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IDENTIFICATION OF FAULTS IN ROTATING SHAFTS AT RATED SPEED

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Abstract. *Malfunction identification in rotor systems by means of a model based approach in the frequency domain during long lasting speed transients (coast down procedures in large turbo-groups), where a huge amount of vibration data at different rotating speeds is usually collected, has proved to be very effective. This paper explores the possibility to adapt this method to the situation when the vibration data are available at one rotating speed only, which in real machines is generally the normal operating speed.*

Keywords*: rotor dynamics, diagnostics, identification.*

1. Introduction

The authors have developed a malfunction identification procedure for rotor systems based on the model of the system (shafts, bearings and supporting structure), on suitable models of the different malfunctions and on a least square approach in the frequency domain. This procedure analyses the vibration data usually collected by monitoring systems in correspondence of the bearings only, at different rotating speeds selected from a long lasting coast down transient, which is considered as a sequence of steady state operating conditions. The procedure has been applied to several turbo-groups of power stations, and proved to be effective and robust, provided that the machine exhibits linear behaviour and has time invariant characteristics. Generally turbo-groups composed of steam turbines and generators fulfil this condition; on the contrary turbo-groups composed of gas turbines and generators exhibit behaviours which are scarcely repetitive, which do not fulfil this condition.

A diagnostic system should be able to predict an impending fault also during normal operating conditions at rated speed. This is obviously a much more difficult task, because the available information at one speed only are much less. Some single speed identification procedures in the time domain (Platz *et al.*, 2000, Markert *et al.*, 2000) seem to be successful, but have never been used with real machine vibration data.

The aim of this research is therefore to check if and to which extent the model based approach in the frequency domain is able to identify the impending fault, by using the data of one rotational operating speed only, both in case of test rigs and of real machines.

The approach uses the finite beam element model of the shaft, bearings are represented as usual by stiffness and damping coefficients, the supporting structure is represented by mass, spring and damper systems, or , when available, by a modal model of the structure. In the identification procedure instabilities are not taken into consideration. Also single faults only are considered, the possibility of multiple coexisting faults is not accounted for.

All malfunctions are modelled by means of equivalent exciting forces. These are generally applied in one location only along the rotor, and are often composed by one rotating force only (e.g. unbalance) or by two equal and opposite bending moments applied to the nodes of one element (e.g. local bow due to a rub, transverse crack, coupling misalignment), or to the extremity nodes of a set of elements representing a huge part of the rotor (in case of thermal bow or distributed axial asymmetry). Therefore malfunction identification is reduced to just one equivalent external force (or couple of bending moments) identification: one or two complex unknowns to be identified, and the measured data are generally at least four complex vibration vectors (at least two bearings and two directions of measurement) at each rotating speed. The number of known quantities is higher than the number of unknowns, therefore the least square approach can be used also in case that only one rotating speed is taken in consideration (the normal operating speed), instead of a set of n different rotating speeds chosen during the long lasting speed transient.

The data which are used for the identification procedure at one rotating speed only, are much “poorer” with respect to the huge amount of data which are at disposal during the speed transients.

Especially some important frequency dependent features of the equivalent forces are completely lost: the information that the unbalance depends on the square of the speed, and that the local bow and similar malfunctions are represented by a constant amplitude force system, cannot be exploited in the identification procedure at one rotating speed only.

In order to explore the capability of the method in malfunction identification, despite the described shortcomings, several different conditions in test-rigs and in real machines are taken into consideration. An other important feature of an identification procedure is also its capacity to discriminate among different malfunctions which exhibit similar symptoms.

The investigation will show also which are the limits of the procedure in this sense.

2. Description of the method

Assuming a finite beam element model for the rotor, the effect of a fault can be simulated in the frequency domain, by applying to the rotor different sets of equivalent forces. The problem of the identification of the position and magnitude of the fault is then reduced to an external force identification procedure, described in (Bachschmid *et al.*, 2002). The final equations are recalled here below.

The difference, between the measured vibration of rotor system that has a fault and the reference case, represents the vibrational behavior due to the fault, which is called sometimes "additional vibrations". These vibrations are then used in the identification procedure. By applying the harmonic balance criteria in the frequency domain, the following equations are obtained for each harmonic component, in which the force vector $\mathbf{F}_{f_n}$, has to be identified:

$$\left[-(n\Omega)^2\mathbf{M}+i\,n\,\Omega\,\mathbf{C}+\mathbf{K}\right]\mathbf{X}_n=\mathbf{F}_{f_n} \tag{1}$$

Since linearity in the system is considered, the effect of m faults developing simultaneously can be considered by means of the superposition of the effects for each harmonic component:

$$\mathbf{F}_{f_n}=\sum_{i=1}^{m}\mathbf{F}_{f_n}^{(i)} \tag{2}$$

Moreover, the *k*-th fault acts on few d.o.f. of the system, therefore the vector $\mathbf{F}_{f_n}^{(k)}$ is not a full-element vector which is convenient to be represented by means of:

$$\mathbf{F}_{f_n}^{(k)}=\left\{\mathbf{F}_L^{(k)}\right\}\overline{A}^{(k)}(\Omega) \tag{3}$$

where $\{\mathbf{F}_L^{(k)}\}$ is the localisation vector which has all null-elements except for the d.o.f. to which the forcing system is applied, and $\overline{A}^{(k)}(\Omega)$ is a complex number representing the amplitude and the phase of the fault.

Then, introducing the admittance matrix of the system, eq. (1) becomes:

$$\left[\mathbf{E}\left(n\Omega\right)\right]\ \mathbf{X}_n=\mathbf{F}_{f_n}(\Omega) \tag{4}$$

Normally, only a developing fault is going to be identified and the equivalent force fault identification problem in eq. (4) is overdetermined since the number of the observation (the measured vibrations) are grater than the number of the parameter of the fault that have to be identified. Least square identification is used in order to evaluate the module, the phase and a residual of a particular fault type in this way.

The equivalent force system, is applied in each node of the rotor model. The effect on the measured d.o.f.s $[\hat{\mathbf{X}}_{\mathrm{B}_n}]$ due to unitary force systems applied in the first node on the model is calculated. This is done by inverting matrix $\left[\mathbf{E}(n\Omega)\right]$, obtaining the transfer matrix $\left[\mathbf{H}(n\Omega)\right]$.

$$\mathbf{X}_n=\left[\mathbf{E}\left(n\Omega\right)\right]^{-1}\mathbf{F}_{f_n}(\Omega)=\left[\mathbf{H}\left(n\Omega\right)\right]\mathbf{F}_{f_n}(\Omega) \tag{5}$$

Then, the vibrations of the d.o.f.s, which are measured, are separated from the all the d.o.f.s of the system, by considering only the rows of $\left[\mathbf{H}(n\Omega)\right]$ corresponding to the measured d.o.f.s..

It results:

$$\left[\hat{\mathbf{X}}_{\mathrm{B}n}\right]=\left[\mathbf{H}\left(n\Omega\right)\right]\Big|_{\substack{mesured\\ d.o.f.s}}\left[\mathbf{F}_L^{(1)}\ \ \vdots\ \ \mathbf{F}_L^{(m)}\right] \tag{6}$$

Now the array of the complex values $\overline{A}^{(i)}$ (i.e. the modules and phases) of the equivalent force systems applied in the first node that fits best the experimental data $\mathbf{X}_{\mathrm{B}m_n}$, have to be estimated. The fitting is done in least square sense, since the number of the unknown (the modules and the phases) is less than the equations. The problem is equivalent to:

$$\min\left\|\left[\hat{\mathbf{X}}_{\mathrm{B}_n}\right]\begin{Bmatrix}\overline{A}^{(1)}\\ \vdots\\ \overline{A}^{(m)}\end{Bmatrix}-\mathbf{X}_{\mathrm{B}m_n}\right\| \tag{7}$$

whose general solution is given by means of the pseudo-inverse calculation:

$$\mathbf{A}^{(1)}=\left(\left[\hat{\mathbf{X}}_{\mathrm{B}n}\right]^{\mathrm{T}}\left[\hat{\mathbf{X}}_{\mathrm{B}n}\right]\right)^{-1}\left[\hat{\mathbf{X}}_{\mathrm{B}n}\right]^{\mathrm{T}}\mathbf{X}_{\mathrm{B}mn} \tag{8}$$

The modules and the phases of the complex values in the m rows of $\mathbf{A}^{(1)}$ are the identified faults in the first rotor node. Finally the residual in the first rotor node is determined, first obtaining the calculated response due to the identified fault in the first node:

$$\mathbf{X}_{\mathrm{B}n}=\left[\hat{\mathbf{X}}_{\mathrm{B}n}\right]\mathbf{A}^{(1)} \tag{9}$$

and then normalizing it:

$$\delta_{r_n}^{(1)}=\left(\frac{\left[\mathbf{X}_{\mathrm{B}_n}-\mathbf{X}_{\mathrm{B}m_n}\right]^{*\mathrm{T}}\left[\mathbf{X}_{\mathrm{B}_n}-\mathbf{X}_{\mathrm{B}m_n}\right]}{\mathbf{X}_{\mathrm{B}m_n}^{*\mathrm{T}}\ \mathbf{X}_{\mathrm{B}m_n}}\right)^{1/2} \tag{10}$$

The procedure is then iterated for all the n_r nodes of the rotor. If a fault only is considered, a set in $\mathbb{R}$ of relative residuals given by eq. (10), ordered by the node number, is obtained:

$$\boldsymbol{\delta}_{r_n}=\left(\delta_{r_n}^{(1)},\ldots,\delta_{r_n}^{(n_r)}\right) \tag{11}$$

The s-th node location that corresponds to the minimum value of eq. (11) indicates the most probable location of the fault, whose estimation is given by the corresponding value of eq. (8).

3. Application to test rigs and industrial machines

Several different cases are examined, and the results are discussed trying to find reasons for successful and unsuccessful identifications.

3.1. Case 1 125 MW turbogas group affected by known unbalance.

Figure 1 shows the results of a quite successful identification procedure at one rotating speed only (2283 rpm), which is not rated speed, compared to the result of the identification procedure which uses the data collected at all the available rotating speeds (figure 2): the position is identified with good accuracy, the amount is more overestimated using one speed only. The value of the residual is low in both cases indicating good quality in the identification procedure. But if another value of the rotating speed is used, the identification can be also completely unsuccessful.

In order to have an overview of the identification procedure success at the different rotating speeds the identified quantities (position, amount and phase) are plotted as a function of the rotating speed.

Figure 3 shows the value of the residual, the node where the unbalance has been identified, its module and phase.

These values are compared to the true values (dotted lines). As can be seen only in the speed range 2050 – 2700 rpm the identification is possible. This is probably due to the irregular behaviour shown in figure 4 and figure 5, which represent additional vibrations obtained by subtracting from “faulty” vibrations the reference “healthy” vibrations. This irregular behaviour is apparently due to a scarcely repetitive behaviour of this kind of machine in different run down transients, and cannot be represented by linear and time invariant models.

Figure 6 in which the values of the residuals related to the unbalance is compared to those related to a local bow which have similar symptoms (both generating only 1xrev. vibration components), shows that when one speed only is considered, it is impossible by analysing the values of the residuals to distinguish between malfunctions which generate similar symptoms. In these cases only the likelihood of the identified positions for unbalance or local bow could help in distinguishing the impending fault: locations in which local bows are unlikely to develop or unbalances can not be applied might discard one or the other malfunction.

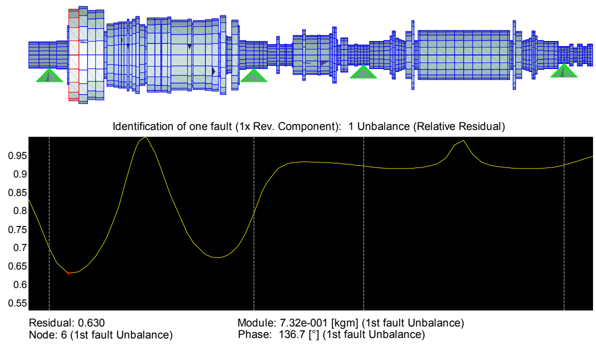

Figure 1. Case 1: gas turbine and generator, 125MW - Identification results using the data corresponding to one rotating speed only (actual fault: unbalance, node 6, module 0.373 kgm phase 157.5°).

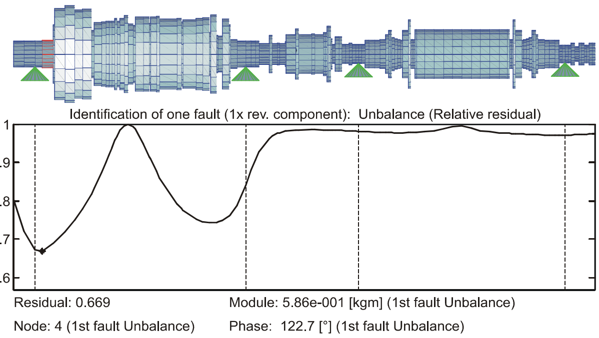

Figure 2. Case 1: gas turbine and generator, 125MW - Identification results using the data corresponding to all the available rotating speeds (actual fault: unbalance, node 6, module 0.373 kgm phase 157.5°).

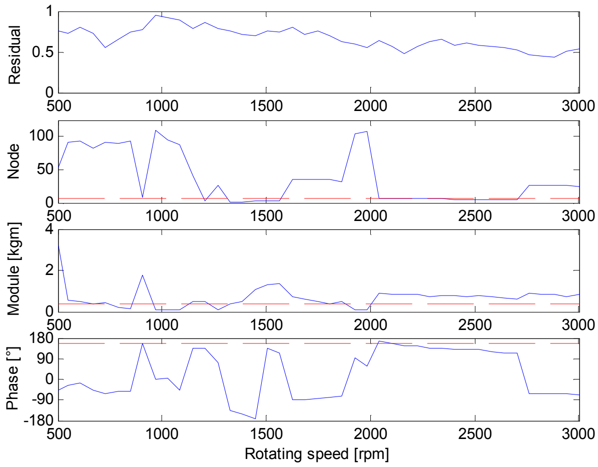

Figure 3. Case 1: residual, node, module and phase identified as a function of the rotating speed.

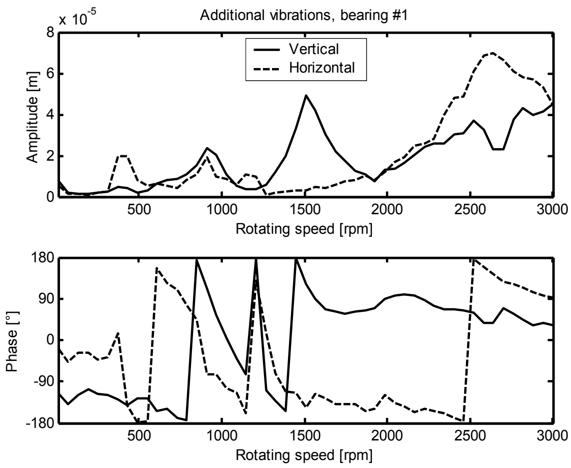

Figure 4. Case 1: additional vibrations due to the unbalance on node 6, module 0.373 kgm phase 157.5° measured in bearing #1.

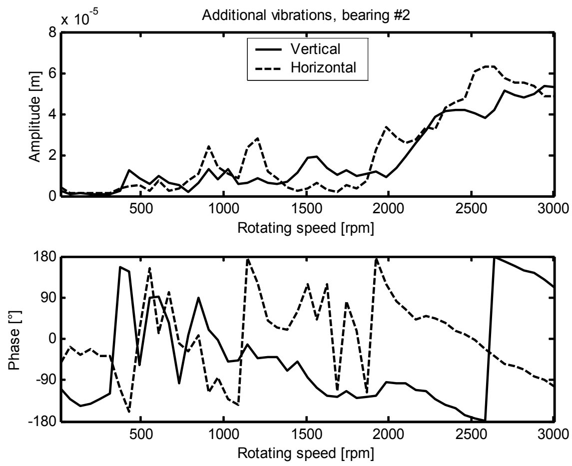

Figure 5. Case 1: additional vibrations due to the unbalance on node 6, module 0.373 kgm phase 157.5° measured in bearing #2.

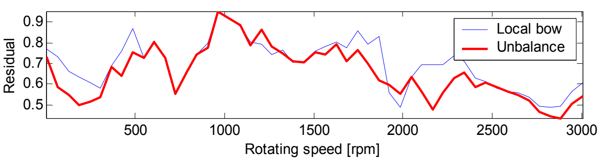

Figure 6. Case 1:comparison between the residuals of different fault types.

3.2. Case 2 Modiarot test-rig with unbalance

The test rig Modiarot developed at the Dept. of Mechanics of Politecnico di Milano is composed by 2 shafts on 4 oil-film bearings on a flexible foundation. The model of the shaft and the applied unbalance are shown in figure 7.

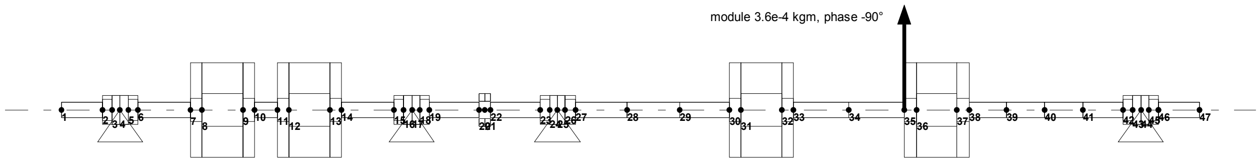

Figure 7. Case 2: modiarot test rig.

The mathematical model includes a modal representation of the flexible foundation. Figure 8 and figure 9 show the additional vibrations: several resonances of shaft and of supporting structure occur in the lower speed range (600 – 1400 rpm), apparently excited by the applied unbalance only.

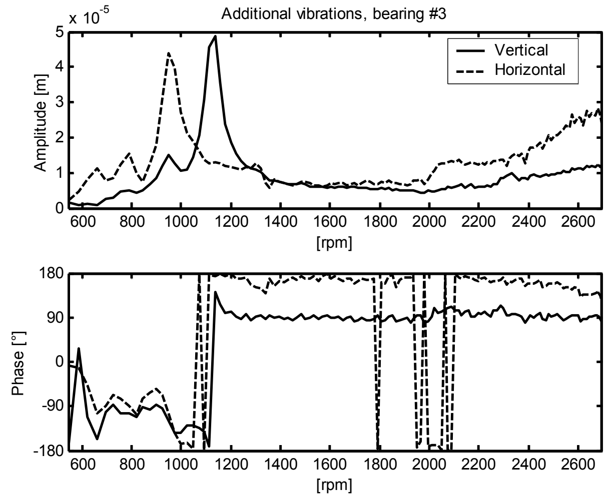

Figure 8. Case 2: additional vibrations due to the unbalance on node 35, module 3.6e-4 kgm phase -90° measured in bearing #3.

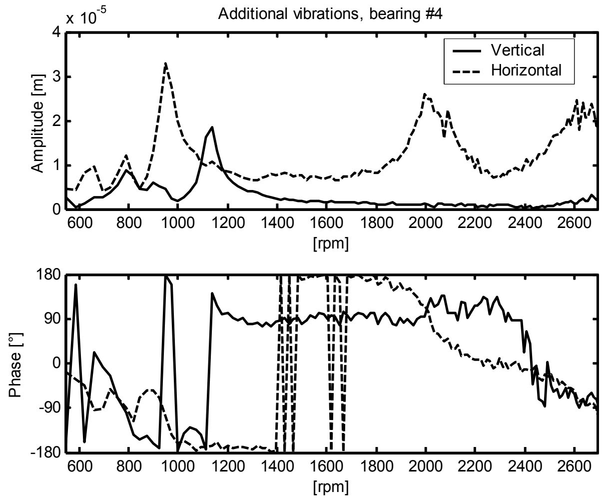

Figure 9. Case 2: additional vibrations due to the unbalance on node 35, module 3.6e-4 kgm phase -90° measured in bearing #4.

Figure 10 shows the results of the identification procedure for single rotating speeds: in the higher speed range (1400 – 2700 rpm) the identification has success, position, module and phase are identified with good accuracy, the low value of the residual indicates good quality of the identification procedure.

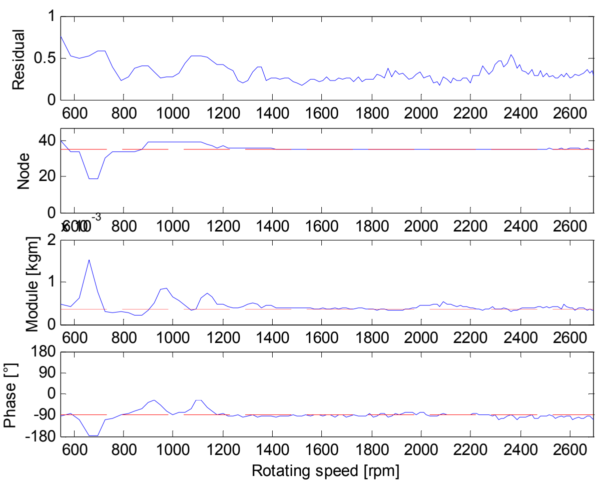

Figure 10. Case 2: residual, node, module and phase identified as a function of the rotating speed.

But in the low speed range where resonances occur the identification results are gradually lost. This confirms that where high dynamic responses of the mechanical system are measured, with rapid changes of amplitudes and phases, the model based identification fails because the model is not able to follow exactly the experimental behaviour.

Figure 11 shows again that unbalance cannot be distinguished by local bow or other causes which generate only 1xrev. components.

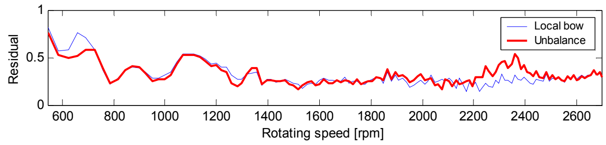

Figure 11. Case 2: comparison between the residuals of different fault types.

3.3. Case 3 Modiarot test-rig with coupling misalignment

In the same test-rig a coupling angular misalignment of 5.45 mrad with an angular phase of -120° has been introduced. The obtained additional vibrations in bearings 3 and 4 are represented in figure 12 and figure 13, where a rather "rich" dynamic behaviour is shown. In this application the test rig was operated up to its maximum speed of 5700 rpm.

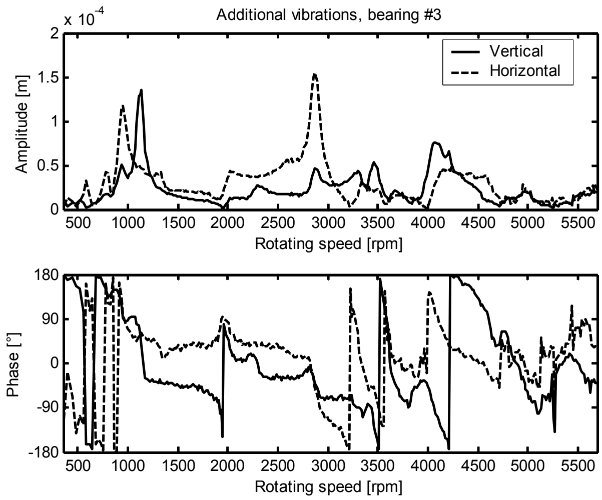

Figure 12. Case 3: additional vibrations due to the angular coupling misalignment, module 5.45 mrad phase -120° measured in bearing #3.

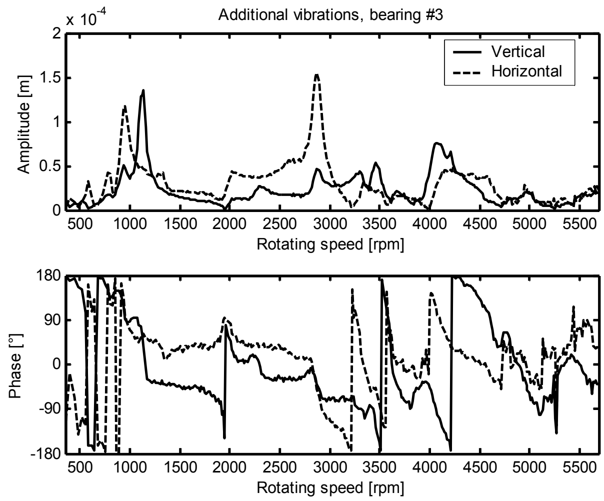

Figure 13. Case 3: additional vibrations due to the angular coupling misalignment, module 5.45 mrad phase -120° measured in bearing #4.

Figure 14 represents again the identification results: the identification of position, module and phase is accurate for all speeds in the range 1400 – 2700 rpm (as in previous case), and the corresponding residual is low. In the low speed range and in the high speed range the position is always identified, but the amplitude is affected by consistent errors and its phase by smaller errors.

Figure 15, in which the residuals of angular misalignment are compared to those of unbalance, shows as usual that single speed identification is unable to discriminate between malfunctions with similar symptoms.

Considering instead a selection of rotating speeds obtained by discarding speed ranges in which resonances occur, the angular misalignment is not only identified with high accuracy, but, due to the low value of the residual (less than .4) is identified as the most probable malfunction candidate, among all possible candidates (Bachschmid and Pennacchi, 2003).

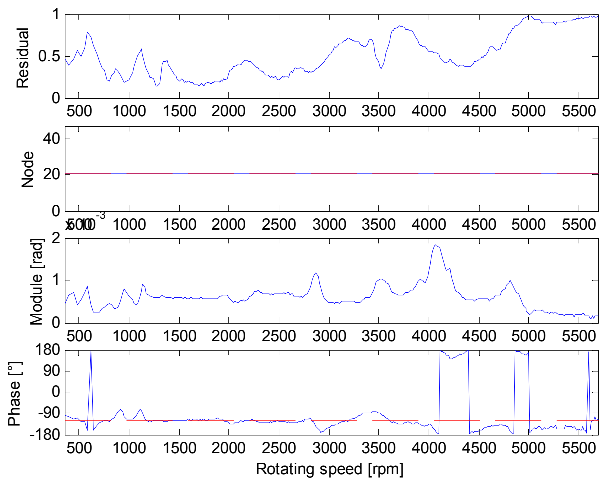

Figure 14. Case 3: residual, node, module and phase identified as a function of the rotating speed.

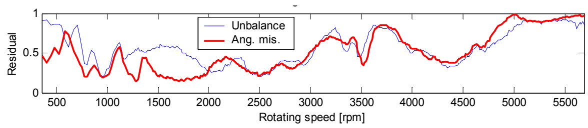

Figure 15. Case 3: comparison between the residuals of different fault types.

3.4. Case 4 320 MW turbo-group with local bow due to rub

The 320 MW turbo-group, which shaft is represented in figure 16, experienced a rub at normal operating speed, which generated a local bow in the position shown in the figure.

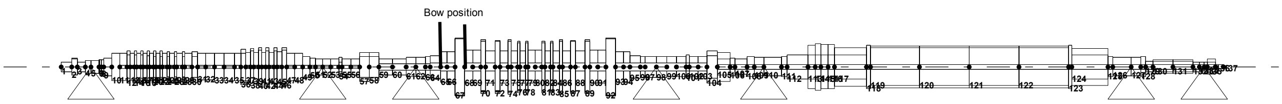

Figure 16. Steam turbogenerator model.

During the coast down transient the measured vibrations represent the actual "faulty" vibrations from which then the additional vibrations used for the identification are obtained by comparison with a previously recorded coast down transient. The single speed identification results are shown in figure 17: the results can be considered good, discarding the speed range below 600 rpm, where the oil film bearing behaviour becomes uncertain due to hydrostatic lubrication contribution, not considered in the model.

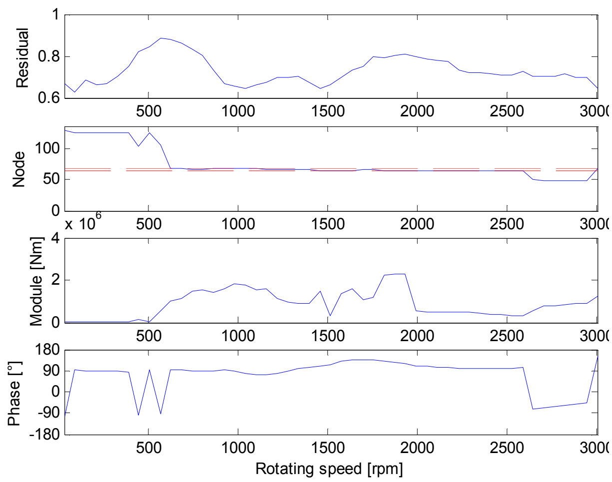

Figure 17. Case 4: residual, node, module and phase identified as a function of the rotating speed.

The bending moment amounts which generate the local bow are changing, but also the rub condition was changing during the run down with changing vibration amplitudes. Further the bending moment values for a given local bow depend obviously also on diameter and length of the element in which the bow has developed. Therefore different bending moments will result from identification when the identified position changes from one element to another.

At rated speed however the position, amount and phase are identified with good accuracy, which is confirmed by the low residual. Comparison with other malfunctions, omitted for brevity, did not give any further indication about the nature of the malfunction.

3.5. Case 5 320 MW turbo-group with unbalance

For another 320 MW turbo-group, represented in figure 18, run down transients before and after a balancing weight application were available, so the additional vibrations due to an unbalance could be obtained.

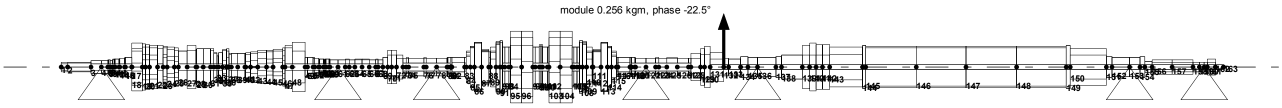

Figure 18. Steam turbogenerator model.

Figure 19 shows that the single speed identification process has success in almost all the speed range, but the best results are obtained in the speed range between 1500 and 3000 rpm. Position amount and phase are identified with high accuracy. In particular also at its rated speed the identification has success. In the lower speed range the position is identified, but amount and phase become more uncertain.

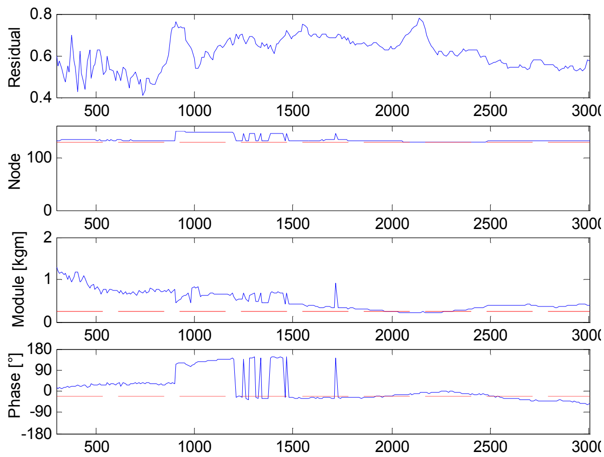

Figure 19. Case 5: residual, node, module and phase identified as a function of the rotating speed.

4. Conclusions

Single speed model based identification of faults or malfunctions can be successful, provided that the system model is linear and time invariant.

Conditions close to resonances should be avoided for succeeding in the identification procedure. This fact is probably due to inaccuracies of the models, which are unable to predict the vibrations of complex systems in these conditions where amplitudes and phases are rapidly changing.

Discrimination between different malfunctions which generate similar symptoms (e.g. faults that generate only 1xrev. vibration components) seems to be impossible with single speed identification: only considering different rotating speeds this discrimination is possible.

Despite these drawbacks of single speed identification, it seems to be possible to use this approach also in industrial applications, checking accurately the obtained results and being aware of its limitations. Balancing procedure results could be predicted by identifying the unbalance in correspondence of each one of the available balancing planes: same unbalance placed in opposite angular position (180°) in the balancing planes should be able to reduce consistently the measured vibrations, independently of the real type of malfunction which caused the measured additional vibrations. Therefore the behaviour of a rotor with a coupling misalignment or a bow could be corrected by suitable balancing masses.

Single speed identification could suggest position and amount of suspected fault, but this identification should be validated by the data collected during coast down transients.

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