard shop practices are fully appropriate. By experience, this standard can be defined by 1x radial, evaluated eccentricity on the rotor body, journals, and coupling rims being limited to 0.001", and the perpendicularity of coupling faces to 0.0005". To some extent, eccentricity of the rotor body up to 0.002" can still allow for standard procedures to be marginally acceptable. However, by experience, probably 80% of rotors coming into a shop for service exceed one or more of these limits.

Excessive eccentricity creates problems in two ways, depending on if it exists in the rotor body alone, creating an offset or skewed mass axis of the rotor relative to its geometric axis, or if the eccentricity is in the couplings or journals. Using "standard" shop balancing methods on rotors with excessive rotor body eccentricity leads to problems in the field, although body eccentricity or bows are resolvable if using a modified and suitable balancing method (using 2N+1 balancing planes) on a high or low speed balancing machine [4]. Eccentricity or off-squareness of the couplings leads to problems when utilizing standard coupling-based rotor alignment and bearing positioning methods in the field, which commonly then creates misalignment and induced eccentricities in the rotor train as a whole.

In order to assure successful field alignment and operation, all rotor components from the journals outboard ***must*** adhere to the eccentricity limits mentioned above. Any flaws or defects in these areas must be corrected by machining in the shop prior to rotor balancing. However, to correct these flaws, they first must be properly identified by a thorough runout measurement and mathematical evaluation. All too often, shop procedures simply ignore taking runout of the coupling faces altogether, while journal and rotor body measurements are obtained, but with insufficient measuring locations and without a single common reference axis, rendering the data useless for a true determination of existing mass axis eccentricity. The rotors are then sent back to the field, without evaluating or correcting existing flaws in the journals or couplings, or without properly evaluating for body eccentricities or bows, which require a specialized balancing method if present.

At this point, with unknown coupling condition and potentially unsuitable balancing method, it is a flip of the coin as to whether the reassembled rotor train will start successfully, whereas it could have been entirely guaranteed with proper assessment and treatment in the shop.

Regarding rotor balancing, standard shop procedures (using only one or two balancing planes) are simply not effective when dealing with flexible rotors with bows of more than ~0.002", especially with regard to rotor behavior after reinstallation. The key is recognizing that for an eccentric or bowed rotor being spun solo as in a balancing facility, the rotor's principal rotational axis switches from the geometric axis to the mean mass axis while passing through the first system critical speed [5]. This change in axis alters the rotor's static elevation in each bearing, and its relative position and orientation in space. All balancing performed above the first system critical by standard methods (like placing static and couple shots) will inadvertently balance the rotor around its mass axis. This fact is generally unknown or ignored, and even so, the amplitudes and bearing forces might look good in the balancing facility.

However, when reinstalled in the field with the coupling constraints of adjoining rotors, forcing rotation around the geometric axis at all speeds, both the new balancing weights as well as the original mass eccentricity will now act as unbalance at high speed. This installed, "balanced" eccentric rotor will likely experience high vibrations and bearing forces, and may be unable to run at all. A plant will likely then resort to field balancing to salvage the situation. In some cases, final field balancing is already anticipated and worked into the outage scope.

With only three days of loss of generation to field balancing, which is a conservative estimate, the cost to a power company is at least 1% of the annual generation capacity of a unit. It is easy to see how expensive it truly is, and field balancing should not be considered as an expected last step of an outage to correct whatever unknown errors might have been missed along the way.

Field balancing is a very often misused procedure, and should only be considered as an emergency approach to improving vibration symptoms, since it is not true balancing or a true solution. Because of the lack of access to most balancing planes, field balancing can only be done for a specific location and at a specific speed, or in other words, "set point balancing". Eccentricity induced by rotor train misalignment cannot truly be balanced. Field balancing can be useful for minor corrections (trim balancing) to bring particular vibration amplitudes below required contract limits. The results are most often a visible decrease in shaft relative displacement, but often at the expense of an increase in bearing reaction forces, or the introduction of internal cyclic bending moments (invisible to monitoring instrumentation, but which still produce forces that generate damage in the long term). This creates the condition opposite to the very definition of a properly balanced rotor, which is defined as producing minimal cyclic forces on the bearings.

Fortunately, the proper solution to assure a successful outage is quite straightforward. Since most turbine-generator problems are present even prior to starting the unit, almost any such problem can be identified and fully prevented ahead of time during an outage if only a few necessary steps of measurement and analysis are included into the outage scope.

Both to maintain a schedule and to avoid field balancing, it is imperative that a plant and service shop work together to understand and incorporate necessary steps and procedures into the planned outage scope and shop service contract from the beginning. The service shop should be fully aware of any modified work requirements prior to the outage and to commencing work, to anticipate appropriate scheduling and allocation of shop resources to accommodate the required procedures.

Overall, guaranteeing a successful outage requires a change in the corporate culture of equipment users as well as at equipment service providers, understanding what must be incorporated into a workscope and why, and making sure the necessary engineering procedures and processes are understood, modified and carried out properly by all parties involved.

Points of Disconnect in an Outage Process, and Best Procedures: Overall, the best outage result is when the responsibility to follow rotors through an outage is in a single set of hands, following, reviewing and assessing all processes from shutdown, to shop inspection and evaluation, machining and balancing, as well as reviewing the turbine alignment prior to restart, and then monitoring the startup.

In practice, the responsibilities within an outage are more typically split between the disassembly maintenance group, the service shop, assembly maintenance groups (including alignment) and the plant operations group. Each group has their own standard practices (mostly following “general industry standards”) which are often totally independent of each other. In this approach, each party can be “right” based on their industry standards and contractual requirements within their own field of responsibility, but if the end result is a machine needing additional correction after startup, then it becomes “nobody’s” fault, and the responsibility and cost to make it right falls on the plant.

The first task to remedy the points of disconnect in an outage is to reassess the total outage scope and shop service contract. There are a few often-ignored activities that **must** be included and some that must be specifically performed in a predefined and agreed upon procedure, to assure a successful outage outcome.

The measurements taken by the disassembly group (especially oil bore readings, and coupling gap readings) are vital to assure successful work in the shop and of the assembly/alignment group. The work in the shop, especially proper TIR evaluation and appropriate machining corrections, is crucial to assure the success of the assembly/alignment group. Proper shop balancing is also crucial to give plant operations a smoothly running machine. In general, each subsequent group does their best with what they are given, and each follows their own standard procedures, but this alone is not enough to assure a successful outage.

Each portion of the outage procedure must be amended to incorporate the necessary steps of measurement to identify and resolve possible errors that would get passed on to the subsequent service group, whose "standard industry procedures" are designed assuming such an error does not exist.

Data Collection Prior to the Outage: One initial aspect of a successful outage is to collect full operational and vibration data during the shutdown of the unit as a baseline condition assessment. At a convenient time for the plant, prior to or at the beginning of the scheduled outage, vibration data should be recorded for a unit shutdown, from unloading, to roll down, to full stop. The findings are important to determine and guide and/or supplement the necessary shop repair scope. This also provides a base reference for comparison of the dynamic performance of the turbine-generators after the completion of the outage.

For any proper diagnostics, the unit should be instrumented with the necessary vibration sensors, which must include two proximity probes per bearing, and at least one seismic sensor per bearing. If not permanently installed, temporary sensors should be installed. This instrumentation is required to generate Bode plots, polar plots, shaft orbits, shaft centerline plots and other means of analysis to evaluate and identify faults in all components of the rotor train, as well as to verify bearing alignment.

In case only a partial outage is planned (when not all rotors or bearings are going to be removed for repair), the shutdown vibration data will point to any concerns regarding the components not scheduled for work and will point out the need for potential compromises during the unit restart, either for what should be done to optimize the unit within the imposed limitations of the outage, or for what the best-case expectations should be upon restart. Depending on the findings, any critical alterations to the planned workscope should also be identified and incorporated prior to sending the rotors to the shop.

Disassembly: During disassembly, oil bore readings should be taken to identify the initial shaft and bearing positions. If oil bore readings are not taken prior to removing the rotors, then there is no point of reference available to compare against for reinstallation. Coupling gap and rim readings should be taken upon removing coupling bolts, but prior to moving any rotors, and will point to possible coupling defects to be verified in the shop. Very often, if a rotor train had been most recently installed and assembled with some misalignment, the resulting bending moments and stresses at the couplings will slowly be relieved over years of operation in the form of subtle bending and microfretting of the coupling faces, until the coupling pair reaches a stress-free but off-square surface of contact. If these coupling defects are not observed at disassembly, or in the shop, and left unaddressed during shop work, then performing realignment based on field readings of assumed-good couplings will suggest improper bearing moves and result in further actual misalignment of the unit, which will lead to high vibrations or bearing forces, rubs or other damage.

Shop TIR Measurement: Shop TIR measurement is commonly performed, but its importance is most often overlooked, and is often done with inadequate procedures in service shops, sometimes wrongly implementing "shortcuts" as well in an effort to save costs. Often, TIR data is only recorded and used to indicate the single "high point" and corresponding phase angle at various axial locations. This procedure arises because of the misguided assumption that a bow or distributed eccentricity of a rotor can be dynamically considered as an equivalent concentrated unbalance. For flexible turbine-generator rotors, this is a completely false assumption, and the physical rotordynamic behavior is very different when distributed eccentricities are present. Sometimes only a single axial location on the rotor body is even measured at all.

Furthermore, each plane of measurement is often recorded without regard to a common reference axis or reference to either journal, with different portions of the rotor measured at different times across multiple setups on the lathe, many times with the couplings and faces skipped entirely. This issue usually arises for the convenience of the shop schedule or lathe availability, or simply to "cut corners" to minimize the cost and the time that a rotor is held in the shop. The runout data thus collected fills the data forms and satisfies the letter of the contract with the equipment owner, but without regard to the meaning and purpose of the data, which when collected without a single, common axis or phase reference is technically useless for a proper engineering evaluation.

It is imperative that all runouts (body, couplings, rims and faces) are taken within a single setup on the lathe, providing a single common reference axis and phase angle reference to evaluate any eccentricities against the center of gravity of the total rotor mass. The purpose is to ensure that no unknown or unresolved mass eccentricities remain outside the operating journal centerline axis of the rotor once installed, and that all couplings and journals are brought to proper tolerances for alignment and operation. It is of no use to measure part of the rotor with reference to one journal, and the other portion with reference to the other journal, if the journals themselves are unknowingly not concentric to each other.

In a balance facility (rotor driven with moment-free couplings), the only points of contact are the rotor journals, and therefore, eccentricity can be everything that is asymmetrically outside of the axis of the journals. Likewise, the journals are the only fixed coordinate reference in this setup, based on the sensors being mounted over the journals. However, when a rotor is coupled in a rotor train and fixed to the axis connecting its coupling centers, then eccentricity can be everything that is asymmetrically outside of the axis connecting the coupling centers. If there is any eccentricity of the coupling rims or faces, or offset between the journals and couplings, then the rotor installed in the field will be constrained to operate around a different axis than in the balance facility, and the full, otherwise well-balanced rotor will now act as a “crank” relative to the eccentric couplings that hold it.

Proper TIR procedure must entail that each rotor is examined and evaluated in fine detail for journal concentricity, taper and offset, as well as the roundness and concentricity of coupling rims, faces, fits and bolt holes, and face perpendicularity. The TIR of the rotor body should be measured at minimum at every axial location of diametral change along the rotor. For all TIR measurements, a reading must be taken and recorded every 30 to 45 degrees (preferably 12 or a minimum of 8 points circumferentially)

With all TIR data properly recorded, the most important step in the process is to mathematically evaluate the data for 1xRev and 2xRev eccentricity relative to the common reference axis of measurement, to gauge compliance with shop standards, and to determine what measures must be taken to address runouts exceeding allowable limits. Simply looking at the raw data numbers, and especially only noting the “high point”, does not actually provide an indication of the true rotor eccentricity. The 1xRev evaluation provides a measure of mass center offset (its magnitude and phase) at each given axial plane of measurement, and provides a “map” of the bow or eccentricity present on the rotor. The 2xRev evaluation provides a measure of the ovality or out-of-roundness at each axial plane of measurement.

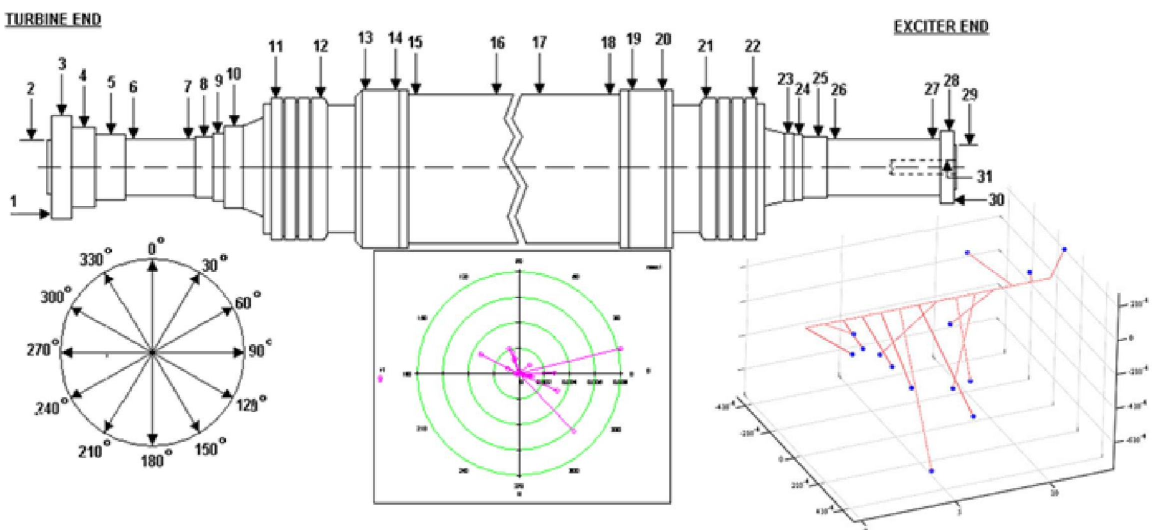

Figure 1: Proper TIR Measurement Locations and Example of an Evaluated 1x Eccentricity Distribution

Machining Corrections: Prior to balancing, any non-concentric defects found in the journals or couplings, such as any non-perpendicularity or radial offset of the couplings or any journal offset or taper must be machined to proper tolerances (including reaming or honing the coupling bolt holes to achieve new perpendicularity to machined coupling faces). The coupling corrections are imperative to allow for successful alignment in the field if using standard OEM-based alignment procedures, where bearing alignment is determined based on coupling gap and rim measurements, as well as to prevent any induced eccentricity in the rotor train when pulling couplings together, or transferring any moments from rotor to rotor once they are coupled together.

It should usually be possible to address rotor bows through balancing alone, though ***only*** if 2N+1 balancing planes are present on the rotor (where "N" is the highest mode reached by the rotor within its operating speed range). This means a typical turbine rotor operating above its first critical **must** be balanced in a minimum of three planes if eccentricity is present. Therefore, if a center balancing plane is not present, a machining solution must be incorporated. If material and thermal conditions allow, a balancing groove or boltholes can be machined into the midplane of the rotor. If it is not possible to add a machined center balancing plane, and if thermal straightening methods are not practical, then the only successful solution is to machine the full rotor to throw the centers.

Following all machining corrections, final TIR measurements of the rotor must be again taken and similarly mathematically evaluated to verify the quality of the repairs. This final TIR evaluation is also important to guide the subsequent balancing procedure in the shop.

Shop Balancing: Prior to balancing, if an attached stubshaft is required to mount the rotor in a balancing facility, runouts of the rotor and mounted stubshaft should be taken and evaluated as well to assure no eccentricity is introduced by the stubshaft.

For rotors with all evaluated 1xRev body eccentricity less than ~0.002", standard shop balancing procedures in two planes can be acceptable, although the result and process could be further improved through the use of 2N+1 balancing planes. However, if rotor body eccentricity exceeds runout limits (more than 0.002" eccentricity), the rotor **must** be balanced using 2N+1 balancing planes to guarantee acceptable vibrations in operation without the need for subsequent field balancing. This requires that the first critical speed response must be solved by placing weights simultaneously in three planes, with the proper axial distribution. For more flexible (typically generator) rotors operating above their second or third critical speeds, the full balancing process must be performed using five or seven planes, respectively (though the first critical correction still uses three planes). [4]

In general, when balancing flexible rotors with distributed mass eccentricities, a specific approach and procedure is required in order to guarantee a smoothly running unit upon reinstallation. The 2N+1 plane method (also named by Z-R Consulting as the Quasi-High Speed Balancing Method) is based on a novel view of the unique rotordynamic behavior exhibited by eccentric rotors. [5]

The Problem of Balancing Eccentric Rotors by Standard Methods: The traditional balancing viewpoint is to generally see "vibration" as a dynamic problem, and to measure and resolve dynamic responses by compensating with additional dynamic forces created by balance weights. Most rotordynamics specialists and balancers take all vibration responses as unbalance responses (as a bending deflection caused by centrifugal force).

They focus on angular placement of balancing weights to resolve it, through influence coefficients and modal balancing, but neglect or misidentify the rigid mode responses arising from distributed rotor eccentricity. This misunderstanding leads to various problems, like reinstalling a “balanced” rotor only to be unable to run due to high vibrations, or losing valuable time performing 80 to 100+ balancing runs in a high-speed balancing facility in very challenging cases.

Implicit in the traditional approaches, like the N-method [1] and N+2 method [2,3], and other hybrid methods, is the assumption that an equivalent superposition of forces from multiple unbalances or distributed eccentricities can be freely considered as a single “effective force” that can be counteracted by a single counterweight and/or a couple shot. But for eccentric flexible rotors, traditional balancing does not truly compensate the axial distribution of eccentricities, leaving residual internal moments and producing rotor distortion from too-concentrated correction weights. Rotors operating under these conditions will eventually self-correct these moments, but as a result may develop coupling face deformations or even develop cracks.

The behavior of a significantly eccentric rotor differs significantly from representing the same process on a Jeffcott rotor, where a massless shaft would freely distort and bend at the point of connection to an eccentric disk, giving the unrealistic image of the mass axis shifting into the common original line of the geometric axis and reaching a “self-balancing state”. The standard balancing approaches are conceptually related to this representation, with the static and couple balancing weights intended to generate dynamic forces to bend and push the rotor into a straight line (inadvertently about the rotor’s principal mass axis) to minimize response amplitudes.

However, 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 and precess 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”.

Before further discussion of rotor balancing, it is important to first define what exactly a well-balanced rotor is. A rotor's balance condition cannot be determined based solely on "vibration amplitudes" or displacement readings. A truly well-balanced rotor is defined as producing no cyclic bearing forces, not necessarily by generating zero displacement amplitude, which in the case of an eccentric or bowed rotor is generally not possible. However, for balancing rotors with excessive body eccentricity, even verifying low bearing force responses, while necessary, is not sufficient to properly balance the rotor to run smoothly in the field.

Unfortunately (and unbeknownst to or ignored by almost everybody), any eccentric rotor when running solo (unconstrained) naturally shifts to rotate around its mean mass axis when passing through the first critical speed region, and no longer rotates around its geometric axis at high speeds. This “re-alignment” across the first critical speed region is determined by the size and orientation of the rotor’s distributed eccentricity. If this switch in rotation axis is not intentionally recognized and prevented within the shop balancing strategy, then the rotor is unknowingly "well-balanced" in the shop around a skewed mass axis, rather than the rotor's intended geometric axis (the line connecting the radial journal centers). When such a rotor is once again coupled in a rotor train and constrained to its geometric axis, the balance weights and the original eccentricity then both generate forces and dynamic responses on the installed rotor upon reaching speed or load with torque applied, the result being high vibrations at startup.

(Many theoreticians see these events and analyze them under the guise of the "parallel axis theorem", which suggests the effects can be ignored as nothing more than a trivial change in the polar moment of inertia. Unfortunately that is not a correct or complete view of the dynamic condition in real-life rotors, since it ignores the fact that a torque moment [5] acts as a driving force of vibrations in the presence of mass eccentricity, in addition to the effect of the centrifugal force, which is considered as a "fictitious force".)

A Discussion of the 2N+1 Plane Balancing Method: A key distinction in the foundation of the 2N+1 plane approach is to consider eccentricity as a static problem as it exists on the rotor at standstill. The idea is that axially distributed eccentricities produce a mean mass axis at some distance and skew from the intended geometric axis of the rotor. The goal of balancing should then be to statically shift this mass axis to be again coincident to the geometric axis, around which the rotor will rotate when coupled. (That is, to shift the effective radial center of gravity at any given radial plane of the rotor to coincide with the geometric axis, but not to bend the rotor in any way.) This must incorporate a sufficient and proper axial distribution of correction weights to effectively mirror the axial eccentricity distribution, and in the process not produce internal moments (axial or radial) or rotor distortion when at speed, and minimize cyclic forces in the bearings.

In this way, the eccentricity of a rotor is first recognized as a static condition independent of speed and resonant frequency, which leads to a balance solution as an extension of the rigid-mode or rigid-body balancing concept that any fully rigid rotating shaft can be balanced in any two balancing planes. In all cases, the rigid modes of a rotor must be fully balanced first, before addressing residual modal deflection at high speed if it should appear. [3]

Because rotors in practice are never truly rigid, it is necessary to divide a flexible rotor into "rigid elements", in which each rigid element (or modal element) behaves through the full operating speed range as a rigid beam, each of which can be balanced in two planes. The axial division of the rotor is derived from the minimum number of elements required to define all critical speed shapes seen in a rotor in a Finite Element model, such that no eigenvalues and corresponding eigenvectors will be missed within the speed range of interest [6]. The corresponding formula for the number of nodal points (including both nodes and anti-nodes), which then correlates to the minimum number (and location) of required balancing planes, is 2N+1, where N is the highest operating mode of the rotor. For example, a turbine operating above only its first critical speed requires 3 planes, creating two "rigid elements", comprising each half of the rotor, each of which can be considered to behave as a rigid beam. A flexible generator rotor may require five or seven divisions and balancing planes. By dividing the rotor into effectively rigid components, and by balancing each "rigid element" in two planes (with the inner planes effectively shared by two neighboring elements), the skewed or offset mass axis of the rotor is shifted to be coincident to the geometric axis of the rotor, without inducing any bending or distortion at any speed.

Most rotors entering a service shop have a mixture of distributed eccentricities (which cause rigid-mode responses) and localized unbalances (which can cause bending deflection responses at higher speeds). The premise behind the balancing strategy is to first fully compensate the rigid modes (from distributed eccentricity) through appropriate axial balance weight distribution. In particular, the first critical **must** be balanced simultaneously in three planes. Once the rigid modes are resolved, and the effective principal mass axis of the rotor is brought coincident to the rotor's designed geometric axis, it prevents the switch of rotation axes through the critical speed region (since with all asymmetric eccentricities resolved, the torque moment is minimal or nonexistent [5]). This assures that the eccentric rotor remains spinning around and is balanced around its geometric axis for all speeds.

All rigid-mode eccentricities must be fully compensated for (balanced) before continuing rotor balancing at higher speeds, if remaining responses are still observed. Any balancing at higher speeds **must** use only pure modal weight distributions, such that the net sum of forces and moments of any modal weight set is zero, so as not to disturb the initial rigid mode solution. A generalized balance weight distribution is shown below in Figure 1.

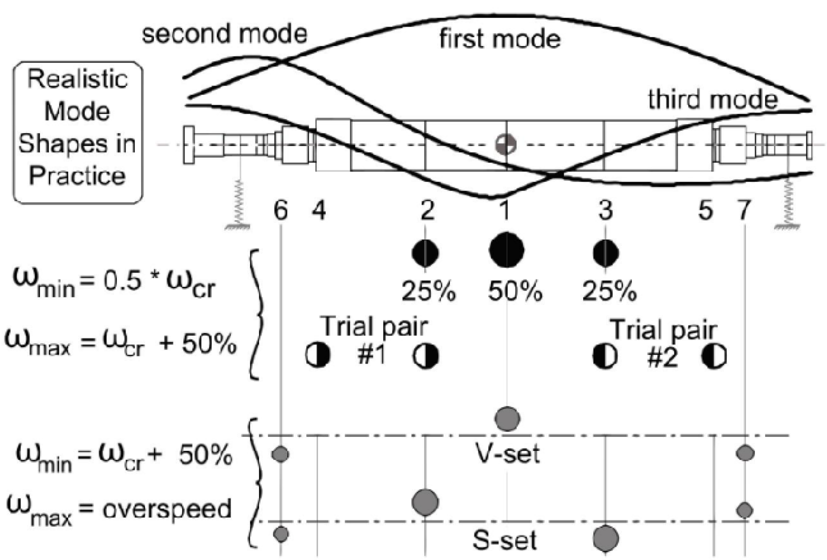

Figure 2: Identification of Balancing Correction Planes and Possible Weight Distributions

In most cases, however, by successfully resolving the rigid modes at lower speeds, the modal/resonant responses at higher speeds will be practically eliminated as well. This concept is opposite to standard current balancing methods [1, 2], which focus on or require modal balancing at speed. In the 2N+1 method, the modal responses are not the focus, but are automatically resolved by intentionally and properly resolving the rigid modes at lower speeds. By contrast, the traditional balancing premise [1] is to coincidentally resolve any rigid mode eccentricities (if they weren't ignored in the first place) while focusing on balancing modal responses at speed. For highly eccentric rotors, this approach often requires an excessive amount of weights when balancing by influence coefficient methods, and can require many more runs to find a solution (which even when found, is less than ideal if eccentricities are present).

The amount of weights used in the 2N+1 balancing method is similar to or less than the amount that would be used in standard balancing methods, with the primary difference being in their axial distribution. This balancing procedure is furthermore accomplished at speeds less than 50% above the first critical speed, saving time and energy costs in the balancing facility. If the rotor response in a Bode plot remains flat (that is, not up-sloping, but it need not be at zero) up to this speed threshold, it can be assured that the rotor will remain balanced up to running speed as well, without the necessity of balancing each subsequent critical at speed, as in traditional methods. Another beneficial aspect of utilizing this balancing method is that it can be used also in balancing flexible rotors on low speed balancing machines.

If the final balancing weight distribution results were to be mapped out as a curve, it should ideally be a mirror image of the curve of the runout reading shape of the rotor. The final product of this total approach is a shifting of the mean mass axis of the rotor to coincide with the geometric axis, essentially simulating the condition of a fully concentric rotor, without dynamic distortion, and eliminating internal moments (from applied torque) that would result from axial asymmetry of eccentricities or from traditional placements of balance weights. The rotor can be considered "dynamically straight", as it maintains its intrinsic static shape at all speeds (even if that shape is bowed). A rotor balanced in this way is guaranteed to run smoothly once properly aligned and installed in the field following OEM instructions.

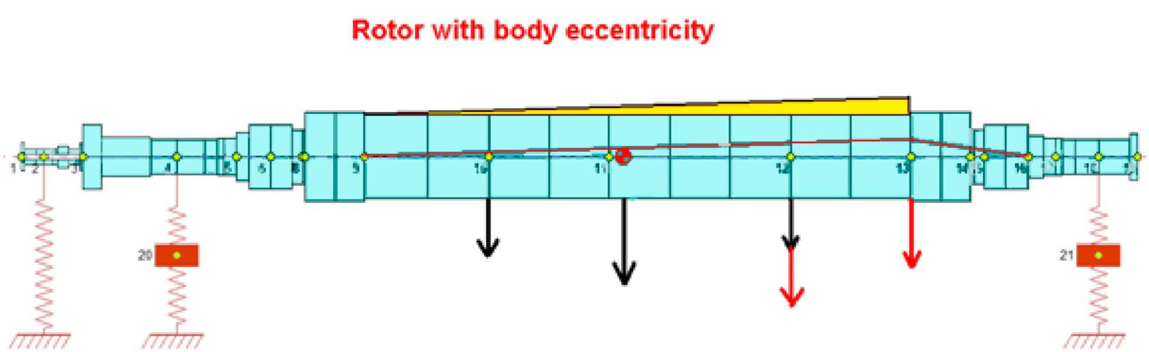

Figure 3: Final Balance Weight distribution for Sample Rotor with Distributed Eccentricity, using 2N+1 Balancing Planes

A detailed procedural description and discussion of the premise and reasoning behind the Quasi-High Speed Balancing Method, including a more thorough explanation of the unique rotordynamic behavior of eccentric, flexible rotors can be found in previous work by the authors [4, 5].

Incorporating Finite Element Analysis: In the course of an outage, another activity that can be beneficial in many cases is Finite Element modeling and analysis of the rotor train. This is particularly relevant if an operating unit had encountered notable problems during prior cycles of operation, such as high bearing temperatures or bearing damage, or excessive vibration. Many of these problems can arise from improper bearing alignment or an improperly set catenary curve, or from holding speeds improperly set at a system resonance (lateral, axial or torsional), and in some cases from deficient bearing design. Such errors (in addition to other causes) can also lead to blade cracking or breaking of blade lashing wires. Analyzing and resolving the root cause of encountered faults or failures as part of a planned outage workscope, including optimizing the catenary curve and bearing alignment (which has often been modified ad hoc by practitioners in the field applying bearing preloading to achieve "bearing stability") to minimize all transferred moments, can greatly improve unit reliability over the long term, preventing losses from unexpected damage or repairs and the associated lost production.

Installation and Alignment: Since large turbine-generator rotors all contain rigid couplings, problems with coupling eccentricities and rotor alignment are significant, as they transfer torque and bending moments through the rotor train, potentially becoming a source of vibrations and other rotordynamic problems, including subsynchronous instabilities. If all couplings are verified in the shop to be within proper tolerances, then when rotors arrive back on site following all shop work, following standard OEM-based alignment methods and adhering to the original OEM catenary guidelines should provide a good result.

In addition, any bearing "preloading" should be discussed with the assemblers as to why it is done. Bearing preload is typically incorporated during assembly to reduce possible bearing instability in operation, though such apparent instability itself arises most often from unresolved eccentricities in the rotor train, from individual bowed rotors or induced from misaligned bearings. Applying preload might appear to work temporarily, but in the longer term it bends the shaft and can possibly wipe the bearings. In most cases, preload should not be necessary if all rotor train eccentricities were properly addressed and resolved in the shop. 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.

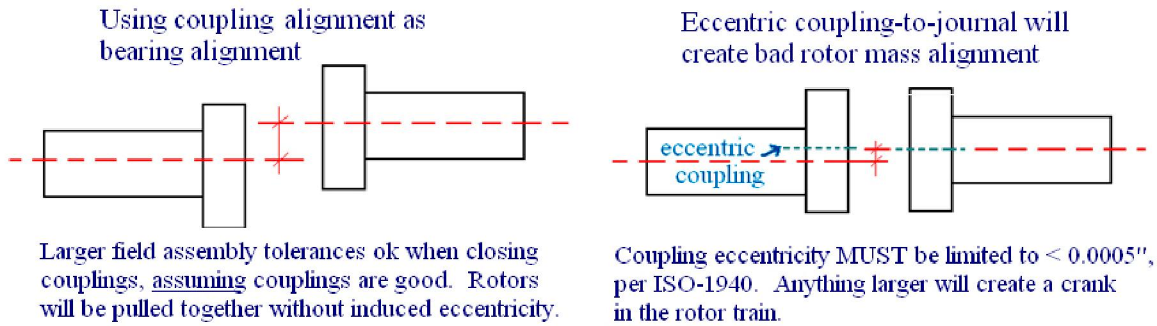

Figure 3: Effect of Offset Coupling Rim or Coupling-to-Journal Eccentricity on the Resulting Rotor Mass Alignment

It is important to note that the assembly tolerances for coupling readings for field installation are usually many times greater than shop tolerances for coupling condition (referenced to journals). Shop tolerance for coupling face perpendicularity, for instance is 0.0005" or less, while assembly tolerance for coupling gap readings during alignment is usually 0.002" to 0.003". However, these larger assembly tolerances also carry the ***assumption*** that the rotors are within eccentricity tolerances (namely the rim and face) when using the rotor itself in order to measure, position, and verify its own bearing alignment. Clearly, if using off-square or eccentric couplings for alignment, the result is generally much less than ideal.

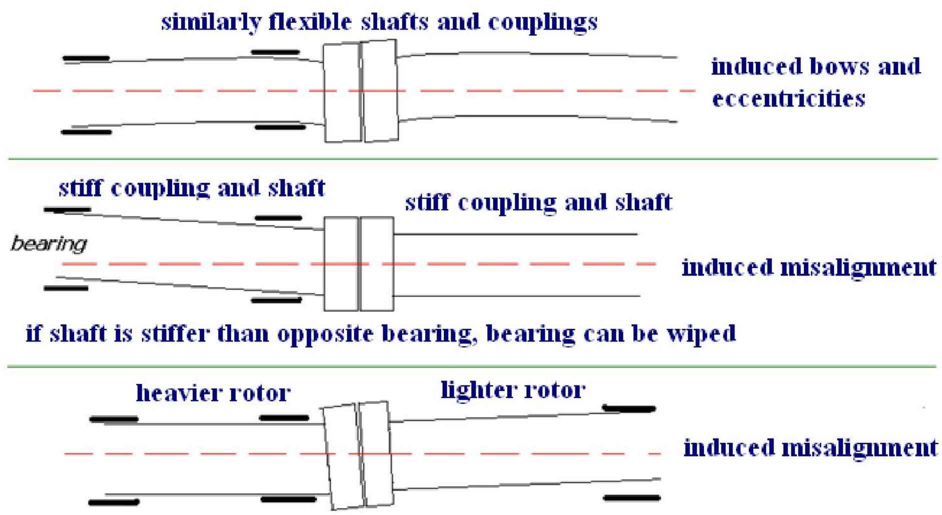

Figure 4: Effects of Off-Square Couplings on Rotor Train Alignment

Without a proper, full TIR evaluation of the couplings, excessive existing coupling defects that were not detected and corrected become buried and unseen within the tolerances for field assembly. If moving an outboard bearing to achieve a "fair" coupling gap of 0-0, an off-square coupling face will introduce error at the ratio of the rotor length to its coupling diameter. If the L/D ratio is 8, for example, two mils of non-perpendicularity of a coupling face (with zero rim offset) will wrongly suggest positioning the outboard bearing 16 mils out of alignment. If both adjacent couplings have such unknown flaws, and are positioned only to the edge of the 2 to 3 mil assembly tolerance, the induced misalignment can be easily be 20 or 40 mils, the effect of which will only become evident at the unit startup when torque is applied.

Ultimately, the assembly group simply performs their task to the best of their ability given the condition of the rotors provided. If neglecting to identify or correct or compensate for excessive runouts in the shop (that is, to machine all couplings to specs, and to bring the mass axis coincidental to the journal axis during balancing), the resulting likely high vibrations force the need for subsequent "balancing" in the field - of course, at the expense of the equipment owner. This cannot truly resolve the rotor train condition, but even if "successful", only masks the symptoms of high displacement at the expense of transferring the dynamic forces elsewhere into bearing/pedestal vibration, coupling bending moments, high bearing operating temperatures, or creating other long-term damaging conditions.

Conclusion: Misunderstanding the true physical behavior of rotor bearing systems leads to the creation of various insufficient shop procedures and processes (which are furthermore disconnected from field assembly activities), and as a consequence costs the power industry millions in an attempt to mask vibration problems by field balancing. That is in addition to the even greater cost of lost production time while applying "field solutions" to problems for which the root causes could and should have been resolved in the shop.

The root of the problems encountered in an unsuccessful post-outage restart is almost always in incomplete work that was performed in the shop. The root of the insufficient shop work usually lies in applying shop procedures designed for new concentric rotors to bowed or eccentric rotors with coupling defects, not understanding the unique rotordynamic behavior generated by the presence of mass eccentricity. When rotors with such defects arrive in the field for installation and alignment, the alignment procedures that are otherwise fine for new, concentric rotors, lead to incorrect bearing positioning, and installers unknowingly set the rotors in a crooked alignment line. At the same time, rotors that might have appeared "well balanced" in the shop run with high vibration in the field, due to utilizing OEM balancing methods designed for otherwise concentric rotors, which don't properly resolve distributed rotor body eccentricities or bows.

By many current shop practices, the eccentricity limits specified in various documents fulfill the specific requirements for a given particular step in the overall shop process as it stands alone (or separately for that in the field assembly), but each subsequent procedure includes only inferred assumptions of quality tolerances that were ultimately never verified in previous steps, and the total procedure does not consider the combined, cumulative, absolute requirements to assure acceptable vibrations of the assembled turbine generator during a startup after an outage.

The procedures presented in this paper are designed to fit readily within a standard outage workscope, and only add minimal time and expense to acquire all required data for proper analysis, but are guaranteed to save the plant substantial costs for future repairs or downtime in the long run. The methods and approach differ from many standard shop procedures, and are designed to be more thorough, and to eliminate the chance of any problems or "surprises" once back in the field. When the rotor leaves the shop, it is assured that it will be ready to run upon proper installation. Significant value can clearly be had by incorporating key verification steps into a total outage procedure, saving substantial time, cost and headaches in the long run.

If proper and thorough evaluations of rotor TIRs and coupling eccentricities are performed, in combination with using an improved balancing method utilizing 2N+1 balancing planes, there will be no need for field balancing after an outage, and the unit can be returned to service without delay.

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