Analysis of the Shaft Thermal Bow Induced by Rotor-to-Stator Rubs

P. PENNACCHI, A. VANIA
Department of Mechanics
Politecnico di Milano
Via La Masa, 34 – I20158 Milano
ITALY

Abstract: - The friction forces due to light rotor-to-stator rubs in rotating machines can cause a time-varying thermal bow of the shafts. Owing to this, stable partial arc rubs and full-annular rubs can induce mainly progressive changes in the synchronous vibrations of the rotor-trains. Sometimes, stable or unstable synchronous spiral vibrations can occur. The equivalent bending moments that cause the shaft bow can be estimated with model based identification methods developed in the frequency domain: these techniques minimise the error between the machine experimental vibrations and the system response provided by the model. In this paper, the results obtained by the analysis of the experimental response of a turbine-generator unit that was subjected to rotor-to-stator contacts are shown together with the strategy used to identify the causes of the rubs. The rate of the increase of the time dependent bending moments induced by the rubs, estimated in this investigation with an identification technique, provided useful information about the thermal characteristics of the system which should be modelled by means of equivalent parameters in more complex and accurate methods in which the motion and thermal equations of the fully assembled machine are integrated in the time domain.

Key-Words: rotating machine vibrations, rotor-to-stator rubs, spiral vibrations, diagnostics, fault identification.

1 Introduction

Radial and angular misalignments of the rotor-trains as well as high shaft vibrations induced by several machine malfunctions can cause rotor-to-stator rubs in rotormachinery. Heavy rubs can cause impacts, chaotic motions as well as subsynchronous and supersynchronous vibrations of the shafts. On the contrary, light partial arc and full-annular rubs often cause progressive changes mainly in the synchronous vibrations (1X). Sometimes, depending on the mechanical and thermal characteristics of the system, as well as on the shaft rotating speed, stable or unstable synchronous spiral vibrations can occur [1-4]. In fact, the heat introduced into the shaft by the friction forces caused by the rotor-to-stator contacts induces a shaft bow that, in consequence, causes synchronous vibrations which are added to the initial vibration vectors due, for instance, to a machine misalignment or to the unavoidable residual unbalance of the shafts.

In general, the point of the external surface of the shaft at which the contacts occur is called hot-spot. Often, in the case of light rubs, the position of the hot-spot moves slowly but continuously along the circumference of the cross-section of the rotor during the rub evolution. The rate of this change depends on several factors like the severity of the contacts as well as the mechanical and thermal characteristics of the system. At the same time, the magnitude of the time-varying bow of the shaft changes together with the amplitude of the 1X vectors due to the sum of the initial vibrations and the vibrations induced by the rubs only. Owing to the simultaneous and continuous changes in amplitude and phase of the 1X vibrations the end of these vectors describes a spiral curve. If the amplitude of the 1X vector increases the spiral vibrations are unstable. That is, the rotor heating due to the friction forces can cause a shaft bow whose magnitude grows with time together with the amplitude of the vibrations, while the phase of the 1X vectors turns progressively.

The dynamic stiffness of both stator and journal bearings significantly affects the machine dynamic behaviour. Owing to the occurrence of high vibration levels and low thicknesses of fluid-films in seals and journal bearings non linear effects can become considerable. If the rubs occur during runups and coastdowns the shaft bow evolution can become more complex owing to the passing through shaft flexural critical speeds as well as to the changes in rotating speed and unbalance forces.

In real machines the rub occurrences can be extinguished by changing some suitable process parameter. However, also changes in the orbit shape and wideness as well as consequent changes in the shaft centreline position inside bearings and seals can allow the available radial clearance to become sufficient to extinguish the rubs. In this case, the end of rotor-to-stator contacts and friction forces nullifies the heat introduced in the shaft. If the rotating speed is sufficiently high the rotor-trains are straightened and the shaft thermal bow decreases in short time.

The equivalent bending moments that cause the shaft bow induced by the rubs can be estimated with model based identification methods developed in the frequency domain [5, 6]: these techniques minimise the error between the machine experimental vibrations and the response obtained with the model of the fully assembled machine. In the past the authors developed model-based diagnostic techniques aimed to identify type, severity and location of faults in rotating machines as well as methods aimed to measure the accuracy of the results provided by the identification process [7].

In this paper, the results obtained by applying these methods to the experimental response of a turbine-generator unit that was subjected to partial arc rubs are shown together with the strategy used to identify the causes of the rotor-to-stator contacts.

However, the bending moments due to the shaft bow can be evaluated with more complex and detailed models in which the motion and thermal equations of the fully assembled machine are integrated in the time domain. That is these methods allow the rub evolution to be studied at any time. These techniques require equivalent parameters of the machine thermal properties to be estimated. The accuracy of the results obtained with these sophisticated techniques can be significantly affected by the care with which the model has been tuned. Therefore, the rate of the increase of the time dependent bending moments induced by the rubs, estimated in this investigation with an identification method, provided useful information about the thermal characteristics of the system which should be described by means of equivalent parameters in more complex and detailed models.

2 Case History

The diagnostic strategy developed to investigate the rotating machine vibrations caused by light rotor-to-stator rubs has been validated by the analysis of the experimental response of a turbine-generator unit installed in a cogeneration power plant. The machine rotor-train was composed of a gas turbine, a generator and a steam turbine. The two turbines were located at the opposite ends of the generator (Fig. 1).

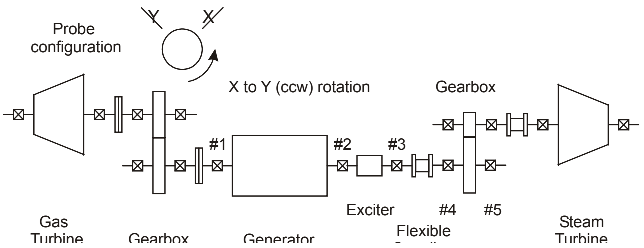

Fig. 1 Machine train diagram.

The two turbines were joined to the generator by means of geared transmissions. Two flexible couplings were mounted in the rotor-train between the gear shafts and the shafts of generator and steam turbine. The operating speed of the generator was 3000 rpm while the operating speeds of the gas turbine and the steam turbine were 3612 rpm and 6803 rpm, respectively. The power of the unit was 50 MW. The generator rotor was supported on three oil-film journal bearings. The numbering of these bearings used in the paper is shown in Fig. 1.

Two proximity probes were mounted 90 degrees apart (XY) on each journal bearing. A monitoring system allowed the machine vibrations to be controlled during runups and coastdowns as well as in the operating conditions. An order analysis of the harmonic content of the machine vibrations was carried out continuously.

2.1 Rub detection

At the end of a machine outage required to execute planned maintenance actions the power unit was restarted. The vibration levels were acceptable both at the operating speed and in the ranges close to the flexural critical speeds of each shaft of the machine-train. However, when the electric load reached 46 MW the amplitude of the 1X vibrations on bearing #3, very close to the exciter, increased slowly but continuously. During a time interval of about four hours the amplitude of the major axis of the 1X filtered orbit increased from 40 μm pp to 152 μm pp.

As the trip level of the generator was set to 180 μm pp it was decided to do an attempt to reduce the vibration levels by decreasing the electrical load. A reduction of the load of only 4 MW was sufficient to stop the trending of the generator vibrations and to cause a progressive decrease of the 1X vibration amplitudes. Afterwards, further attempts to reach the maximum power value of 50 MW caused further significant increases of the levels of the generator vibrations (1X), especially at bearing #3. Every time small reductions of the electric load caused the vibration levels to decrease in short time.

Figure 2 shows the trend of the 1X vibrations measured at bearing #3 during a period of about sixty hours during which some amplifications of the vibration levels had occurred. In general, significant turns of the phase were associated with the peaks of the 1X vibration amplitude.

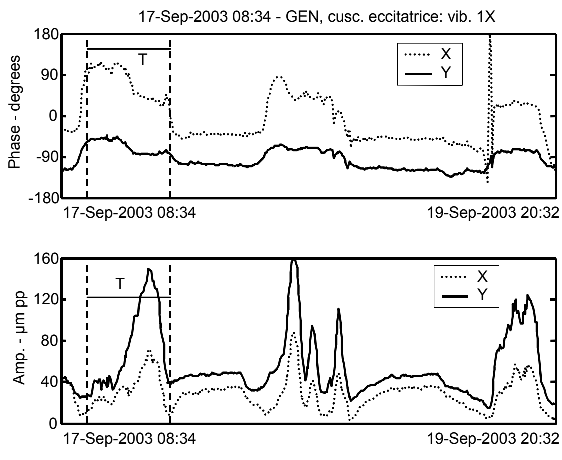

Fig. 2 Trending of amplitude and phase of the 1X vibrations measured at bearing #3 (exciter) in running state.

In order to investigate the causes of this abnormal dynamic behaviour of the generator the 1X filtered orbits as well as the journal centreline position inside the bearing were analysed. This investigation indicated that the abnormal events that affected the generator vibrations were caused by rotor-to-stator rubs. Figure 3 shows the shaft centreline path occurred in bearing #3 during a runup carried out after the maintenance. The shaft rotation was counter-clockwise (CCW). Therefore, the shape of this centreline curve (solid line) is abnormal as at the operating speed the journal position was in the lower left quadrant of the shaft centreline plot. Owing to this it was supposed that an undesired lateral pre-load caused, by a machine misalignment, was acting on bearing #3.

In Fig. 3 a centreline curve obtained using the transient data collected before the machine outage is shown (dashed line). The presence of an abnormal behaviour is confirmed by the comparison between these two centreline curves. In normal operating conditions the vertical component of the journal position evaluated at 3000 rpm was significantly higher than that one measured after the maintenance.

In Fig. 3 the two centreline curves have been plotted by supposing that when the machine was stopped the journal lay on the bottom of the bearing, however, just owing to a lateral misalignment the initial position of the two curves could be different.

The abnormal shaft centreline position affected significantly the geometry of the oil-film inside the bearing. Owing to this also the dynamic stiffness of bearing #3 was abnormal. In fact, the analysis of the 1X filtered orbits showed that the inclination angle of the major axis of the 1X elliptic orbits, evaluated with respect to the horizontal axis, was about 150°. In general this angle is ranging from 20° to 60°. Anyway, it should be lower than 90°.

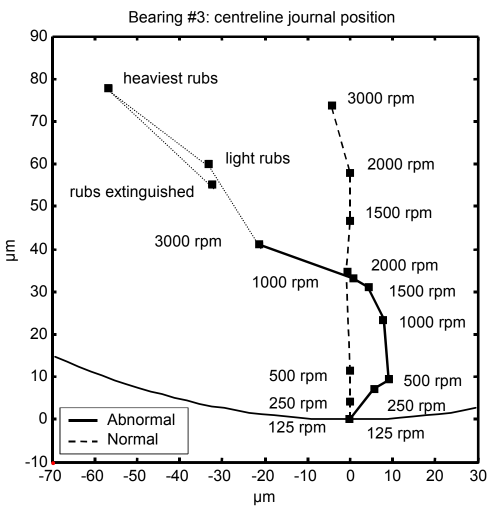

Fig. 3 Centreline plot of bearing #3. Comparison between gap voltage data collected before and after the maintenance.

Figure 4 shows some 1X filtered orbits of the journal inside bearing #3 measured during the rub evolution. It is possible to note that the abnormal inclination of the major axis was always present during the rub events. Moreover, the ellipticity of the orbit, that is the ratio between the amplitudes of minor and major axes, was scarcely affected by the orbit wideness. This is confirmed by the results of an analysis of the orbit shape that are shown in Table 1.

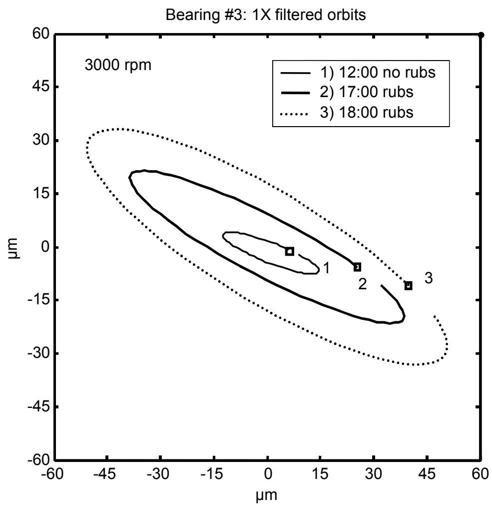

Fig. 4 1X filtered orbits of the journal inside bearing #3 measured during the rub evolution.

Table 1 Shape analysis of 1X filtered orbits at bearing #3.

Rub evolution	Time	Degree of ellipticity	Major axis amplitude	Major axis inclination
			μm pp	degrees
absent	12:00	0.1957	28.9	159°
present	15:00	0.1656	39.6	166°
present	16:00	0.1268	64.6	167°
present	18:00	0.2628	118.0	149°
present	19:00	0.3389	141.1	150°
present	19:10	0.3420	152.4	146°
extinguished	20:45	0.4636	62.8	161°
extinguished	21:30	0.2351	38.6	141°

Visual inspections of the generator shaft, limited to the visible portions located near the exciter, were carried out both in the operating condition and after a planned coastdown. These inspections showed that light but evident contacts between the generator shaft and a seal mounted on a bakelite casing of the exciter had occurred.

At first, the amplitude of the 1X vibrations were rather low however, owing to an horizontal misalignment of bearing #3 it is possible to suppose that the available radial clearance of the seal was rather small. The rise of the electrical load caused changes in the shaft centreline position that reduced further on the available radial clearance and induced shaft-to-casing contacts. The changes in the average journal position inside bearing #3 that occurred during the rub evolution are shown in Fig. 3. The additional lateral displacements of the journal in the lower left quadrant of the bearing confirmed the presence of an unusual behaviour caused by a machine misalignment.

The flexibility of the part of the casing involved in the rubs allowed light contacts rather than heavy impacts to occur. Owing to this, stable partial arc rubs occurred at every shaft revolution. The rotor heating due to the friction forces caused a shaft thermal bow whose severity increased with time as long as the electric power was decreased. Even little changes in this process parameter showed to be sufficient to cause changes in the shaft centreline position that allowed the rubs to be extinguished.

3 Fault Identification

As said above the evolution of the rotor-to-seal rubs can be simulated with complex and accurate methods by means of which the motion and thermal equations of the fully assembled machine are combined and integrated in the time domain. These techniques allow the time-dependent bending moments induced by the shaft bow to be evaluated.

On the contrary, in this investigation the same bending moments have been estimated with an identification method developed in the frequency domain [5-7]. For the sake of brevity this diagnostic strategy developed in the past by the authors to identify rotating machine faults using model-based methods is not described in this paper. Essentially the most common faults and malfunctions can be modelled with suitable sets of equivalent excitations, that is forces and moments which are applied to nodes of the Finite Element Model of the machine-train by means of which the system response is simulated. The magnitude, phase and location of each term of the equivalent excitations are identified by minimising the error between the experimental vibrations and the system response evaluated with the model. Weighted least squares error methods have been used to identify the equivalent excitations.

A Finite Element Model composed of the generator rotor and the slow shaft of the gearbox that joined the generator with the steam turbine has been created (Fig. 5). In this case study the shaft bow induced by the rubs has been modelled with two opposite bending moments whose amplitude and phase were time-dependent. Contrarily to the common fault identifications in this case study the location of the equivalent excitations was well known.

The bending moments that should be identified should be able to simulate the dynamic effects due to the shaft bow only. Therefore, in the fault identification process the experimental 1X vibrations of the generator that can be ascribed only to the rubs have been considered. These vibrations have been evaluated at each measurement point by subtracting the 1X vibration data collected just before the rub occurrence to the 1X vibration vectors measured during the rub evolution. If the non linearity in the system response are negligible these additional vibrations can be considered the effects due to only the shaft bow. The vibrations measured only at the bearings #2 and #3 have been used to identify the time dependent equivalent excitations. In this investigation, only the data contained in the interval shown in Fig. 2, associated with the first rub occurrence, have been considered.

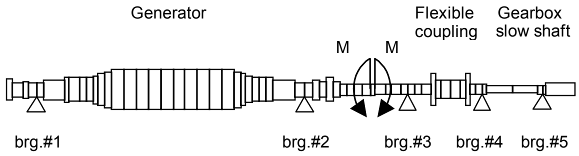

Fig. 5 Finite Element Model of the generator rotor.

Figure 6 shows the amplitude and phase curve of the bending moments evaluated over the duration of the first rub occurrence and in the period just after the rub extinction. The changes in the amplitude of the bending moments were progressive. Also the phase of these excitations showed a significant turn in the first part of the rub evolution. The rate of the changes of the moment amplitude depended on the thermal properties of the system as well as on the mechanical properties of the obstacle and the severity of the contacts. Therefore, the results obtained with this investigation can be used to tune some equivalent parameters of complex models in which the thermal equations combined with motion equations are integrated in the time domain.

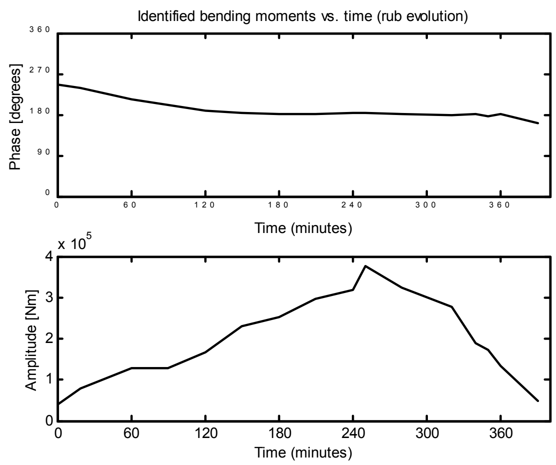

Fig. 6 Amplitude and phase of the identified bending moments that simulate the time dependent shaft bow.

In Figure 7 the 1X additional vibrations measured at bearing #3 in the X and Y directions are compared with the respective numerical response evaluated with the model by considering the identified bending moments as system excitations. It is important to emphasise that the dynamic stiffness of the oil-film of bearing #3 had been suitably tuned to simulate the effects of the abnormal shaft centreline position experienced by the monitoring data. Owing to this, also for the simulated vibrations the amplitude of the 1X vectors in the Y direction was higher than that one evaluated in the X direction.

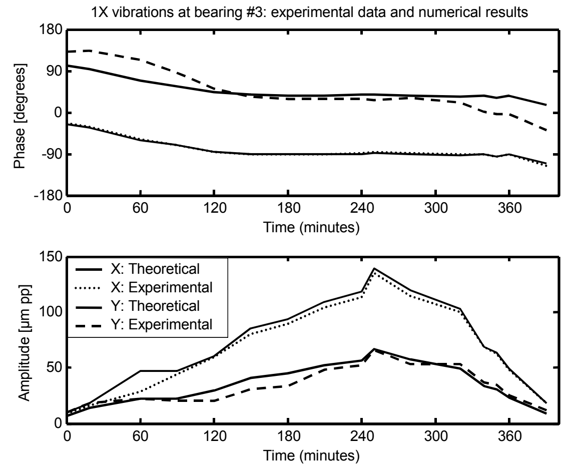

Fig. 7 Comparison between the experimental additional vibrations (1X) at bearing #3 and the respective vibrations caused by the identified bending moments.

Let us denote $\mathbf{X}_{\text{exp}}$ the vector that contains the additional vibration data measured at the bearings #2 and #3 during the time interval that has been considered to identify the fault. Moreover, let us denote $\mathbf{X}_{\text{th}}$ the vector that contains the vibration data provided by the model. An estimate of the relative error, ε, obtained with the identification process can be defined with the following expression:

$$\varepsilon = \sqrt{\frac{(\mathbf{X}^*_{\text{th}} - \mathbf{X}^*_{\text{exp}})^{\mathrm{T}} (\mathbf{X}_{\text{th}} - \mathbf{X}_{\text{exp}})}{\mathbf{X}^{*\mathrm{T}}_{\text{exp}}\ \mathbf{X}_{\text{exp}}}} \qquad (1)$$

where * indicates the conjugate vector. This error is called residual. It has been evaluated for each identification of the time dependent bending moments estimated during the rub evolution. Figure 8 shows the curve of the residuals associated with every identification of the fault.

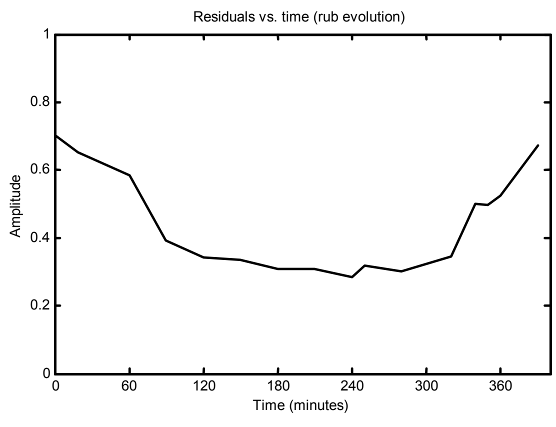

Fig. 8 Residuals evaluated at different rotating speeds.

When the magnitude of the shaft bow was not very high also the levels of the respective additional vibrations were sufficiently low. In these cases the relative error of the identification process showed higher values. On the contrary the lowest residuals have been obtained with the fault identifications associated with the highest vibrations of the generator shaft. In addition to this an estimation of a global residual has been carried out by considering the experimental additional vibrations measured during the complete rub evolution and the respective numerical responses provided by the model. This global residual was 0.3567. This identification error was rather low as confirmed by the good accordance between the experimental vibrations and the numerical response (Fig. 7).

Figure 9 shows the polar plot of the additional 1X vibrations measured on bearing #3 in the Y direction in a shorter time window whose end has been set just before the rub extinction. The end of the 1X vectors that have been considered describes a spiral curve. This phenomenon was more evident at the beginning of the shaft bow evolution, that is when the vibration phase showed a significant turn. The changes in the phase of the 1X vibrations became less important when the vibration amplitude and the severity of the shaft bow increased.

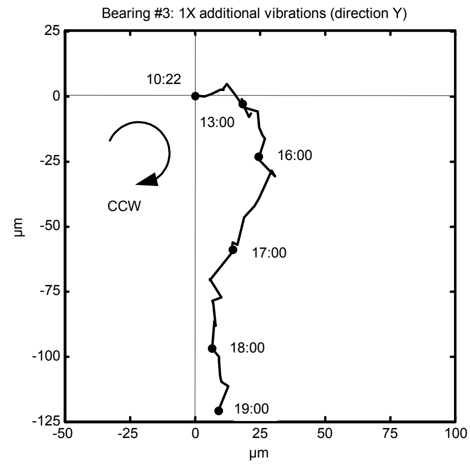

Fig. 9 Polar plot of the 1X spiral vibrations occurred on bearing #3 owing to the rotor-to-stator rubs.

Additionally, an analysis of the harmonic content of the vibrations induced by the rubs has been carried out. This investigation showed that no subsynchronous vibrations occurred. On the contrary, changes in the twice per revolution (2X) vibrations of the generator shaft were concomitant with the amplification of the 1X vibrations. Figure 10 shows the trend of the 2X vibrations measured at bearing #3. However, it is possible to presume that the changes in the 2X vibrations were not a direct consequence of the rubs. In fact, the linear correlation coefficient between 1X and 2X vectors showed to be very high: 0.9089. This correlation showed to be partially non linear when the 1X vibrations reached the highest levels. Therefore, it is possible to suppose that the abnormal centerline position of the journal inside bearing #3, caused by the machine misalignment, induced non linear effects in the oil-film forced transmitted to the shaft. This can be considered a secondary effect caused by the rubs. That is, it is possible to suppose that only one partial arc rub between stator and rotor occurred during a complete shaft revolution.

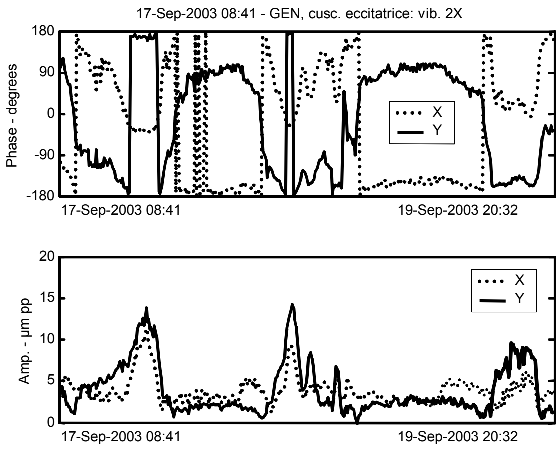

Fig. 10 Trending of amplitude and phase of the 2X vibrations measured at bearing #3 (exciter) in running state.

4 Conclusion

Rotor-to-stator rubs occurred to the generator of a power unit have been detected by means of the analysis of experimental vibrations. The friction forces induced by stable partial arc rubs caused a shaft thermal bow whose amplitude increased with time. Owing to this, unstable 1X spiral vibrations occurred. The time dependent bending moments associated with the bow of the generator shaft have been estimated with a model-based diagnostic technique developed in the frequency domain.

The causes of the rotor-to-stator contacts have been identified by the detection of fault symptoms obtained with the analysis of both journal orbits and shaft centerline plots.

The results of the preliminary diagnostic investigation described in this paper can provide useful information about the not trivial estimation of equivalent mechanical and thermal equivalent parameters of the models used in more complex and accurate methods in which the motion and thermal equations of the fully assembled machine are integrated in the time domain to simulate the rub evolution.

The results obtained by applying one of these methods to the analysis of this case history will be shown in a further paper.

References:

[1] Liebich, R. , Gasch, R., "Spiral vibrations – modal treatment of a rotor-rub problem based on coupled structural/thermal equations", Proc. of *IMechE 6th Int. Conference on Vibrations in Rotating Machinery,* 1996, pp. 405-413.

[2] Kellemberger, W., "Spiral vibrations due to the seal rings in turbogenerator thermally induced interactions between rotor and stator", *Journal of Mechanical Design*, Vol.2, 1980, pp. 177-184.

[3] Larsson, B., "Rub-heated shafts in turbines", Proc. of *IMechE 7th Int. Conference on Vibrations in Rotating Machinery,* Nottingham, UK, 2000, pp. 269-278.

[4] Bachschmid, N., Pennacchi, P. Venini, P., "Spiral vibrations in rotors due to a rub", Proc. of *IMechE 7th Int. Conference on Vibrations in Rotating Machinery,* Nottingham, UK, 2000, pp. 249-258.

[5] Bachschmid, N., Pennacchi, P., "Model-based malfunction identification from bearing measurements", Proc. of *IMechE 7th Int. Conference on Vibrations in Rotating Machinery,* Nottingham, UK, 2000, pp. 571-580.

[6] Bachschmid, N., Pennacchi, P., Vania, A., "Identification of multiple faults in rotor systems", *Journal of Sound and Vibration*, 254 (2) (2002) 327-366.

[7] Pennacchi, P., Vania, A., "Experimental and theoretical application of fault identification measures of accuracy in rotating machine diagnostics", *Mechanical Systems and Signal Processing*, 18 (2004) 329-352.