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PRACTICAL BALANCING OF FLEXIBLE ROTORS FOR POWER GENERATION

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ABSTRACT

Balancing technology is still relatively new. Thirty years ago it was primarily still part of the skilled trade and was often obscured. Today there is enough reference literature printed during the last 20 years alone on general balancing and balancing of flexible rotors, that could fill a room, (Ref: N. Rieger). The majority of papers and other references deal with theoretical derivation of equations based on Jeffcott rotor model. With the growth of rotor sizes specifically of electric generators in power plants, so grew the need to develop not only a theory, but also the way to practically balance these rotors. The economy of manufacturing required pushing the rotors to more and more slender; lower and lower stiffness ($\propto \frac{EI_{xx}}{L^3}$) designs, in relation to its mass moment of inertia (I_m), these rotors were more difficult to balance.

The first to encounter the problem of balancing these rotors were the OEMs. On two different shores of the Atlantic Ocean, two basic balancing theories known as balancing in "N", or in "N+2" balancing planes and operating rotor modes were developed. Later, with the development of the microcomputer the influence coefficient method had gained popularity among the power plant community and despite good experiences from both sides the controversy over which one produces better results was left open. In this paper a review of the “N” and “N+2” methods including notes on influence coefficients (IC) is conducted from a practical standpoint. The conclusion by the Authors is that there is no "better" or "worse" balancing method, only the more or less economical in a given situation, and neither gives a unified method to satisfy every rotor. General guidance is also provided over which method to use for best results in balancing large turbo-generator sets.

INTRODUCTION

Flexible Rotor Balancing Methods, 1928 – 1984 (Ref: Rieger, verbatim)

"The need for flexible rotor balancing procedures emerged in the post World War 2 period, due largely to the rapid increase in the size of the rotating equipment used in the power generation, chemical processing, and aircraft turbine industries. The balancing technology needed for these developments in equipment size was developed somewhat ahead of the equipment need. A modern method for flexible rotor balancing can be traced back to the 1928 patent by Linn F. C. [3], who proposed a modal method for the balancing of single span flexible rotors. The influence coefficient method can be traced to Thearle E. L. [1] who proposed a procedure for balancing three-bearing turbine-generator sets. DenHartog J. P. [4] described this procedure for rigid rotors in his well-known book.

Modal methods were well established for generator rotor balancing when Groebel L. P. [5] wrote a descriptive note on their use in 1952. The use of orthogonality relations in theory of this procedure was described by Meldahl A. [6] in 1954. Major contributions to the theory of modal balancing of flexible rotors were made by Bishop R.E.D.,Gladwell G.M.L., and Parkinson A.G. in England, in a classical series of papers [7] [8] [9] [10] [11] and others between 1959 and 1968. Practical application of this method to a range (turbines, generators, pumps, synchronous condensers) of heavy rotating equipment was concurrently also demonstrated in England by Moore [12] [13] [14] between 1964 and 1972.

During this period, other modal techniques for balancing of rigid, quasi-rigid and flexible rotors were developed in Europe by Federn K. and Kellenberger W. (Ref) [15] [16] [17] and by Miwa S. [18] in Japan. This procedure became known as the comprehensive Modal Balancing method because of the range of rotor types to which it was successfully applied. Considerable controversy surrounded the parallel evolution of the N and N+2 modal procedures, arising primarily from questions surrounding the need for correction (or elimination) of the rigid rotor modes. The resolution of this question appears to lie in the relevance of the rigid modes in the balancing process, and it has been discussed extensively with examples by Rieger N. F. and Shou S. [19], and Bishop R.E.G. and Kellenberger W. [2] [17]. " Such modes must be corrected where the rotor is rigid or quasi-rigid (Class 1 or Class 2 rotors in the ISO specification), but they are typically unimportant for flexible (Class 3) rotors in rigid supports some authors claim."

This statement is valid only theoretically, and with an assumption that residual unbalances are negligible, and may be represented as concentrated unbalance only. Further, such claims ignore the effect of rotors overhang, i.e., generator rotor with large coupling unbalance running solo in the balancing facility.

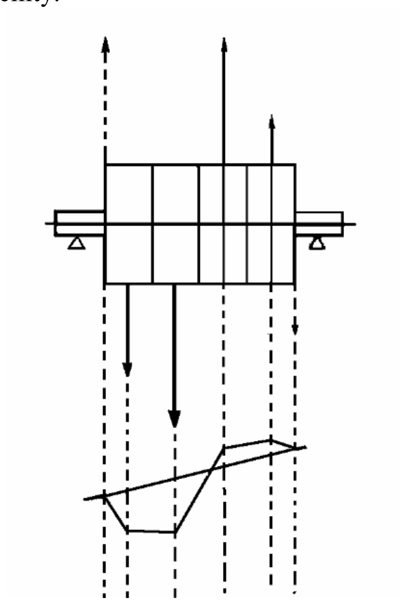

Fig. 1: Internal moments acting on a Rigid Shaft

Actually, the limitations of the ability to localize unbalances are explained with the example of a rigid rotor (Fig. 1). On this rotor, various unbalances act in a multitude of planes. It is not possible with any type of measurement technique to analyze these unbalances plane for plane and to equalize them directly at their origin. Although the closest we can get to measure these unbalances is indirectly through measurements of rotor runouts and evaluation of eccentricity. On a rotor-bearing-pedestal system only the sum of unbalance effects can be recognized, e.g. by measurement of the bearing forces during rotation, and this sum effect can be compensated by balancing corrections in two arbitrarily selectable balancing planes. The remainders of such balancing are the so-called "internal moments" which are shown as a moment area. The existence of these internal moments in a rigid rotor is of no significance to the running behavior. The rigid rotor is not subject to any deformations caused by centrifugal force and mass displacements which could lead to a speed-related change of the balancing condition.

It is important to keep in mind that unbalances are not directly measurable; only their effects can be recognized and that all the theories of balancing are based on two major assumptions: 1) Rotor response to unbalance is linear (like in a Jeffcott rotor). Meaning that the response of the rotor to an applied excitation (unbalance) is solely based on its linear elastic response; 2) Unbalances are small enough so they satisfiy the above statement. This implies that unbalances produce deformation within the linear elastic range of the rotor.

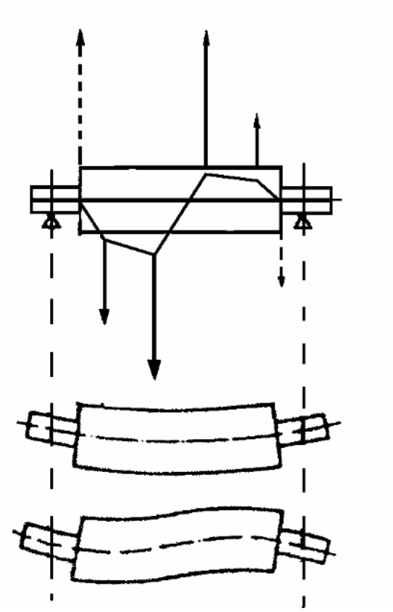

Fig. 2: Internal moment acting on a Flexible Shaft

Of course, the term "rigid" is a simplistic abstraction. The transition from a rigid to an elastic rotor, as well as a transition between mass and rotational centerlines, is indistinct (fluent). Only when the lowest critical speed is far above the highest operating speed will the shaft behave like a rigid body.

In contrast to a rigid rotor, the internal moments on an elastic rotor cause speed-dependent deformations, which very well affect the balancing conditions and the running behavior decisively (Fig 2). The objective of the balancing process of such rotors is therefore the reduction of the internal moments, which, as a rule, can be performed only in more than two balancing planes.

Balancing in a limited number of planes will only result in an approximation, not the ideal condition, and providing the linearity assumptions are met, then the quality of this approximation decisively depends on the original condition: the smaller the original unbalances, the better the result. The problem, and thus the dispute arises between proponents and opponents of various balancing methods because of need for different balancing machines to accommodate one or the other method (need to measure forces at balance machine pedestals).

"Goodman [20] gave the underlying theory of the influence Coefficient method in a form suitable for balancing both rigid and flexible rotors in 1962, and he wrote the first computer program for influence coefficient balancing of flexible rotors at that time. Goodman's report also contains the theory of both the so-called Exact Point Speed method and the Least Squares influence Coefficient method. Lund J.W. and Tonnesen J. [21], Rieger N.F. [22], Tessazik et. al. [23], Badgely R.H. and Rieger N.F. [24] and others contributed to the continuing development of this procedure from 1964 on. The Influence Coefficient method emerged as the first computerized balancing procedure. The computer program performs efficiently organized sequence of operations using trial weight response data, to identify the needed correction weights and phase angles, at the specified balance locations."

The biggest problem encountered when balancing using the influence ciefficient method (IC) is when the rotor exhibits nonlinear spring behavior. In such a case only experience and diagnostic ability counts.

The modal method and the influence coefficient method have been combined into a single procedure known as the Unified Balancing Method by Darlow, Smalley and Parkinson [25] [26]. Further optimization of this method has been performed by Zorzi [27], using modal procedures to optimize the influence coefficient corrections process and by Kanki et.al. [29] using LMI optimization method and modal trial weights.

Despite of the development of various methods of flexible rotor balancing, most problems with difficult to balance rotors, whether in the factory environment, balancing rotors solo in high speed balancing facility (bunker), or in an assembled rotor train in the power plant, are due to the rotors' "controlled initial unbalance". ANSI S2.42-1982 says: "Controlled Initial Unbalance" is initial unbalance which has been minimized by individual balancing of components and/or careful attention to design, manufacturing and assembly of the rotor(s).

The best method of balancing tested by the Authors in balance facility, which also gives the result of greatly minimizing, if not eliminating the need for subsequent balancing in the field after rotors are coupled, is based on the experimental method done by Zorzi (27)

Balancing several hundred rotors of all sizes and various flexibilities in a bunker, the Authors used various balancing methods. It was discovered that when a rotor has acceptable eccentricities based on ISO 1940, it becomes totally irrelevant which balancing method is being used. In most cases simple "N" balancing planes with final trimming at 3600 RPM is sufficient.

When a rotor is semi-rigid and bowed, the best method is to balance in three planes, one plane as close as possible to the maximum bow location and the other two planes at the outer ends. This should apply also to low speed balancing of the rotor, unless it is absolutely rigid, or if eccentricities are minimized and controlled prior to balancing, e.g., balancing of multi-disc gas turbine rotors and compressors.

When dealing with a flexible rotor it was learned that in order to save time in the bunker and the number of balancing runs, it is very beneficial for balancer to have a visual presentation of the rotor evaluated eccentricity distribution. The number of balancing planes on the rotor in the bunker should be equal to the number of rotor modes including the modes of shaft overhang within the range of speed to overspeed. This results in N+2 planes. It is of utmost importance to note that the new mass centerline be brought to within a minimum eccentricity from the rotational centerline of the coupled rotor.

In the case of a flexible rotor with large distributed eccentricity, it was learned by experiment that N+2 balancing planes must be used in resolving the rotor's lowest mode.

Balancing planes selected this way then are used to distribute correction weights for the rotor's lowest mode. This would prevent deformation of the flexible rotor which generally occurs when concentrated weights in reduced number of balancing planes are used during balancing in a balance facility. In the remaining steps, when balancing higher modes, whether using "N" or "N+2" method, become totally irrelevant.

THEORY AND PRACTICE

Almost all theoretical references to "Balancing" are based on the Jeffcott rotor model, which is based solely on "elastic unbalance response". These theories can be confirmed experimentally on a Jeffcott model rotor in a test rig. The problem in practice is that rotors are not Jeffcott type and very often their "controlled initial unbalance" (ANSI) is not satisfactory for various reasons.

In regard to theoretical results it is more useful today to study a FE model of an actual rotor rather than relying on 80 years of Jeffcott theory, which is excluding rotor rigid mode, mainly due to the different natures and prediction capacity of the models.

In the factory environment, when manufacturing new rotors, several factors may influence "initial unbalance", machining as well as assembly tolerances and machining errors. These produce "mass unbalance", i.e., create an eccentricity between rotational axis and mass centerline or geometric axis that exceed the limits (ISO 1940) and at which the rigid rotor mode excited by "mass unbalances" become relevant in correcting "elastic unbalances response", using modal balancing methods, as shown previously. (Resulting from the use of the N balancing planes method)

In the service shop environment, where rotors come for refurbishment after many years in service, there may be additional sources of "mass unbalance", bows, machining errors in field (typically from cutting or polishing journals), or from deformation of couplings. All of these sources of "mass unbalance" are geometrical and can be detected and measured by careful set up and measurements of rotor TIR in a lathe prior to balancing.

The TIR readings should then be mathematically evaluated for 1xrev. eccentricity using appropriate software for comparison against the limits of eccentricity stated in ISO 1940.

The evaluation itself is more art than science, because of the different effects of eccentricities at different axial locations and magnitude distribution along the rotor. How critical the eccentricities are is different for each rotor and is usually subjective and part of the service contract requirements. The eccentricity evaluation and correction before balancing is necessary to avoid gross deficiencies, encountered during the balancing process and to settle unattainable requirements (ANSI S2.42-1982 p1) beforehand.

Based on the evaluation of eccentricities and the expected "residual unbalance response", the rotor is either corrected by machining, or it is balanced by a modified "Unified Method of Modal Balancing" or a combination of IC and N+2 method depending upon rotor linearity of response.

GENERAL BALANCING

The act of balancing flexible rotors means to bring the rotor shaft and bearing pedestal vibrations to acceptable limits for operation. In contrast to the general belief is, balancing does not mean finding the hidden unbalance of the rotor and correcting it, but rather to find a suitable set of balance planes and balance masses that together bring the vibration of the shaft and bearing pedestal to acceptable limits.

Therefore the Balancing Process when balancing flexible rotors should consist of:

1. Establishing acceptable limits of controlled initial unbalance (Rotor eccentricity magnitude distribution and individual phase relations).
2. FE Modeling of rotor and determination of mode peaks location and mode shapes for the selection of balance planes.
3. Controlling shop processes in which limits under #1 could be exceeded, or new initial unbalances created, e.g. shrink fitting of components or assembly of slots and wedges on generator.

Further balancing must be classified by:

1. Balancing new rotors - in factory
2. Balancing service rotors – in service shop, balancing in balancing facility (BOS)
3. In-place balancing – assembled rotor train after: a) Initial installation b) Installation of rotors returned from service c) Operational mishap

BALANCING IN THE FACTORY

"Balancing technology is still relatively new. Thirty years ago, it was primarily still part of the skilled trade and was often obscured. In the mean time, extensive technical literature is available also on this subject. Today, hardly any manufacturer of turbine-generators can claim that he practices his very own methods. The differences in the method are small and due to the specific requirements of the individual products". (ref R. D'ham, ver.)

As is commonly known, the goal is to obtain good running characteristics of the machine at operating speed as well as over the entire speed range, particularly at critical speeds, i.e. minimization of the following variables:

- dynamic loads at bearings

- shaft deflection to prevent clearance interferences
- vibration transmission to the outside
- dynamic shaft stress

The dissimilarity of rotors and their requirements with regard to the balancing technology makes a classification into 3 categories appropriate (another classification can be found in document ISO 5406-1980):

- “Rigid" rotors, which typically include only the HP and IP turbine shafts in larger steam turbines.
- Elastic rotors with a maximum of two modes, which include most of the LP turbine shafts, large four-pole turbine-generator shafts, as well as two-pole generator shafts of smaller and medium size, and also the shafts of rotating rectifier exciter machines.
- Highly elastic rotors with more than 2 modes, i.e. primarily two pole turbine-generator shafts of higher rating whose extreme degree of slenderness imposes higher demands of the balancing method.

This classification also identifies the increasing degree of difficulty of balancing.

For turbine construction it is possible to high-tune individual shafts by means of design. This is in principle not possible for turbine-generators. Thus, generator construction has always been the "pace setter" in balancing technology.

Based on previous discussion of modal balancing methods (N and N+2 methods), it is by no means unimportant which balancing route is used to reach the objective. This is not only valid with regard to manufacturing cost, but also to the transparency of the method. If the balancing at one time or another does not meet the expectations, a clarification must be brought about whether an incorrect method of balancing for the rotor type and controlled initial unbalance is the problem or dimensional deficiency of the rotor itself is. The clarification is only possible if the followed method of balancing is sufficiently transparent and the dynamic behavior of the rotor understood trough FE modeling.

When using the IC method for balancing of flexible rotors, many measurement runs are required with test weights (trial runs). The question presents itself whether the influence coefficients, which had been determined earlier on rotors of the same type can be used. Then, if true, the balancing effort could be reduced considerably. Although extensive data banks were set up by manufacturers through out the years, which make the retrieval of empirical values possible; practice has shown that test weights cannot be eliminated for multi-plane balancing since the method converges only with exact influence coefficients.

BALANCING OF SERVICE ROTORS IN THE SHOP

In theory, the balancing process in a Service Shop should not be any different than that in the Factory environment. In practice, it was seen that the majority of service rotors have dimensional deficiencies, which very often are either ignored or their importance, and their affect on dynamics of coupled rotor is not recognized. The entire industry around field balancing had grown because of this.

Speaking in terms of economics ignoring or not recognizing and not addressing rotor deficiencies in detail in the service shop, is a method of transferring the cost of correcting resulting vibration to the users and operators of that equipment in the field.

For the best result, the service shop and the owner of a rotor going through repair or refurbishment needs to recognize that the rotor must be brought dimensionally to the specifications of the new rotor, at least at critical points, i.e. coupling rim, face, rabbit fit and bolt hole centerline, in relation to both rotor journals. On the other hand the effect of other dimensional deficiencies like excessive body runout, or other component eccentricities or bow; in most, but not all, cases, could be corrected in the bunker through the balancing process, rather than by machining preceding the balancing. The success of correcting body dimensional deficiencies by balancing depends highly on rotor rigidity. The more rigid the rotor, the less chance for good running behavior when re-installed and coupled in the field it gets.

TRUTHS AND FALLACIES OF BALANCING

Balancing was always considered partly as black magic. Over the years, the balancing process was reduced to activities by mechanics and young engineers alike using a simplified process in the form of a cookbook without a full understanding of rotor dynamics. From this some rules were created which must be scrutinized.

Fallacy #1

Bearing requirements

Many authors claim that for best balancing results, bearings in a balance facility must have same characteristics (stiffness and damping) as bearings in the field. The only problem which may occur in the balance facility is that highest operating mode, like 3rd critical speed mode, happens to be above operating speed, e.g., >3600 in the bunker. This can happen often on slender generator rotors which have a critical speed very close to or exactly at operating speed in the field.

In a balance facility the 3rd critical may not be reached before 4200 – 4300 RPM. If that fact is ignored, then the rotor can be balanced well in the balance facility up to 3600, but not run well when installed back in the field, for the simple reason that the 3rd critical mode was left unbalanced!. If this rotor is balanced through overspeed no problem should arise in the field

In another case, the rotor can be designed to operate on three bearings, but it can be balanced in a balance facility only in two bearings. These two cases can be explained because the rotor eigenvectors (mode shape) are mostly dictated by the rotor's geometry only, and rotor's eigenvalues (critical speeds) are the result of material and bearing properties. (Fig.3)

When balancing a rotor it is important to distribute the weight correction according to its excited mode shape regardless its critical speed. Therefore it is obvious that not only a type of bearing, but even the number of bearings is not mandatory to achieving good rotor balance conditions in a balance facility under one set of system stiffnesses, and be balanced also for another set of boundaries conditions in the field. This is best shown in the paper on balancing an axially non-symmetric rotor operating on three bearings, but balanced in the balance facility on two bearings.[28]

The conclusion from previous case is, that although the critical speed frequency may vary on rotor run in balance facility or assembled in the field, that alone should not be a reason to use same bearings in a balance facility as those in the field. Practice had shown that the shift in frequency of critical speed in balance facility rarely exceeds 20%. Besides the rotor eigenvectors are always the same despite a change in eigenvalues. Eigenvectors are mostly the result of rotor geometry, while eigenvalues are primarily affected by material properties and bearing pedestal damping stiffness characteristics. (Fig. 3)

Fallacy #2

Slow Roll Runout

It is often said that for exact balancing a slow roll runout must be subtracted from the vibration proximity probe reading.

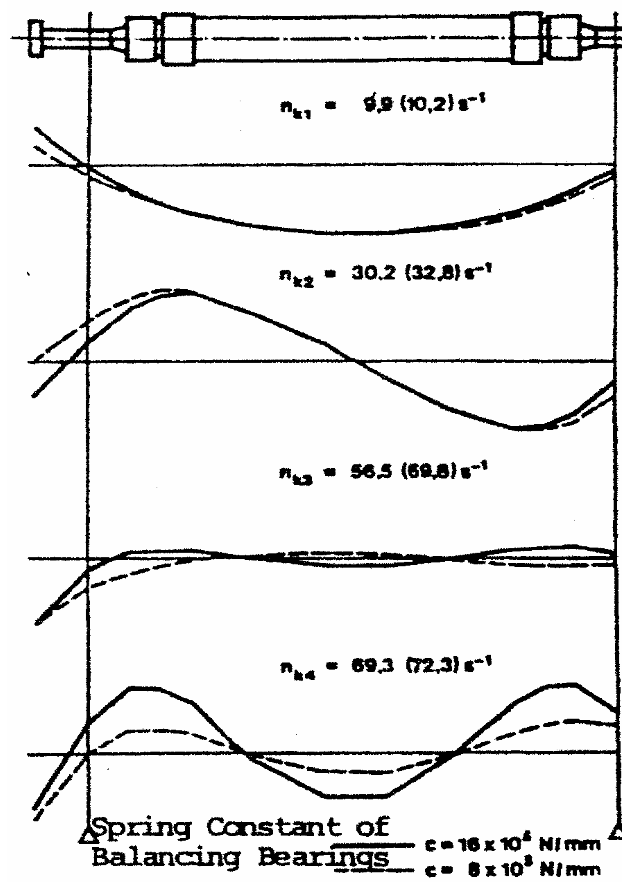

Fig. 3: Influence of bearing stiffness in the mode shape of the rotor

Slow roll runout correction is a good idea when the area under the probe is damaged or in any way adds to a high vector reading at the measuring location. But Slow Roll Runout Subtraction may be counterproductive or sometimes even dangerous when other dimensional deficiencies exist on the rotor (bow, coupling eccentricity, etc.). Slow roll subtraction could lead an inexperienced balancer to ignore bearing-pedestal vibration up to the point of damaging it while reducing shaft readings. So, when Slow Roll Runout is applied, it must be done with caution and understanding. A complete rotor runout measurement with subsequent eccentricities evaluation is a far better way of dealing with the issue.

On the other hand, properly used Slow Roll Runout readings can be used as a tool in diagnosing eccentricities and misalignment of the coupled machine in the field.

Fallacy #3

Coupling overhangs have no influence on rotor balancing.

Experience has shown that balancing in the balance facility is done correctly only if large free shaft projections (overhangs) do not become effective. The result of modal balancing to a large extent is independent of the balancing facility bearing stiffnesses (fallacy #2). Even shafts of Category b) which were balanced in soft bearings hardly require additional balancing corrections upon transition to operational bearing conditions. On the other hand however, if a so called “L-mode” becomes effective with significant deflection of the overhang, these simple relationships no longer apply.

For rotor category c), the following facts are briefly mentioned:

- During balancing of the individual shaft in two bearings (rotor in isolation) mode shapes may occur which later on do not appear during operation with the coupled shaft system under different boundary conditions. Nevertheless, these mode shapes must be taken into account to obtain a balancing result, which can be interpreted (Fig. 3).
- The number of excited mode shapes of a highly elastic shaft in the balancing facility is generally larger than in the coupled shaft system at the power plant. However, experience has shown that a good result can be achieved by systematic consideration of all mode shapes, including the overhang driven “L-mode”, in the balancing facility.
- Modal balancing leads to difficulties with rotors of category c). In the range above the second mode shape, weight sets, which solely affect a certain mode shape, are no longer easily found. This complicates the separation of the relationships between unbalance and vibrations and leads inevitably to an alternative method like the Influence coefficient method.
- The presence of an “L-mode” should be checked before any balancing attempt is made. Generally overhangs are very unstable and non-linear; they may also present higher harmonic vibration such as 2X, and create a Morton effect. The overhang response to unbalance is extremely sharp due to its low damping and usually remains at high vibration amplitudes after crossing its critical speed, even well above it. The threshold of stability is difficult to predict making the task no easier. See Hidalgo J. Dhingra A. [30].

Due to the above characteristics it is common practice to couple a carefully design stub shaft and bearings to the overhang in order to restrict the overhang displacement during balancing in the balance facility to prevent potential damage to the shaft.

Fallacy #4

Field balancing is absolutely necessary

“The concept of set balancing (coupled, multi-rotor train) is evidently beyond the scope of practical balancing technology at present" (Ref. Rieger). The Author further implies that assembled rotors must be treated and balanced separately from balancing individual rotors.

The fact is that when individual rotors are evaluated and balanced, and controlled initial unbalances are less than those specified in ISO 1940, for a particular group, then there will be no need to balance the assembled rotors again in the field. The only reason that field balancing may be needed is if a "new initial unbalance" is inadvertently created because of assembly errors exceeding specified tolerances or from external influences from the operating processes, or if a system resonance exists with very low damping. Even if a resonance of the system exists on a assembled machine, it should not be excited if the "initial controlled unbalance" had been reduced in previous processes and individual rotors balanced. The existence of resonant condition may make the machine "sensitive" to unbalance, but eventual high unbalance response cannot be attributed to lack of balance in balance facility, and rebalancing actions are surely no correction for deficient balancing precision in the shop. In such case a proper way to approach the problem is to determine the source of the resonance.

FIELD BALANCING

The most common reasons for field balancing are the eccentricity between geometric axis of two rotors and the non-perpendicularity of coupling faces to respective rotor geometric axis. In both cases the errors cause introduction of new moments, which did not exist on either rotor when balanced individually.

Field balancing becomes necessary more often than it should be, usually because of "savings" in the shop processes and allowing a rotor with larger than "allowed" eccentricities to be processed. If, in fact, field balance becomes necessary after all and the rotors were balanced at high speed to "acceptable" levels, it is because the eccentricities at couplings (rim, bolts centerline, rabbet fit) were ignored and left with eccentricities higher than permissible by ISO 1940 (G1, G2.5). Even more frequent problems occur with shrunk-on couplings, or other shrunk-on components of the rotor due to incorrect processing in the shop. In general, running conditions at the plant site are usually worse than in the balancing pit. Despite that, balancing in the field becomes necessary only when a new condition arises during the assembly process of rotors with deviations left from the shop.

Therefore it is essential that diagnostics be implemented that will determine whether the rotor is in fact out of balance, or whether some other mechanical malfunction is in progress, prior its field balancing.

BALANCING UNDER OPERATIONAL CONDITIONS

For subsequent balancing correction on turbine-generators, which are assembled for operation, only a limited number planes are available (Fig. 4). This fact emphasizes the restrictions already mentioned above since the root cause of the vibration problem cannot be really addressed. The balancer is concerned with the interference effects and tries to counteract these effects with the least possible effort. In the course of balancing, he meets contradictions as expected because of "unresolved internal moments" and he cannot improve the running condition at all measurement points. While the machine is in operation the balancer must also take into account adverse side effects (rotor distortion due to thermal expansion, alignment changes, cost of each run, etc). Such contradictions can often be softened or even eliminated by adding suitable balancing planes (using coupling bolts as balance planes). Therefore, the task of the balancer deals with optimization of running conditions rather than with true balance. The desired compromise must not be restricted to purely technical aspects but must also take into account economical considerations. The high cost of an operational shutdown and the goal of high availability of large turbine-generators must be kept in mind.

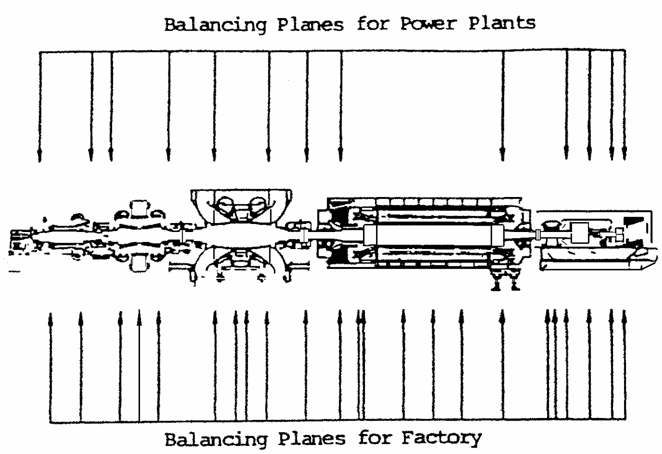

Fig. 4: Available balance planes in the factory and in the field

The balancer is obligated to adapt himself to the operational activities and to utilize operationally caused shutdown periods for his actions. He is directed to only set carefully sized test weights in order not to endanger the unrestricted availability of the turbine-generator. Test unbalances induced by undersized test weights have the disadvantage that their effect does not clearly stand out from the "scattering" of the measurement values. Because of the scatter data and the lack of optimum available balance planes, then the balancer is usually force to use IC.

The balancer must recognize the limitations of the his chosen balance method as early as possible so that unnecessary costs do not arise. A novice may perhaps not see the light until after the 10^{th} trial that his balancing effort is a lost cause. On the other hand, with a certain systematic approach and on the basis of experience and knowledge of the rotordynamic characteristics of the machine such a situation can be recognized earlier and the necessary conclusions can be drawn. The insistent question about the cause of such disturbances is not easily answered most of the time. As already mentioned, the cause cannot be located with sufficient accuracy from the analysis of the vibration distribution across the shaft system of a turbine-generator.

It is highly recommended that vibration readings, (<u>both</u> amplitude and phase angle) be taken periodically on turbine-generators for safety reasons. This practice will increase the accuracy of a damage analysis in case of a disturbance and will help on deciding which specific action needs to be taken. The obvious question arises again and again why turbine-generators must be rebalanced at all at the power plant. Rebalancing actions are surely no correction for deficient precision in the shop. Balancing during operation is used to compensate for the influences of operational disturbances.

Some of aforementioned disturbances are briefly explained, without the claim of being complete:

- Excessive sensitivity of a system to residual unbalances, e.g. due to vicinity of resonance. It is a known fact that each turbine-generator is analyzed for its coupled critical speeds during the design stage. The assumption can be made that the coupled critical speeds of the shaft system are kept far enough from operating speed by carefully executed design measures. However, there are a number of additional local effects that can lead to isolated local resonances.. Such system resonances, which are a function of the boundary conditions, cannot always be addressed during the design stage.
- The axial vibration of a bearing pedestal crossbeam, due to rocking motion of the shaft, is mentioned as an example. (Fig.5)

- Thermal Influences. While thermal influences hardly become effective in turbine shafts despite higher temperatures – influences which may change the bowing of the shaft and thus the balancing condition – such influences may not always be excluded in generator shafts. It must be kept in mind that slight localize temperature differences on the rotor, especially for long, slender shafts, can cause considerable bowing. The shaft of 600 MW generators in whose winding a power of approximately 3 MW must be converted to heat is given as an example. Ideally the heat must be dissipated symmetrically, but if there is a temperature difference of 1K between the poles over the entire rotor body length, then the rotor will bow at the center approximately 120 µm. This would correspond to approximately 30 times the customary balancing tolerance.

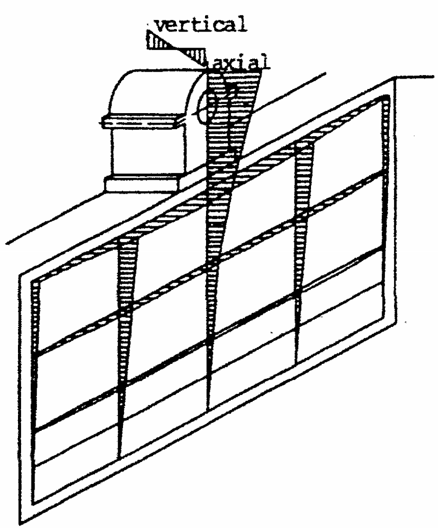

Fig. 5: Axial vibration of a cross beam

- In addition, there are rotor bending influences as a result of the differing thermal expansion of copper conductors, slot wedging and shaft. These components are joined to each other via high centrifugally related frictional forces. Each wedge is subjected to tensile and compression stress as a result of the static sag of the shaft, which is almost 3 mm for large flexible rotors. Minor differences in the sliding behavior of the components produce bending that result in excitation of vibrations.
- In all cases, the generator shaft imposes extremely high requirements on the design in view of its degree of slenderness as well as the mentioned non-homogeneity of its construction.
- As a further reason for rebalancing some overall influences are mentioned which may develop in the course of operating time and during maintenance inspections: blade erosion and the related rework, rework of the journal areas, changes in the runout of the shaft, consequences from blasting operation during cleaning which may cause no-uniform material removal on the circumference, etc.

For reasons of cost and time savings, rebalancing is carried out on site during maintenance inspections. Experience shows that in general a good or at least acceptable running condition of the entire turbine-generator can be achieved. However, the required effort differs in widely. In the normal case, success in restoring the running conditions with few balance corrections is achieved on the basis of empirical values gained on the same turbine-generator or one of the same type.

Experience has shown again and again that an improvement of the vibration condition of an individual shaft is not assured by any means via rebalancing in operating condition. This finding is not only theoretically of interest, but it has a very practical meaning for efforts carried out during machine maintenance inspections. The correction performed in the power plant apparently applies only to a specific constellation of rotors, their coupling and alignment conditions. If this constellation is changed within the scope of a maintenance inspection, the balancing condition of the shaft system may have to be corrected once more.

CONCLUSION

Different balancing methods have been discussed from the practical stand point of balancing turbo-generator sets. Their advantages and draw backs compared, not from a rigorous mathematical point of view but from the practicality of implementation. As to the likelihood of success for different types of rotors to achieve or exceed balancing industry standards no method was proven superior in all cases.

In the view of the Authors the words of Rieger still strongly hold true "No quantitative guidelines have been established to guide designers of high speed flexible rotors, and as yet there is no classification of what constitutes effective flexible balancing for specific machine types".

Because the number of variables and scenarios in which balancing is applied in power generation are practically infinite, the Unified Theory of Balancing, which will satisfy ALL rotors under ALL possible scenarios (rigid, flexible, small, large, and all grades, as well as for shaft and bearing vibration reference limits, service environment, etc) is close to impossible. The balancing method which comes closest to fulfilling all requirements, in the Authors opinion, is a method in which the rotor's lowest mode is corrected using a number of balancing planes equivalent to N+2 for each particular rotor, and vibrations are corrected based on both shaft and bearing readings. Otherwise, the guide to "best" balancing method will depend on:

1. Economic factors (process costs and procedures and cost allocation.

2. Available balancing machines, instrumentation and expertise.

3. "Good enough vibrations" agreement to acceptable vibration limits based on established standards and contracts between client and balancing service shop, using any of the available balancing methods as long as "controlled initial unbalance" is strictly managed.

Because there is no better or worse balancing method, the balance engineer should be aware of all of them and their suitability for each individual condition to achieve the best desired results.

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