

Development of a New Balancing Approach for Significantly Eccentric or Bowed Rotors

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Abstract. This paper presents a new balancing method pertaining especially to significantly eccentric, flexible rotors. When dealing with these rotors in service facilities, standard balancing methods often require excessive runs, and vibration problems are frequently experienced upon rotor reinstallation in the field. The method presented here is based on a novel view of the unique rotordynamic behavior in these cases. The key is to fully compensate rigid modes first to bring the effective principal mass axis coincidental to the rotor's designed geometric axis, preventing a switch of precession axes through the critical speed region. This must incorporate proper axial distribution of weights using $2N+1$ balancing planes (where N is the highest operating mode). The end result is a "dynamically straight" rotor that remains balanced without distortion or bending at any speed. This method requires fewer runs, and the rotor is guaranteed not to need field balancing upon properly aligned installation.

Keywords: balancing, eccentricity, inertia, critical speed, dynamically straight

1 Introduction

The goal of this paper is to shed some light on the occasional "difficult" practical cases dealing with large turbine-generators, where a rotor appears to not adhere to the behavior otherwise predicted or expected based on standard rotordynamics theory. These situations are especially prevalent in the rotor service industry when working with highly eccentric or bowed, flexible rotors, well outside the eccentricity tolerances specified in ISO 1940. While further in the past, these cases tended to be "odd exceptions", in recent years as more machines are aging, they are tending more toward being the rule in the service industry. The lead author, having worked with and balanced hundreds of "difficult" large rotors over 35+ years, has developed a balancing method that reliably works in all situations, often requiring many fewer runs and requiring fewer weights than standard approaches.

Current balancing methods and procedures have worked well for many years, and this paper is not intended to challenge the current balancing approaches, but rather to add a

new balancing method that can be generally applicable, but with a key advantage when applied to balancing highly eccentric flexible rotors in balancing facilities. This paper also briefly touches on certain physical phenomena inferred and observed in highly eccentric rotors, particularly through the critical speed region, to support why this balancing approach is effective, and in some cases necessary. A more detailed and novel look at the rotordynamic behavior behind this balancing approach is found in other associated papers by the author.

2 **Balancing Approach**

The traditional 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. 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.¹

These standard approaches of balancing at speed are proven and have worked very well in most cases for 50+ years, especially in rotors within ISO 1940 eccentricity tolerances with localized unbalances. However, balancing results for flexible rotors with large distributed eccentricity are often marginal at best, especially viewed in light of rotor behavior in the field following post-outage reinstallation. There are many cases where a highly eccentric or bowed rotor that was relatively “well-balanced”, when independent and uncoupled in a balancing facility fails to run smoothly upon installation (coupled to other rotors through rigid couplings). This arises from ignoring the fact that an eccentric rotor will alter its position (shaft centerline) and orientation in its bearings when passing the first critical speed region. This “re-alignment” is determined by the size and orientation of its distributed eccentricity, and leads to the rotor tending toward a natural “state of least action” where it tends to spin/precess around its principal mass axis (if it is not physically constrained from doing so). Balancing above the first critical speed is then inadvertently done around the mass axis, rather than the rotor’s intended geometric axis (the line connecting the radial journal centers). This in particular creates problems 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 once reinstalled and at speed or load with torque applied.

This 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

¹ Rotors operating under these conditions will eventually self-correct these moments, but as a result may develop coupling face deformations or even develop cracks.

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 (about the rotor’s principal mass axis) to minimize response amplitudes.

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”. It is important to remember that 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 rotor is generally not possible. The latest technology in measurement sensors, though contributing much to diagnostics in rotordynamics, has also led to an over-reliance on “relative shaft vibration readings” alone as a measure of balancing success, ignoring the corresponding forces produced in the bearings.² The value of shaft displacement sensors is still great in rotordynamic diagnostics and in the balancing process to gauge rotor response and discern weight placements, but should not be used alone to define a rotor’s balance condition.

2.1 Quasi-High Speed Balancing Method (or 2N+1 method)

A key distinction in the author’s balancing 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.³ The goal of balancing should then be to statically shift this mass axis⁴ to be again coincident to the geometric axis, around which the rotor will rotate when coupled. This must incorporate a sufficient and proper axial distribution of correction weights to effectively mirror the 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, which leads to a balance solution as an extension of the rigid-mode or rigid-

² In operating plants, “relative shaft vibration readings” are often used as a limit for “safe” operation, often unnecessarily forcing a turbine-generator unit to shut down over safety concerns for what is very often a benign cause with minimal true transmitted forces. Worse yet is the situation when balancers reduce the displacement amplitudes below “alarm level,” but only by unintentionally creating a disproportional, damaging increase in bearing reaction forces.

³ This mean mass axis becomes the principal axis of rotation/precession above the critical speed region, when inertial forces become dominant in controlling rotor behavior.

⁴ 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.

body balancing concept that any 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.

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. The axial division of the rotor is derived from the minimum number of elements required to define a rotor in a FE model, such that no eigenvalues and corresponding eigenvectors will be missed within the speed range of interest. 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.

This method is somewhat similar in concept to Fredric Ehrich’s patented “Pseudo-High Speed Balancing Method” [4], which uses a low-speed balancing machine with force sensors alone (calculating corrections using influence coefficients), and is intended for a manufacturing environment. In contrast, the author’s method is intended primarily for high-speed balancing machines, and for the practical purpose of obtaining sufficiently clear measurement signals, is based on shaft displacement readings, but while simultaneously observing bearing seismic signals.

As an example, a typical turbine rotor that operates above its first critical speed will pass through a response condition with 3 nodal points, in this case with each endplane a nodal constraint and the midplane a bending peak. Each half of the rotor (a modal element or “rigid element”) behaves like a rigid beam for the purpose of balancing, with a shared boundary condition at the midplane. Therefore, each half is balanced in two balancing planes as a rigid body, the midplane being used twice, resulting in an initial balance weight distribution (of ounce-inch effect) of 25% on each end, and 50% in the midplane. This distribution is then axially adjusted as needed (and the angle adjusted as well if required), with the adjustment determined by placing trial shots in pairs of equal oz-in effect (endplane and midplane), and then using influence coefficients as customary, but distributing the resulting calculated adjustment into pairs as well. In general, once the first critical peak response is sufficiently reduced, additional weights are no longer needed, but rather only axial redistribution is required. As a consequence, fewer balancing weights are required in the balancing process.

As a side note, if a large turbine rotor only has two balancing planes (endplanes) and has a significant distributed eccentricity or bow, it cannot be properly balanced (in that it will not run smoothly once reinstalled, even if it may appear balanced in a balancing facility). In this situation, a center balancing plane *must* be machined in, if thermal conditions and material properties allow, or machining must be done to otherwise minimize the eccentricity⁵ (throwing the journals, etc.) [5,6]. If a very large turbine has 4 balancing planes (such as two flow turbines), the 25%-50%-25% distribution still applies, with the 50% center weights distributed across both midplanes, as a total of $2(N+1)$ planes. If a turbine rotor exhibits a rocking motion (the rigid mode analog of the second critical re-

⁵ Significant eccentricity existing outside the journals *must* be corrected by machining only.

sponse), this need not necessarily be considered as a true second mode (implying 5 planes are needed), but can be addressed by an axial redistribution between the midplane and two endplanes, determined by placing trial shots in pairs.

For a more flexible rotor such as a generator rotor that operates above its second critical, each half of the rotor at the second critical speed behaves in a manner self-similar to the full rotor at the first critical (though the halves are generally out of phase to each other), with a resulting five nodal points and requiring five balancing planes, to balance four “rigid elements”, again with the shared nodes between elements effectively used twice. In these more flexible rotors with higher modes, the first critical is still fully resolved first using three planes (the midplane plus either quarter-planes or endplanes). For higher modes, it is then necessary to place trial weights in pairs as before, with a key being to always place the pair of weights on each end of a single “rigid element”. Depending on the eccentricity distribution or rotor response, the trial pair can be placed on a quarter-plane and midplane, or on a quarter-plane and endplane. The angular placement should remain consistent between the trial weights. The resulting calculated adjustment must be applied similarly in pairs.

Similarly, a very flexible generator rotor operating above the third critical (or first flexural mode) can be viewed in the same manner, incorporating 7 nodal points and balancing planes, creating three self-similar “humps” (in the idealized mode shape), each of which incorporates two rigid elements to be balanced as a rigid beam in two planes. In this case as well, the trial shots to calculate the axial re-distribution should be placed in pairs on each end of a single “rigid element”.

The result of this process is effectively an axial redistribution of the initial first critical correction to account for the axial asymmetry of the eccentricity distribution. In practice, the rotor should then pass through its first critical with only a minimal response, if any, and its amplitude should remain flat above the first critical, with no trending upslope in amplitude with added speed. As a measure of success, if a rotor reaches a speed 50% above the first critical and its amplitude trend remains flat at both bearings, the rotor is effectively balanced for all higher speeds. This is the source of naming the method “Quasi-High Speed Balancing”, since nearly the entire process can be completed below the speed of 50% above the first critical.

Beyond this, the rotor should still be tested to running speed, and any additional residual unbalance responses at higher speeds (from a chipped turbine blade, for instance) must be balanced using only pure modal distributions (a V-shot for out-of-phase ends, or an S-shot for in-phase ends), such that additional corrections have a zero-sum force and moment distribution so as not to alter the previously adjusted position of the mass axis.

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 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 guaran-

teed to run smoothly once properly aligned and installed in the field, and has been proven in practice numerous times, even with the most “historically difficult” rotors.

2.2 Example of the QHSB-Method For a Flexible Generator Rotor

The following example provides a description and plots for the balancing process of an actual flexible, eccentric generator rotor (50Hz), using the Quasi-High Speed Balancing Method (in $2N+1$ planes). Total indicator runout measurements showed the rotor body had an eccentricity of around 2 to 3 mils, in a somewhat randomly undefined distribution shape, though primarily confined to the same side of the rotor.

The rotor was determined to operate above its third critical speed, and therefore, there were seven balancing planes used. The critical speeds and mode shapes were determined ahead of time through a finite element model. A summary diagram of the balancing planes and balancing weight distributions used is shown in Figure 1, and can be referenced during the explanation of the procedure.

The five central, pre-selected balancing planes are those determined from finite element modal analysis to be at the modal peaks and nodes of the first and second mode resonant responses. These planes, however, are used to resolve the rigid modes of the rotor resulting from the mass eccentricity. By successfully resolving the rigid modes, the resonant responses will be practically eliminated as well.

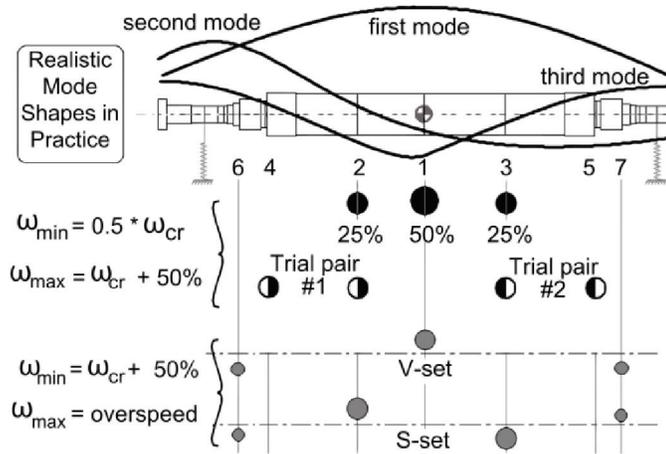


Fig. 1. Identification of balancing correction planes and possible weight distributions

The rotor is first rolled in its original unbalanced state to the maximum speed achievable, without exceeding preset vibration amplitude limits. An image of the Bode plot can be seen in Figure 2. In this example, the rotor’s run-up was halted at 540rpm upon ob-

serving the rapid increase in amplitude while approaching the first critical speed. To roll to higher speeds, a balance shot must be placed first.

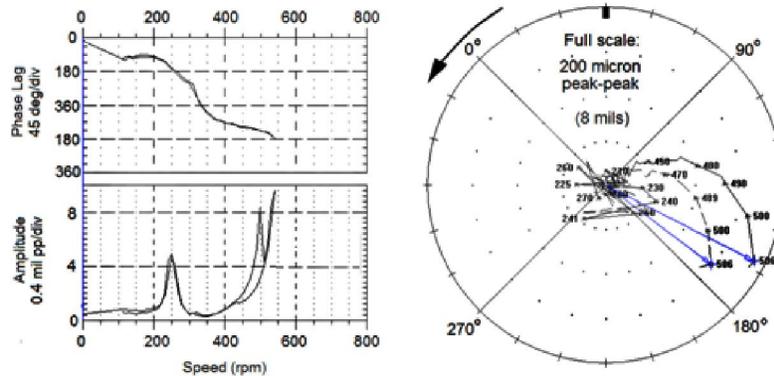


Fig. 2. Bode and Polar plots of the first run showing high amplitudes approaching the first critical speed, used to determine the direction of the static unbalance vector

The rotor response is also shown using polar plots. The signals from sensors located at the same reference angle from the two journals are plotted on same polar plot in Figure 2. The full scale of this first polar plot is 8 mils. As can be seen on the polar plot, the phase shift had reached approximately 70 degrees.

Based on the angles of the static response vectors at the highest speed reached, the rigid lateral mode is corrected first with weights distributed in the three central planes (planes 1, 2 and 3) as shown earlier on Figure 1, with the weight placement angle chosen based on the direction of the response vector seen in the polar plot. This first set of weights, however, should attempt to reduce only ~ 75% of the first critical peak, since some available correction weight should remain for subsequent axial redistribution. With the subsequent run, it is crucial to observe that there is no notable angular shift of the polar plot orientation along with a reduction in peak amplitude, which would indicate that the angle of the weights is incorrect. The balance weights are placed in a proportional distribution where approximately half is placed at the midplane, and the other half is divided equally at each quarter-plane. The first critical correction weights are estimated using the 10% rule suggested by Bentley Nevada, which states that the effective centrifugal force generated from a trial weight at operating speed should not exceed 10 percent of the total rotor weight.

The response of this run is used to generate the first set of influence coefficients. The Bode plot is also monitored for an up-sloping amplitude trend above the first critical speed, which indicates that the eccentricity is not axially symmetric and is generating a rocking mode, and axial weight redistribution is required (Figure 3).

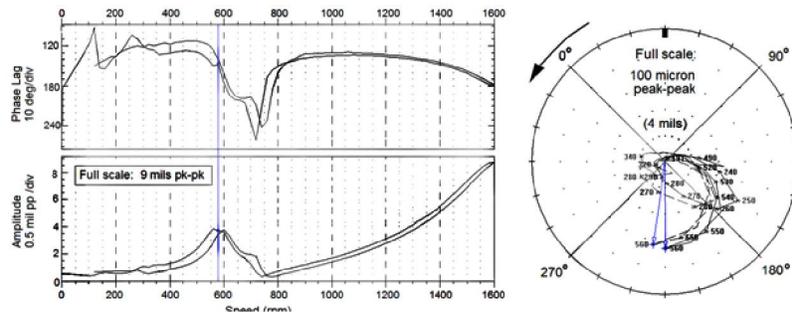


Fig. 3. Bode and Polar plot following the initial first critical correction in 3 planes.

The first critical solution should be adjusted by placing additional trial weights in pairs in planes 1 and 2 (at 12.5% of the “100%” eventual first critical correction in each plane). The angle of the weights can be further adjusted within ± 5 degrees from the initial phase angle if needed. If improvement is seen, the weights should be left in, and if no improvement, they should be removed. The same trial weight should then be placed in planes 1 and 3, and run again. From this, influence coefficients will provide the best solution, and the calculated adjustment must be distributed and applied as a pair. Ideally, the first critical should now be 100% corrected with a response no larger than the initial slow-roll runout from eccentricity. As the rotor is run above the first critical again, if there is still an up-sloping amplitude trend, then it is necessary to do a similar procedure of paired trial weights in the outboard planes.

In this case, two more sets of trial weights of equal oz-in effect (and of the same weight as the “12.5%” pairs used in the first critical response correction) were placed, first in planes 2 and 4, and then in the next run in planes 3 and 5. This distribution should be similarly refined using influence coefficients until it can be seen that there is no longer a rising upslope in the amplitude displayed on a Bode plot at speeds above the first system critical speed. With proper axial redistribution of the weights, the “rigid modes” of the rotor should be resolved. A Bode and polar plot are shown in Figure 4, where a flat amplitude trend can be seen with increased speed.

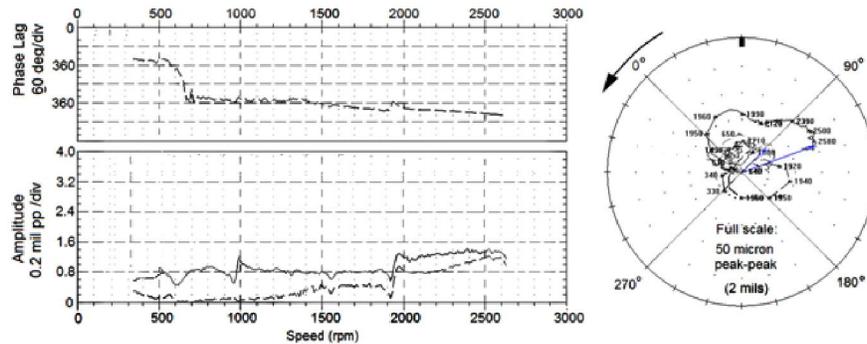


Fig. 4. Bode and polar plots following the axial redistribution of weights based on influence coefficients obtained through the previous two trial runs.

This process ultimately results in five balancing planes being needed to fully solve the rigid mode responses from the eccentricity, which makes the rotor “dynamically straight”. The rotor’s mass axis and shaft axis should remain coincidental through the entire speed range, and no change in the axis of rotor precession will occur above the first critical speed.

If the rigid modes were solved correctly, then a Bode plot should show a flat amplitude trend up to operating speed (with a peak-to-peak amplitude equal to twice the rotor’s eccentricity), with an exception possibly being a brief jump in amplitude at the second and third critical speeds (as can be seen later in Figure 4). If the amplitude has remained flat until a speed of 50% above the first critical, then it can be assured that it will remain flat for all higher speeds as well. The only need is to run once to operating speed (plus an overspeed test if required) to verify whether there is a remaining response from any residual eccentricity or any resonant excitation response of the second or third mode.

As speed is increased on this next run, if any increased amplitude is observed at the rotor’s second system critical speed (a bending of the rotor or its coupling overhangs), any correction should now be done only using only modal weight distributions in an S-configuration using planes 4 and 6, and 5 and 7 (or if necessary, planes 2 and 6, and 3 and 7), in such a way that the sum of their forces and moments equals zero. Since the rigid modes are already resolved, all subsequent weights must not alter the now-adjusted mass axis of the rotor.

Continuing above the second critical to full operating speed, there may be a third critical (first flexural mode) response or an observed change in shape of the amplitude curve seen in a Bode plot, at one or on both ends of rotor. If the amplitudes become predominantly out of phase (as seen on a polar plot), the previous S-set modal weights should be modified. If the dominant amplitudes at running speed are in-phase, a “V” modal weight set should be installed using planes 1, 6 and 7 simultaneously. If the responses are a com-

combination of both, the motion with larger amplitude should be balanced first with the appropriate modal set, with refinement then done using the other modal set.

If after installing the appropriate modal weight distributions there is no further response seen, the rotor balance is complete. A Bode plot of the balanced rotor can be seen in Figure 5. As can be seen, the total vibration at the bearing locations at the end of the balancing process is less than 0.5 mils through the entire speed range.

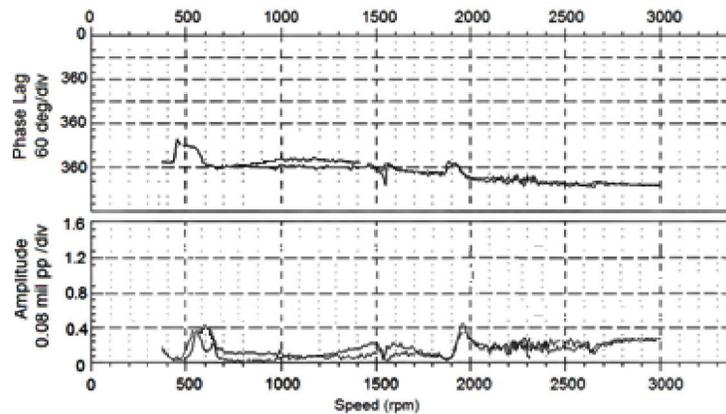


Fig. 5. Final Bode plot of the fully balanced generator rotor

Lastly, in the rare occasion that a long overhang itself has an independent critical speed response, balancing weights in the couplings themselves can be used as part of a modal weight configuration. This last portion, though, is not considered to be a part of the overall $2N+1$ balancing procedure. Any eccentricity of the overhangs must be reduced by machining before commencing any rotor balancing.

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. As long as the Bode plot amplitude curve remains flat for all speeds, the rotor will be running "dynamically straight". When a rotor is eccentric, the flat Bode plot amplitude line is not necessarily at zero, but may be flat at 1 or 2 mils which represents the inherent eccentricity in the rotor, and it cannot necessarily be further reduced. (Or rather, it could be "reduced" only by actually unbalancing and repositioning the rotor to an operating alignment where a nodal point is shifted to the displacement probe's measuring location, which would increase the force of the rotor on the bearings or introduce internal moments in the rotor.)

It is obvious that all possible combinations of unbalance from eccentricities are impossible to predict. Nevertheless, by solving a rotor's rigid modes first, using the Quasi-High Speed Balancing Method in $2N+1$ planes based on the rotor's highest mode shape within its operating speed range, the rotor will be made "dynamically straight". With that, the

effects of the inertia forces from any mass eccentricity can be avoided, making the entire balancing process simpler and more efficient.

3 Conclusion

The Quasi-High Speed Balancing Method is applicable on any low or high speed balancing machine, reduces the number of runs and uses less power during the balancing process. In addition, a rotor balanced by this method eliminates the need for subsequent field balancing following proper alignment and installation.

With many additional runs in a balancing facility and much trial and error, a highly eccentric or bowed flexible rotor may ultimately be brought to acceptable amplitude limits by standard balancing methods, but can be left with numerous unresolved internal moments. In addition, the labor hours spent and the electricity costs for the drive motor used makes the choice of procedures an economic issue as well.

Ultimately, it is the change in the rotor's bearing position and nature of precession from the effects of inertial forces that especially creates problems for standard balancing methods when applied to eccentric, flexible rotors. The goal of balancing an eccentric rotor then is really to prevent the change in precession axis from noticeably occurring, which prevents the change in rotor orientation in the bearings. This is achieved by bringing the mass axis coincident to the geometric axis from the start, and allowing the rotor to accelerate to speed without deflection or distortion. In balancing eccentric rotors, it is crucial to create this condition to avoid unknowingly installing and constraining a rotor to run in a state other than the one in which it was balanced and aligned in the bearings.

By following the Quasi-High Speed Balancing Method using $2N+1$ balancing planes, any eccentric rotor can be brought to a smooth running condition when installed in a properly aligned rotor train.

The next topic to address is why this approach is effective for eccentric, flexible rotors. The answer requires interpreting the unique rotordynamic behavior in such cases. To answer this question in full detail would require far more explanation than is possible to include here, though a reasonable depiction of the rotor behavior is attempted in the associated "Part 2" of this paper series, "Behavior of Eccentric Rotors Through the Critical Speed Range."

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