Proactive Shop Strategy to Avoid Field Balancing after a Major Outage

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Presentation Transcript

[slide = The main problem] We're going to take a look at a type of problem that happens all too frequently during a planned outage, which is a forced shutdown or trip of a unit within the first restart, due to high vibration, which then leads to days or weeks of delays with field balancing or other repairs or modifications. We'll look at a couple of the common and most preventable reasons why this happens, and the steps that can be incorporated into an outage plan *ahead of time* that can guarantee a smooth restart the first time without the need for field balancing.

[slide = High Cost of Post-Outage Work] It seems that in many outages at many plants, this pattern of post-outage vibration troubleshooting and field balancing is even anticipated or is expected as unavoidable, or as a matter of unforeseeable luck, even though it's obviously costly. If we look at the number of plants this happens to, this kind of problem costs the industry many millions of dollars per year, considering the cost in lost time of power production and spent fuel while attempting field balancing to allow the unit to operate.

So let's ask, why is this type of situation a common occurrence in the power industry? And how is this type of outcome even possible if all the work in the outage was performed to the requirements in the service contract and to official OEM specifications, and by "customary" procedures? And more so, how can we reliably avoid this situation after an outage?

[slide = Current Rotor Service Procedures 1] The reason why this overall pattern is accepted by the power industry, while admittedly undesirable, is that everyone assumes that the shop outage practices are already adequate and optimized and as good as they can get, given that they've been performed the same way, and refined and streamlined for the past 50 years or more. And besides that, these methods have been researched and developed by OEMs, and are modeled and supported by academia. So then where can we still improve?

[slide = Current Rotor Service Procedures 2] One key element is that the shop outage practices and balancing methods, and field alignment methods and tolerances were all created by OEMs based on the installation of NEW machines coming from the factory with properly concentric rotors and perfectly square couplings, and these corresponding shop and balancing and field alignment procedures ***require*** that kind of rotor condition in order for the methods and assumptions to apply. And for these new, factory rotor conditions, the standard methods **do** work perfectly well.

[slide = The Risk of Assumptions] The problem comes from applying these same methods and assumptions to used rotors in the service industry, where those assumptions on rotor coupling and eccentricity condition are not necessarily true, and not absolutely verified. In particular, perhaps the biggest problematic assumption is thinking that any found defects can always be balanced later. But this approach is never as successful, and always more expensive, than resolving it in the shop.

[slide – Two Key Causes of Post-Outage Vibration] Through dealing, after the fact, with a great number of cases of post-outage difficulties with high vibration, and after analyzing the root cause and source of the problem, we can say that the great majority of vibration problems following planned outages all come from two key root causes, and often both are present at the same time. When we investigate, the root of these problems is really in static causes which are integral to the rotor, which can be proactively found and resolved in the shop.

The first cause is unmeasured, unobserved, and uncorrected non-perpendicular rotor couplings, and the second cause is unidentified or improperly balanced residual distributed mass eccentricities. Again to emphasize, because these are **such crucially** important problem areas, the key problems are off-square rotor couplings, and unresolved distributed rotor mass eccentricities. If we eliminate these problem areas, nearly every planned rotor outage **can** have a successful first restart.

[slide: Rotordynamic Effects of Eccentricity] To look at why eccentricities and coupling defects create so many problems, we need to take a look at how these two conditions of off-square couplings and rotor body eccentricity affect rotordynamic behavior.

We can define rotor eccentricity as any distributed mass that noticeably alters or shifts the overall mean mass axis of the rotor itself. Distributed eccentricities creates some offset or skew between the rotor's actual mass axis due to eccentricity, and the geometric center axis defined by the centers of the rotor couplings as they attach to other rotors. Eccentricity would come from any more widely distributed asymmetric mass, for example, like a rotor bow, or skewed generator retaining rings.

[slide: Induced Eccentricity from Off-Square Couplings] Or, eccentricity can arise in conjunction with adjacent rotors in a coupled rotor train, where eccentricity is introduced by the effect of a non-concentric coupling rim or boltholes or center fit, or the effect of a non-perpendicular coupling face. As we can see in some examples in the picture, this induces eccentricity across single or multiple rotors, depending on factors like rotor and coupling stiffness.

[slide: Bowed/Eccentric Rotor: Mass Axis not Coincident to Geometric Axis] So let's take a look briefly at rotordynamic behavior. First, every object there is, rotors included, if free and unconstrained and rotated with applied torque, will naturally rotate about its mean center-of-mass axis, simply by the conservation of angular momentum. When this natural tendency is prevented, that is, when rotors are held and constrained by their couplings, and when the geometric line of the coupling centers **does not line** up with the **actual** center-of-mass line along the rotor body, due to eccentricity, then when the rotor is accelerated, this asymmetry will produce **forces,** and these forces will be expressed as vibration, either as rotor displacement amplitudes or as bearing seismic vibration. These forces represent the energy required to prevent and hold the rotor in a forced, non-centroidal rotation. That is, the vibration represents the forces preventing an eccentric rotor from reaching its natural state of spinning about its mean mass axis.

[slide: Resolving Eccentricity] Therefore, as our goal, we can remove those forces, and the related vibration, simply by assuring that the mass axis of the rotor, including whatever eccentricity is present, is brought coincident to the rotor's geometric axis, which it's constrained to run about when installed in the field, while also not bending or distorting the rotor in the process.

The good thing about distributed mass eccentricity, is that it is fully identifiable and resolvable in the service shop prior to operating the rotor. This means it can be found and fixed before installation and startup.

[slide: Service Shop Procedure: Runout Evaluation] When a rotor is in the service shop, mass eccentricity can be identified through TIR measurements, or total indicator runout. In very many outages, those assumptions that a rotor is fully concentric, and that the coupling faces are perpendicular, are **never thoroughly verified or properly measured or evaluated in the shop.** Now, that's not to say that shops don't take runout measurements – they do. But very often runouts are not taken properly or sufficiently for useful evaluation, either to save time or shop cost, or because their importance is overlooked, and this is because their ultimate rotordynamic effects are overlooked. And in some cases, shops **do** have properly thorough internal engineering procedures to follow, but the shops often cut corners because of outage schedule time and commercial pressure, and simply **do not record all the required data**. As a result, the runout condition is in large part **assumed or measured insufficiently, especially regarding coupling faces,** not recognizing the huge significance later on of glossing over this step.

It is extremely important that the TIR data is properly collected and then properly evaluated. We can narrow it down to five essential conditions that must be met in the shop regarding TIR measurements. Many post outage vibration problems occur simply because these five conditions were never met.

[slide: Service Shop Procedure: Runout Evaluation - requirement 1] The first condition is to collect TIR dial indicator readings at a minimum of every 45 degrees, and preferably every 30 degrees radially. This means ideally taking 12 readings at each plane of measurement. Anything less and the subsequent mathematical evaluation can't be properly performed. In addition, TIR readings should be collected at every plane of diametral change along the length of the rotor. It's also important that the rotor is pre-conditioned by slow-roll for a sufficient time prior to measurement, usually at least 8 hours, to remove any bows that may have developed while sitting in transit.

[slide: Service Shop Procedure: Runout Evaluation - requirement 2] The second condition is that this set of data points must be mathematically evaluated for 1 per rev and 2 per rev eccentricity relative to a **common journal** reference, including amplitude and phase angle. Simply noting the measured high point or simply averaging the runout values, and especially without phase angles, is entirely insufficient for evaluation of the eccentricity condition of the rotor. 1x evaluated eccentricity provides the offset between the geometric center and the mass centers at each measured plane, and 2x indicates the ovality or out-of-roundness of a given plane.

[slide: Service Shop Procedure: Runout Evaluation - requirement 3] The third condition, which is in many instances the most crucially important, while also by our experience is the most commonly neglected, is to measure and evaluate runout on all coupling faces, rims, and fits. The reason why having properly square coupling faces is **absolutely essential, is because perpendicular coupling faces are critical to achieving proper field alignment.** Because field alignment is done based on coupling measurements, if couplings are left off-square, it is almost guaranteed that the bearing and rotor alignment will not be correct, which will lead to eccentricity being present in the **overall** rotor train. This induced eccentricity can create a condition **of load dependent** vibration, or some other unusual responses, which often lead to an incorrect diagnosis of the real root cause of the problem, which is really originating from bad couplings.

[slide: Service Shop Procedure: Runout Evaluation - requirement 4] The fourth condition is that all TIR data must be collected on a single setup on the lathe in the shop, and no coupling can be held or constrained in a chuck on the lathe. It is essential to measure all TIR data with a single common reference, and this common reference must be the line of the journals that the rotor will spin on when installed and in operation. This is truly the only way to that the TIR data can meaningfully identify all eccentricity that might be present.

[slide: Service Shop Procedure: Runout Evaluation - requirement 5] And lastly, the fifth condition is to collect TIR data on the journals themselves, using at least three planes per journal, to evaluate all journals for concentricity, taper and ovality, surface roughness, and any diametral deviation outside of tolerances, to within 0.5 mil.

[slide: Service Shop Procedure: Runout Evaluation - tolerances] Following the mathematical evaluation of the TIR data for 1x and 2x eccentricity, if any **coupling** or **journal** is found out of tolerance, they **must** be corrected by machining. There is really no right way around this, and no shortcuts, without causing problems later in the field. This means bringing coupling faces perpendicular to within **1 mil**, and coupling rims and fits to within 0.5 mils, and the journals must be brought concentric and flat to within 0.5 mils. Ultimately, this is in agreement with ISO standards based on ISO 1940-1, or can be done following any other OEM's internal standards for tolerances on **new machines**.

Completing these steps will guarantee that proper and successful rotor and bearing **alignment** can be performed in the field using standard 16-point coupling gap and rim measurements.

[slide = Service Shop Procedure: Rotor Balancing] Now let’s take a look at the second problem area in outage shop work, which is rotor balancing on a high speed or low speed machine.

If faults ***are*** measured and discovered in the TIR steps, the second very problematic **assumption** is the idea that whatever eccentricities or bows are found on the rotor can be left alone and considered as typical equivalent unbalance, and that they can be resolved later in the balancing bunker or in the field as needed through standard balancing methods, thinking that even if we exceed the runout limits, we can still balance it anyway, not recognizing the limits of what balancing can accomplish when dealing with distributed eccentricities.

If rotor **body** evaluated 1x eccentricity is found to be outside of a threshold of about 2 mils, then the rotor requires special treatment to account for the eccentricities. First, though, we have to distinguish if rotor eccentricity is on the main rotor body between the journals, or if it is outboard of the journals. Any eccentricity beyond about 1 mil that is outboard of the journals can only be resolved by machining, and cannot be resolved by balancing.

But on the plus side, nearly any amount of rotor **body** eccentricity, that is, eccentricity in between the two journals, **can** be successfully balanced and resolved in a balancing bunker, as long as certain requirements are met and a specific unique balancing method is applied.

[slide = Balancing Significant Rotor Body Eccentricity - goal 1] To see why a special balancing method is necessary, let’s step back again to look back at rotordynamic behavior. Any eccentric rotor if it is placed in a balancing facility in free-free mode, that is, fully uncoupled, or using moment free drive couplings, will naturally shift to rotate about its mean mass axis above the first critical speed.

This means that any eccentric rotor balanced by standard methods at speed, above the first critical, will be inadvertently balanced about its mean mass axis created by the eccentricity or bow. In a balancing bunker, the rotor can appear well balanced looking at vibration amplitudes, even by using standard balancing methods. But when this eccentric rotor is reinstalled back in the field and coupled to other rotors, it will be held and constrained by its couplings to rotate about its geometric axis, which is **not** the axis that it was balanced about in the bunker at high speed. This means that when this rotor is installed and brought to speed in the field, it's all but guaranteed to have high vibration problems.

[slide = Balancing Significant Rotor Body Eccentricity - goal 2] The goal in balancing distributed eccentricity or a rotor bow is to **first** fully restore radial mass symmetry to the rotor about its geometric axis at lower speeds, up to the first critical, in what we could consider rigid mode balancing, to make sure that all rotation remains about the geometric axis at all higher speeds. This **must** be completed first, before balancing any higher critical speed responses.

Because so much of this modified process, and very often all of it, is performed at lower speeds, using this balancing method can also save a lot of time and runs and cost at the balancing facility, which helps the bottom line and the schedule, and at the same time gives a better result in the field.

[slide = Balancing Significant Rotor Body Eccentricity - goal 3] To guarantee good operation in the field of an eccentric, flexible rotor running above the first critical, which is essentially every turbine or generator rotor, it is crucial to not introduce any bending or distortion into the rotor during balancing, which happens when using too concentrated of weight placement. We don't want to un-bow or straighten the rotor with the balancing weights. Rather, balancing weights need to be distributed axially across a sufficient number of balancing planes, in essence mirroring the eccentricity distribution.

The key requirement of the rotor itself for balancing when any significant eccentricity is present, that is, any body eccentricity more than about 2 mils, is that the rotor must have at least **three** balancing planes, two endplanes and a midplane. If a rotor has more than about 2 mils of body eccentricity, 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, or it can be practically guaranteed that the rotor **will** cause problems in the field, even if the vibration amplitudes in a balancing facility, if using only two planes, might look successful. And if a third center balancing plane cannot be added due to material conditions or 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 and throwing the centers or by thermal straightening, to bring the rotor body eccentricity under the 2 mil threshold.

[slide = Balancing Significant Rotor Body Eccentricity - 2N+1 Balancing Method] While eccentric turbine rotors typically need 3 balancing planes, to generalize the method for any significantly eccentric, flexible rotors, like generator rotors, the number of planes required works out to the formula 2N+1, where N is the highest mode or critical speed that the rotor passes on its way to running speed. Using 2N+1 balancing planes is beneficial for balancing ***any*** rotor, but it's absolutely essential when balancing any flexible rotor with more than about 2 mils of evaluated 1x body eccentricity. Because this balancing procedure is done mostly at lower speeds, we also call it the Quasi-High Speed Balancing Method.

The formula 2N+1 comes about from finite element analysis, and represents the minimum number of axial elements or divisions required to divide a rotor into to define all the mode shapes of a given rotor, based on the highest mode that the rotor will see up to operating speed. The idea is that each rotor element, or section between balancing planes, will then behave sufficiently rigid as a rigid beam through the full speed range, and as such, each element can be balanced as a rigid beam.

There's a well known idea in balancing that any purely rigid rotor can be fully balanced in any 2 balancing planes. So by taking a flexible rotor and dividing it using 2N+1 divisions, each resulting division or element can be assured to behave as a fully rigid beam, and as a result, each element or division can be balanced in 2 planes. This also means that any **first critical speed correction weights** must be placed simultaneously in **three** balancing planes. The inner planes shared by two neighboring elements are essentially used twice.

[slide = Balancing Significant Rotor Body Eccentricity - 2N+1 Balancing Method - diagram] Here is a graphic showing the divided rigid elements of the rotor. As a result of using this axial distribution of balancing weights, any rotor will then maintain its inherent shape, even if it's bowed, without any distortion during operation. We can call this running dynamically straight. This prevents any internal cyclic bending, which can be a problem with generator rotors especially, and prevents high cyclic forces being transmitted into the bearings, which often happens in the field when standard balancing methods are applied to flexible and eccentric rotors in the balancing facility. The rotor also remains balanced about and spinning about its geometric axis at all speeds, and behaves as if it were concentric.

[slide = Balancing Significant Rotor Body Eccentricity - 2N+1 Balancing Method - Example results] It's too involved to get into any details of the balancing procedure itself, but we can see the result. In this example applying this balancing method with 2N+1 planes, we modeled a generator rotor with a distributed sloping eccentricity. We can see the resulting weight placement ends up mirroring the eccentricity distribution, with the minimum axial weight distribution to ensure that no bending or distortion occurs. Because the eccentricity was axially biased, only 4 planes ultimately required weights in this example. We can see that the rotor maintains it shape at speed, compared to the bending distortion that occurs if a standard static and couple balancing approach is used where the first critical is balanced in only one plane, which would also lead to excessive bearing forces and poor performance later in the field, even if in the balancing facility, the specific measuring probe locations at the journals might show low amplitudes.

[slide = Key Takeaways in Balancing Eccentric Rotors] So, in review, we can look at the most important takeaways from balancing when dealing with eccentric flexible turbine or generator rotors. First, any rotor with about 2 mils or more of evaluated 1x body eccentricity must have its first critical speed response balanced using 3 balancing planes. Ideally, because this is so important to a successful restart with low vibrations, this requirement of using 3 planes for the first critical should if at all possible be incorporated in the overall service contract with the shop. For more flexible generator rotors, the total balancing process should use 2N+1 planes for sufficient axial weight distribution. The final balance weight distribution will approximately mirror the distributed eccentricity on the rotor. Overall, it is essential that the **rigid mode forces are resolved first**, which is done at a speed only up to or slightly beyond the first critical speed, before even considering balancing at higher critical speeds.

The most important thing is that this method restores symmetry to the rotor about its geometric axis connecting the coupling centers, which is the line that the rotor is constrained to run about in the field when coupled to adjacent rotors. This balancing method shifts the mass axis of the rotor into this same line, and the result is that the rotor behaves "dynamically straight" as if it is a fully concentric rotor, without any bending or distortion at any speed. Again, all of this has very little to do with resonances, support stiffness and damping, but is primarily looking at and correcting the **rigid mode** behavior of the rotor, which is the true solution to the majority of problems. And we have found by experience that this balancing approach solves or prevents the great majority of vibration problems after a planned outage.

[slide = Summary 1] So, we can once again summarize the main points. Vibrations on assembled rotor trains are mostly caused by misalignment during installation as a result of unknowingly using off-square or non-concentric couplings for alignment reference, and from distributed mass eccentricities or rotor bows that were not properly compensated in a balancing facility.

During an outage, the main focus is so often on finding and correcting damage like cracks and seal rubs and such, and problems like runout and bows become secondary, and are even skipped if the schedule becomes short on time. But, it's these runouts and eccentricities that are the **fundamental cause of vibrations**, and these eccentricities are really the root cause of all those other damages, which are only symptoms. So that's where the focus needs to be, on resolving eccentricities.

[slide = Summary 2] It's unbelievable how time and money could be saved by recognizing the effects and importance of runouts and eccentricities and coupling faults, and incorporating these verifications into the outage process, and correcting them while rotors are in the shop.

As a solution to guarantee a smooth restart, the answer is to leave **no unverified assumptions** as to the rotor condition in terms of runout. This requires measuring and mathematically evaluating full total indicator runout readings, including journals and coupling rims and faces.

It's critical to match service tolerances to existing OEM tolerances **for new machines**, and to machine any coupling or journal to its OEM factory tolerance **prior** to balancing. Any excessive rotor body eccentricity **can** be balanced as long as the proper axial weight distribution is used, using 2N+1 balancing planes, and especially, solving the first critical speed response in 3 balancing planes in all cases.

These steps should ideally be incorporated into every rotor outage plan, and ideally should be discussed and amended into the shop service agreement prior to the start of an outage. By our experience, when these steps are followed, every post outage restart **can** be successful on the first run, with no need for further work or field balancing. ....

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