Proceedings of the XII International Symposium on Dynamic Problems of Mechanics (DINAME 2007),
Varoto, P. S. and Trindade, M. A. (Editors), ABCM, Ilhabela, SP, Brazil, February 26 - March 2, 2007

Some Experimental Results in Rubbing Phenomena Analysis

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Abstract: A test rig designed for the analysis of rubbing phenomena in rotating machinery is described and its dynamic behaviour is presented. A partial seal ring can be approached to the rotating shaft and rubs of different severities can be generated .Orbits of the shaft in bearings and in rub position are measured as well as vibrations and forces of bearing housings and seal support .Noise and torsion vibrations are also measured. Some of the results obtained in different rubbing conditions are shown and compared to the results obtained at the same speed without rubbing. The goal of the analysis is to collect all different symptoms of the rubbing condition which could be measured also in industrial machines for the diagnosis of rubs.

Keywords: rub, partial rub, vibrations , rotordynamics, diagnosis.

INTRODUCTION

Rubbing phenomena can occur frequently in rotating machinery, especially when clearances are tight and problems in alignment conditions between rotating part and stationary parts of the machine have shown. This can occur in the start up procedures of a new machine or of an older machine after an overhaul. The problem can also occur due to a thermal bow of the shaft, or to a thermal distortion of the stationary casing.

Rubs can be of different types depending on where contact occurs on the rotating and on the stationary parts of the machine. Two extreme cases exist: type A , could be called rub on shaft, when contact is in one point only of stationary part and contact point slides continuously on the rotating outer surface of the shaft. Type B, could be called rub on casing, when contact is in one point only of the shaft and the contact point is sliding continuously on the inner surface of the stationary part, this last type is called full annular rub. Type A occurs when shaft vibration amplitude is vanishing small and the shaft static offset is greater than the clearance. Type B occurs when the shaft vibration amplitude is higher than the clearance and the static offset of the shaft is vanishing small. All situations between types A and B are partial arc rub situations: but when the arc of contact is small the rub can be considered type A, when it is large type B.

During rubs of type A the contact may be continuous or intermittent, also with rebounds, depending on stiffness and damping of both rotating and stationary part, and on the size of unbounded orbit as imposed by unbalance and bow of the shaft. If the contact is continuous, the heat introduced in the shaft due to friction has a polar symmetrical distribution and does not produce any additional bow. The heat introduced in the stationary part through the hot spot generally on a sealing ring mounted on the casing of the machine has only little effect on the development of the rub conditions.

When the contact is not continuous then the heat is introduced only during a limited angle of rotation, during which the contact point is sliding on a limited arc of the shaft circumference. The part of the orbit where the whirling shaft gets in contact with the stationary part is defined by the radial deflection: where radial deflection exceeds the remaining clearance, calculated by considering the static offset of the shaft due to alignment conditions, there contact occurs. The same contact conditions are repeated each revolution. Sometimes, when the rub is “heavy” enough, the heat due to friction introduced through the intermittent contact point along the same arc in the shaft, provides unsymmetrical heating of the shaft which gets a thermal bow. In this case also the vibrations, measured in general in correspondence of the bearings of the machine, change and the vibration vector can experience a spiral development, called spiral vibrations, with time. The spiral can be with increasing amplitude, unstable spiral, or with decreasing amplitude, stable spiral, in this last case the phenomenon will become generally periodical. This occurs when shaft starts rubbing and the thermal bow generated by the heat deflects the shaft in such a way that the shaft looses the contact. When heat is again distributed uniformly the process will repeat periodically. The period depends mainly on the shaft diameter in correspondence of the rub location.

Type B of rub occurs when the unbounded orbit of the shaft exceeds the clearance during a consistent part of the shaft revolution (partial arc rub), or during the complete revolution (full annular rub). During the revolution of the shaft always the same point or short arc (hot spot) is in contact with the stationary part and slides along the arc or the complete circumference of the stationary part or casing of the machine. All the heat developed by friction is introduced in the rotating shaft through the same point (hot spot), and the shaft gets a thermal bow In this situation the development of spiral vibrations is rather common. Spiral vibrations, which are also called Newkirk effect, have been described firstly in (Kroon & Williams, 1937) and later roughly modeled firstly by Kellenberger (1980) This phenomenon has then been extensively analyzed with suitable models both for the thermal behavior and for the consequent vibration behavior in (Bachschmid et al, 2000) (Bachschmid et al, 2004).

Many of the physical phenomena which occur during rotor to stator rub have been described firstly in a literature survey of Muszynska (1989) . If the contact is continuous and a full annular rub is generated, the friction force may excite in smaller machines a very dangerous dry friction reverse whirl. If the contact is occasional and associated to rebounds also chaotic motion can result in light shafts. During contact the stiffness “felt” by the rubbing shaft is higher than the flexural stiffness of the shaft: during one rotation the shaft experiences a non linear stiffness and has consequently the typical behavior of a vibrating system with non linear stiffness. All these phenomena have never been measured on big sized industrial machines like steam turbines and generators, which are the main objects of the present analysis on rubbing rotors.

In recent years when the requirement of higher efficiency in turbo-machinery has reduced the clearances between rotating shafts, impellers and blade rows and stationary parts, the analysis of rubbing rotors has become a matter of great interest.

From diagnostic point of view it is necessary from few measurements to recognize rubs. For rubs of type B spiral vibrations constitute a strong symptom of full annular or partial arc rub, but sometimes also spiral vibrations were attributed to other causes (whirling motion in oil film bearings). Rubs of type 1 are more difficult to be recognized, because apparently they affect only lightly the vibrations of the casing introducing some high frequency noise which could be detected by accelerometers or also with acoustic measurements (Stegemann et al, 1993).

A theoretical and experimental research has been started in order to explore all possible symptoms of a rubbing rotor with a rub of type A.

A test rig has been modified to allow to apply rubs of different intensities to a rotating shaft in different operating conditions. A movable support equipped with a partial arc seal ring can be approached to the rotating shaft generating rubs of different intensities. Shaft vibrations in bearings and in the section where contact occurs, vibrations of bearing housings and of the support housing where contact occurs, forces transmitted by all housings to the supporting structure, duration of contact, torsional vibrations and noise emission: all these quantities have been measured.

A small test rig cannot be representative of the behavior of a full size real machine but some scaling effects have been considered so that some results can be transferred from test rig to real machines mainly in cases of the identification if exist or not exist rubbing effects.

THE TEST RIG MODEL

First a draft model of the test rig has been build up in order to simulate its dynamical behavior using a finite element model. The predicted dynamical behavior was used to plane the experimental tests and to place the arc seal rig.

The scheme in Fig. 1 shows the finite element model of the rotor. The shaft is modeled with 29 beam elements, each node has 5 degrees of freedom in order to include also torsion vibrations. Bearings have been modeled with stiffness and damping coefficients. In this first approach the foundation has been considered stiff, and flexural modes as well as torsion modes of the rotor have been calculated. Flexural eigen-frequencies are 16.8 – 16.9 Hz (first mode splitted) and 73.8 Hz (second mode), and the torsion eigen-frequencies are 132 Hz (first mode) and 545 Hz (second mode).

Then an unbalance has been applied to one of the balancing planes of the shaft in order to measure the sensitivity of the bearing response to unbalance forces. The resulting Bode diagrams are shown in Fig. 1: more than 60 microns amplitude in bearing 1 and 40 microns in bearing 2 is the response in correspondence of the first critical speed. The second critical speed is very well damped.

Frequency response / unbalance node 16 (0,0008 kg-m @ 180°)

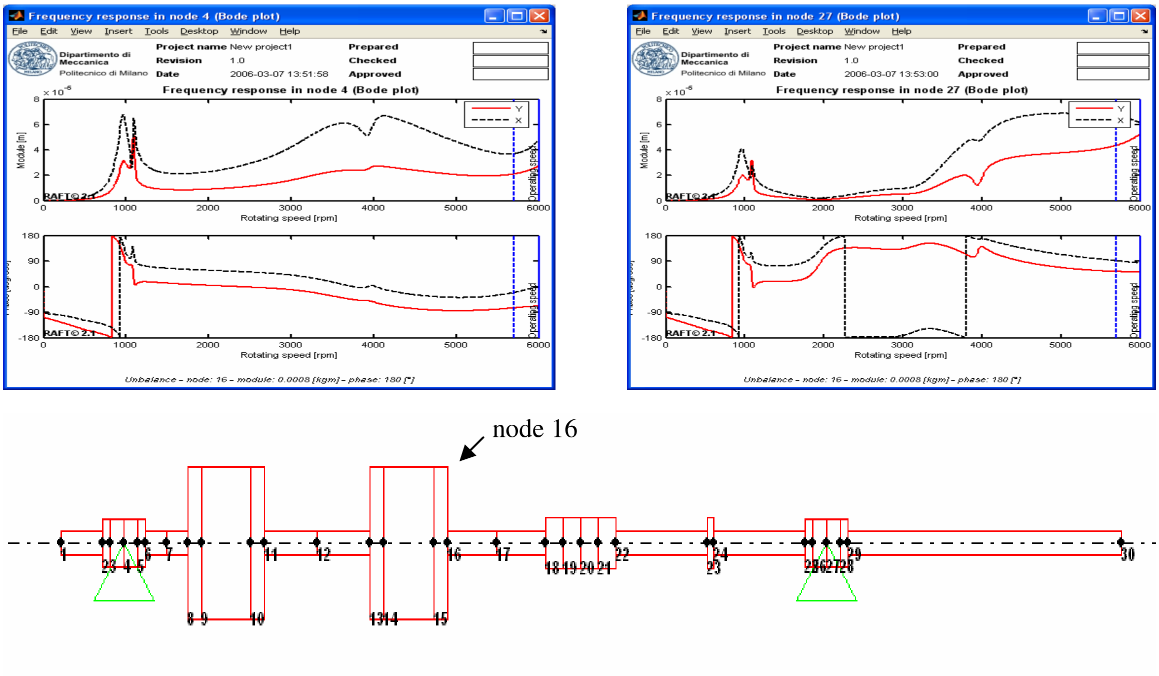

Figure 1 – Finite element model and Bode diagram of unbalance response.

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DESCRIPTION OF THE TEST RIG

The test rig used for these investigations is composed of a horizontal flexible shaft of 1485 mm length with a diameter of 25 mm as shown in Fig. 2. The shaft is connected to an electrical motor driven by inverter by a flexible coupling and is carrying two disks of a diameter of 160 mm and a thickness of 110 mm. The total weight of the shaft is around 500 N. The motor drives the shaft from 0 up to a maximum speed of 6000 rpm, but tests are generally carried out at lower speeds.

The shaft is supported by two lemon shaped oil film bearings and two bearing supports, labelled 1 and 2 in Fig. 2, connected to an oil circulation system to lubricate the bearings. The shaft in correspondence of the bearings has a diameter of 50 mm. The bearing supports are connected by means of two force sensors, in horizontal and vertical direction, to a metallic supporting structure. This makes the entire supporting structure rather flexible. Some supporting structure resonances are within the operating speed range. The entire apparatus is clamped to a massive concrete block isolated from the environment by rubber pads.

Each bearing is equipped with two proximity probes in directions + and – 45° with respect to the vertical direction, as usual in many industrial machines. However after the acquisition the signals from proximity probes are rotated, making the axis of the shaft displacement coincident with the horizontal and vertical directions, this procedure allows to compare all signals in vertical and horizontal directions.

The bearing housings are equipped with two accelerometers for measuring its vibrations in vertical and horizontal direction. A third support, labelled 3 in Fig. 2, is mounted on a cross slide, and can be moved in radial horizontal direction. On this support as shown in Fig. 3, instead of the bearing a half seal ring is mounted, which is brought in contact with the rotating shaft; the diameter of the shaft has been enlarged to 50 mm, and the seal ring has a diameter of 55 mm. The normal contact force is mainly horizontal, and the tangential force vertical. Also this support is equipped with proximity probes, with force sensors and with accelerometers.

Finally also a contact sensor has been designed, tested and installed for measuring the duration of the contact.

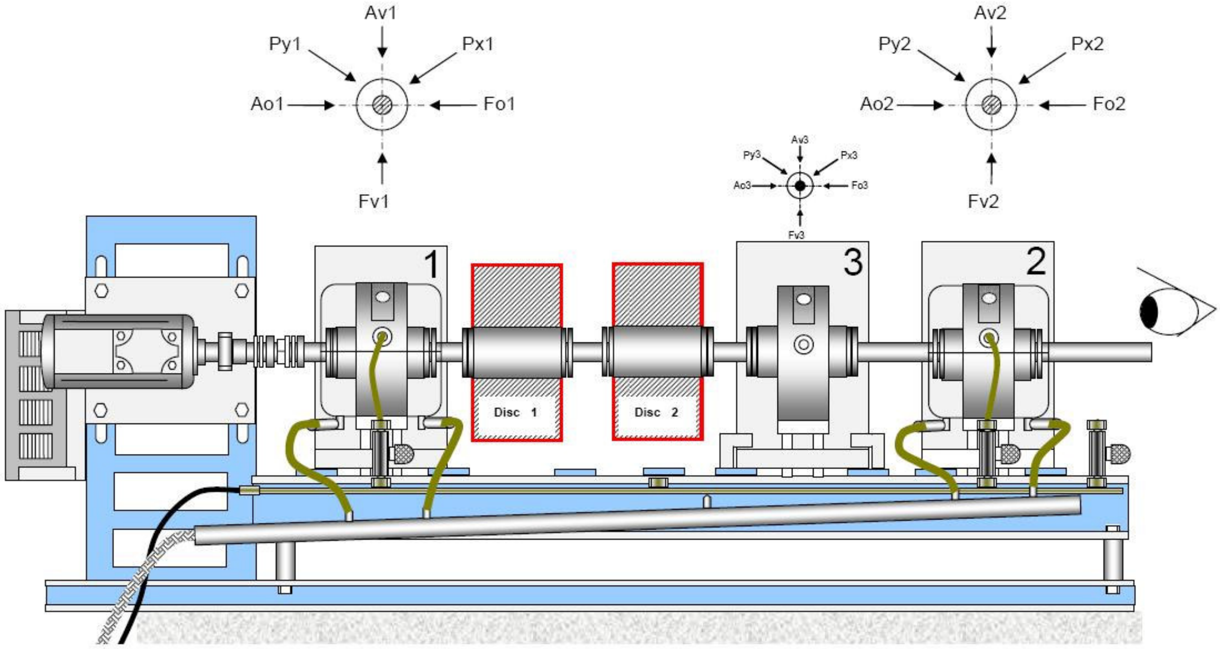

Figure 2 – Schematic diagram of main components of the test rig.

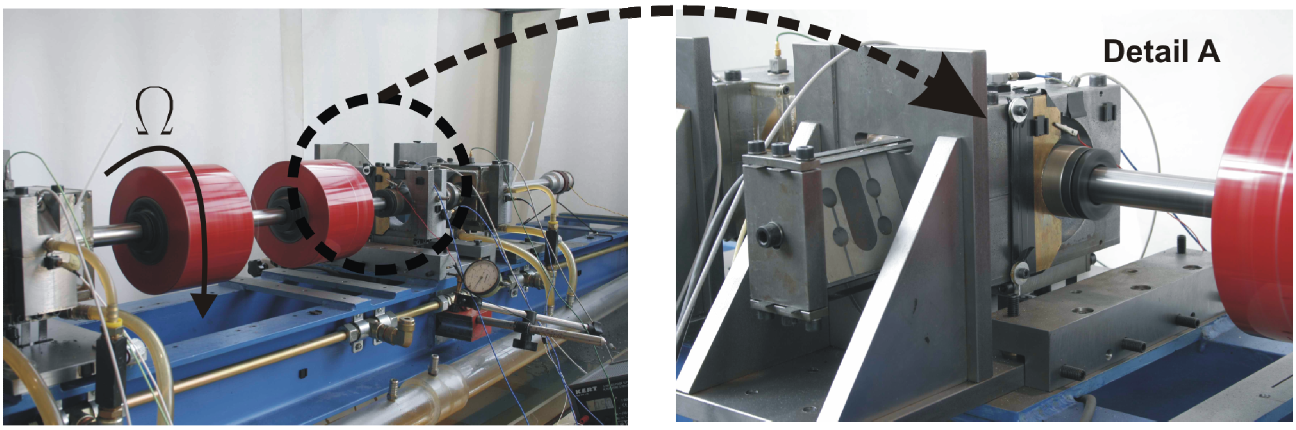

Figure 3 – Test rig photography and detail of the seal ring and its support.

Figure 4 shows the schematic diagram of the probe to measure contact time duration. It is composed by a Wheatstone bridge with one node connected to the shaft, by means of contact brushes, and the other connected to the seal ring. The oil film in the bearings guarantee the electric isolation of the shaft from the base and supports. After tuning, the voltmeter indicates 0 V when no contact occurs, and indicates 2.5 V during contact.

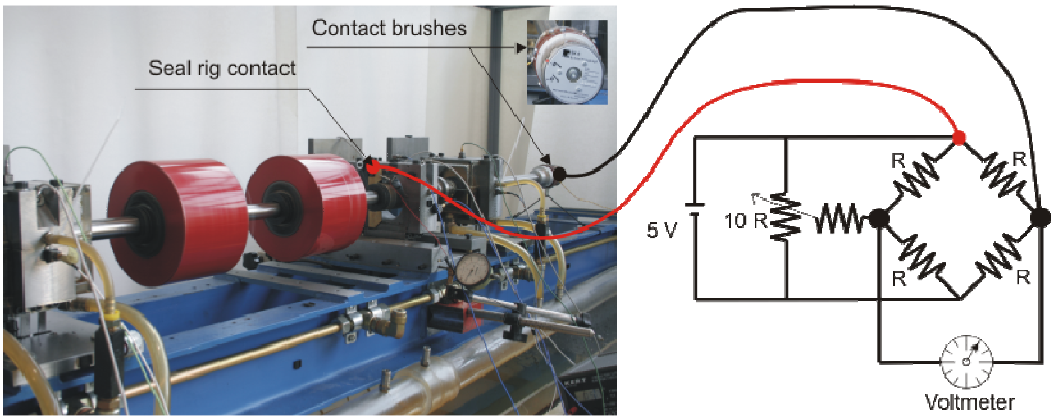

Figure 4 – Schematic diagram of contact time measurement.

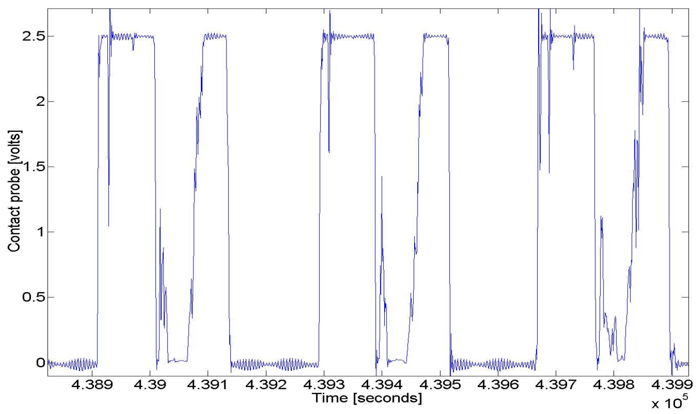

Figure 5 –Typical contact time measurement results: two contacts each revolution are shown.

An example of the contact probe signal is presented in Fig. 5. The small fluctuations in the signal levels are due to noise induced by frequency inverter that drives the electrical motor.

The upper limit in the frequency response of the different sensors are following:

- Contact sensor: > 10 kHz
- accelerometers: 7 kHz
- microphone: 20 kHz
- proximity probes: >1 kHz
- force sensors: 0.1 kHz

Therefore the force sensors in the support 3 are not able to measure the instantaneous contact force, but give a rough indication on the severity of the rub. The force sensors in the supports 1 and 2, where the modifications in the dynamical behavior due the rub is mainly 1x, 2x and 3xrev. of rotational speed, due to the filtering effect of the shaft and the oil film, are able to give a reliable measure of the effects of the rub, being the frequencies of the 3 harmonic components below 0.1 kHz.

FIRST EXPERIMENTAL RESULTS

The simulations have shown that the test rig was suitable for rubbing tests to be performed. After setting up the rig and balancing the shaft, some first experimental results have been obtained. Test with and without additional unbalances have been run, in order to check the sensitivity to unbalancing forces, and to identify all resonances of the system.

Figure 6 shows the Bode diagrams obtained by vector differences of values measured with the applied unbalance of 0.0008 Kg m on disk 2, minus the values measured in the original well balanced situation. The first split critical speed is clearly recognizable, and corresponds well to the simulated one, although the maximum amplitudes are lower: 33 microns in bearing 1 and 28.5 in bearing 2. Also some dynamic response can be noticed at 650 rpm, and strong dynamic responses occur at 2400 and 3000 rpm, which are not predicted by the model. These may be attributed to the supporting structure dynamic response, as will be shown in the following.

In Fig. 7 the left diagrams show the response of the force sensors in horizontal and vertical direction mounted on the seal ring support, which is excited through the supporting structure vibration alone. A clear horizontal resonance occurs close to 1700 rpm, and a sharp vertical resonance occurs at 3000 rpm. The low speed resonance and the effect of the first critical speed of the shaft are also recognizable in the forces transmitted by the supporting structure to the seal ring support.

In the same figure the right diagrams show then the shaft responses measured by the proximity probes in correspondence of the seal ring, which indicate obviously the relative displacements of the shaft with respect to the seal ring support.

In correspondence of the shaft critical speed a relative maximum vibration amplitude of 450 microns is sufficient to generate consistent intermittent 1xrev. Rubs, when the seal ring support is approached to the whirling shaft.

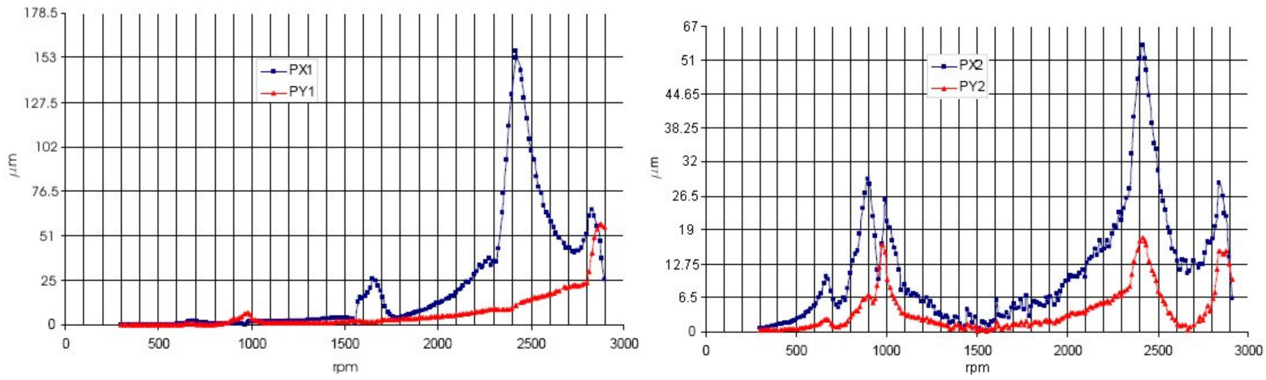

Figure 6 - Experimental Bode diagrams of shaft displacements in bearings 1 and 2 due to specified unbalance (0.0008 kgm)

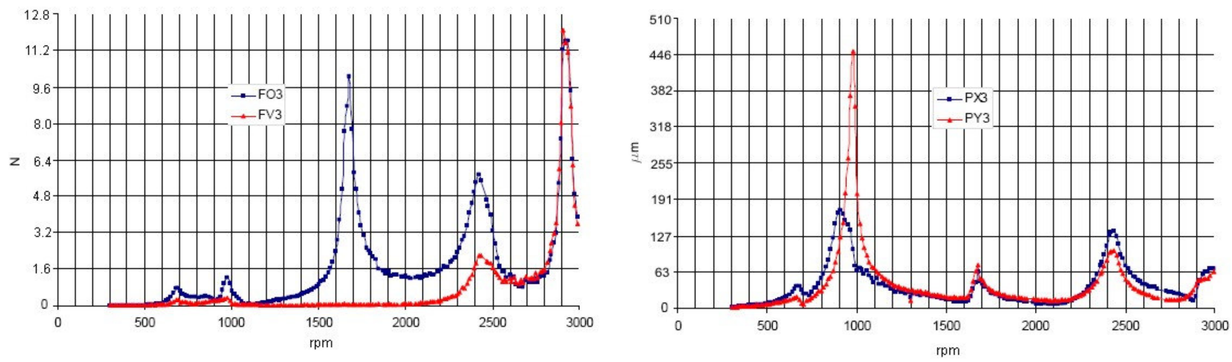

Figure 7 - Experimental Bode diagram of force sensors (left) and proximity probes (right) in seal ring support.

In order to separate the dynamic effects of the supporting structure from those of the shaft, it seemed reasonable to limit the speed range of the shaft for the rubbing tests to 0 – 1500 rpm , with 3 characteristic regions in which investigate characteristic behaviors. Low speed (below critical), close to critical, and high speed (above critical). From point of view of the flexural vibrations the test rig in this speed range is representative of many industrial machines, in which the rated speed is above the first critical speed.

RUBBING CONDITIONS COMPARED TO NON-RUBBING BEHAVIOUR

Since the shaft behavior during rubbing may depend on several parameters such as running speed and intensity of contact force, tests have been made at constant rotating speeds of 600 rpm, 980 rpm (1.st critical) and 1200 rpm, and during speed transients and during changing contact force. The comparisons between behaviors at different speeds, with different contact forces as well as the behavior during speed transients will be the object of a detailed report to follow.

Here the results obtained at the constant speed only, specifically at 1200 rpm, will be shown in terms of shaft orbits in bearings and seal, of contact duration, of contact force, of the vibration of the seal support, of the emitted noise and of the rotational speed variation. This speed which is above the first critical speed of the shaft, could be representative of the rated speed condition of an industrial turbo machine.

All experiments were started without contact and gradually the seal ring support was approached to the rotor shaft. The state of contact was continuously monitored in an oscilloscope attached to the contact probe. Once the contact started generating a light rub, the seal ring was approached still more to the shaft generating severe rub until continuous contact was obtained. Then the support was gradually moved away until the contact was completely loosed. Acquisition system was adjusted with different frequencies of acquisition in such way that at least one point was acquired in each angular degree of the shaft rotation. This methodology results in a sample frequency 7200. Time acquisition was adjusted to permit seal ring approaching and moving away from the shaft slowly, in order to avoid amplitude modulation effects in measured displacements. In the same positions of the seal ring the same results have been obtained, showing excellent repeatability. The sound pressure level was measured in a separated Fourier spectral analyzer, for computing rub against no-rub conditions.

Figure 8 shows the orbits of the shaft measured at bearings 1 and 2 without rub and with two different conditions of rub. The orbit arcs during which contact occurs are depicted in red. Medium rub conditions are generated when the seal ring support is moved to create a small interference between the shaft orbit and the seal ring: in this situation contact occurs only once a revolution of the shaft. When the support is moved further, generating higher interference, with higher contact forces, then severe rub conditions are experienced, contacts with rebounds may occur as shown in Fig. 5, where contact occurs twice a revolution. This can be seen also in figures 8 and 9. Fig. 9 presents the measurements in correspondence of the seal, where the rub effects are obviously more evident.

Light rub modifies slightly the orbits in the bearings, but modifications are strong enough to be registered by a monitoring system. This can be seen better in the spectra of shaft vibrations in bearings 1 and 2 shown in figures 10 and 11.

The orbit modification is stronger in correspondence of the seal, but would not be measured in a real machine.

Consistent strong modifications are obtained in severe rub conditions: orbits are dramatically enlarged and modified in its shape. This behaviour , which differs completely from behaviour at the other rotating speeds, could be due to the non-linear modification of the stiffness "felt" by shaft when it is in contact with the seal: the higher stiffness could bring the shaft critical speed closer to the rotating speed. But this aspect has to be investigated more in detail.

The orbits measured at bearing 2 present some random subharmonic component and a complex shape during rub. This could be correlated with the small size of the orbit and to the reduced load on this bearing. Noise, subharmonic components and rub effects are components which contribute significantly, besides the 1x component, to compose the orbit.

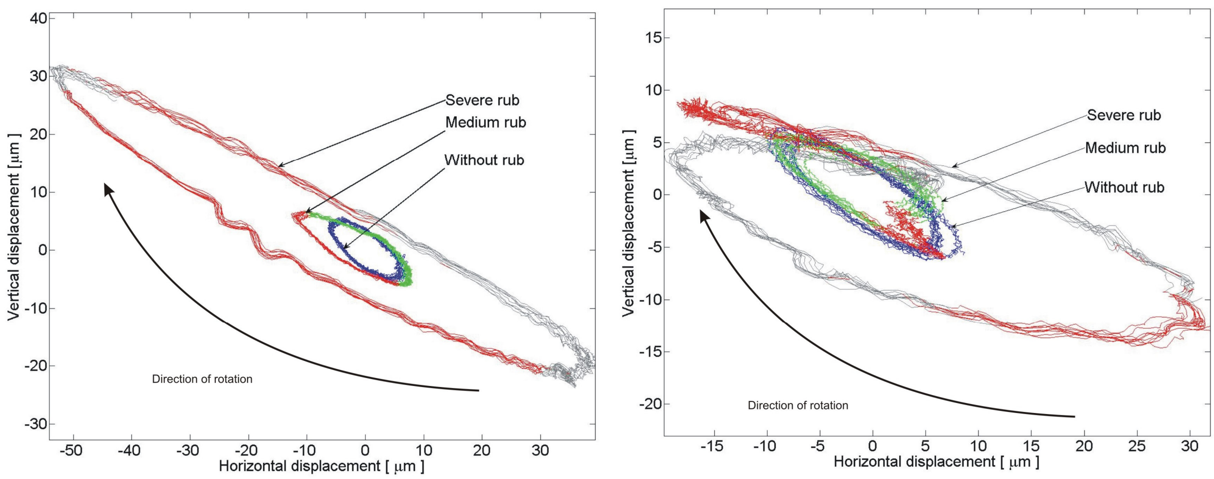

Figure 8 – Shaft orbits at bearing 1 (left) and at bearing 2 (right) of the test rig.

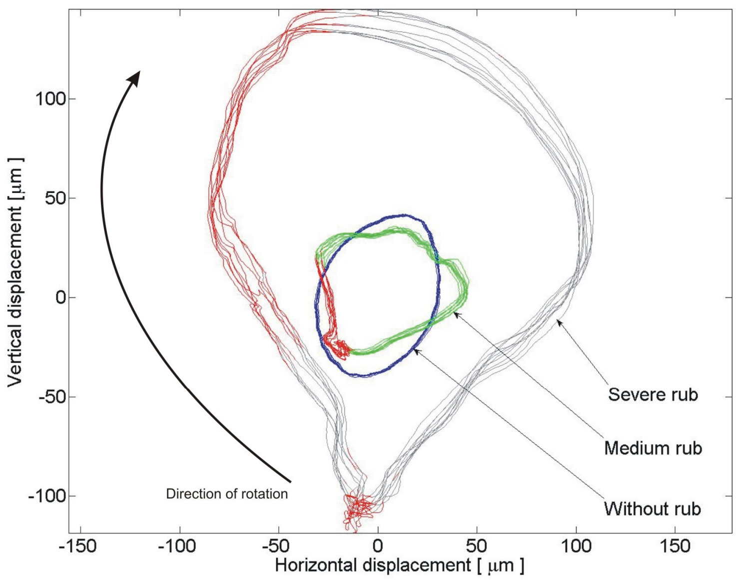

Figure 9 – Shaft orbits at support 3 of the test rig.

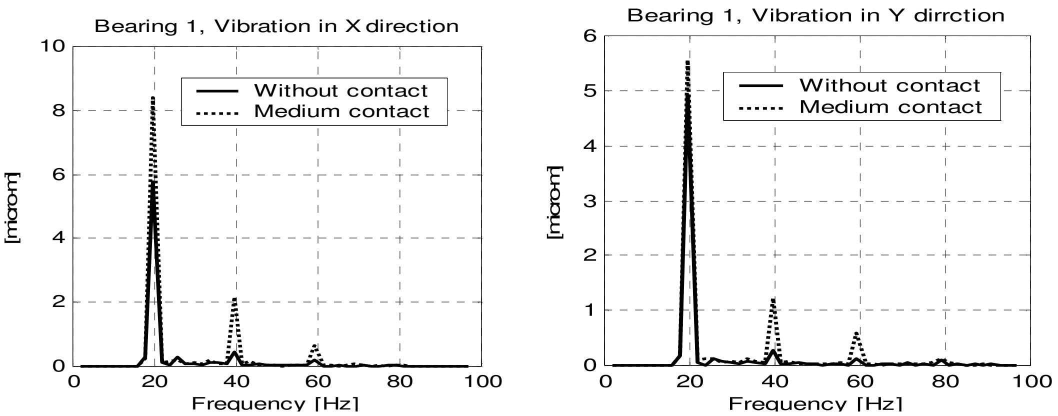

Figure 10 – Spectra of shaft vibrations with and without medium rub in bearing 1

The spectra of the shaft vibrations in bearing 1 of Fig. 10 indicate clearly that the modifications due to light rub are mainly due to 2x and 3x components, which increase consistently their amplitude, rather than the 1x component. Heavy rub will instead increase mainly the 1x component, as shown in Fig. 11

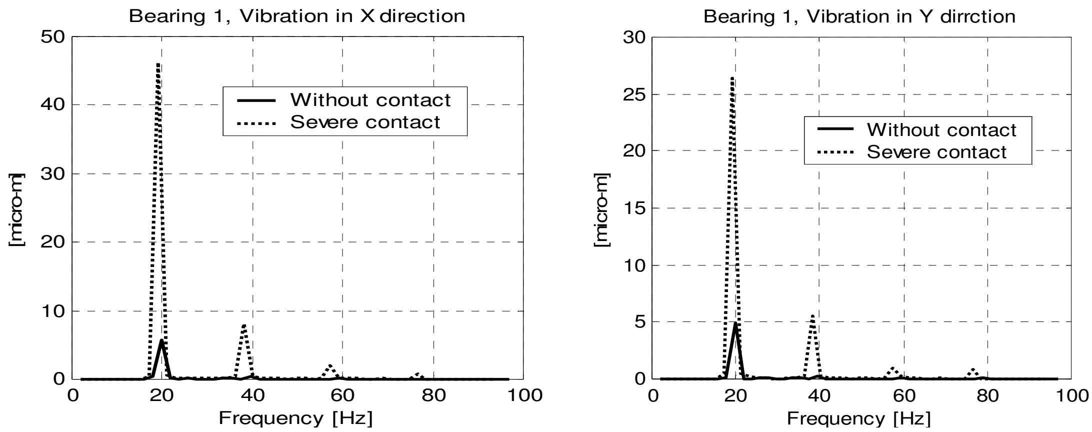

Figure 11 – Spectra of shaft vibrations with and without severe rub in bearing 1

Fig. 12 shows the power spectral density of the vibrations of the seal ring support as measured by the accelerometer, and the associated noise emission, as measured by a microphone placed at a distance of 0.5 meters from support. Both measurements are compared to the same measurement taken in no rub condition. The rub, being its nature mainly impulsive, increases consistently (up to two orders of magnitude) the vibration amplitudes in the complete range of frequency of the accelerometer (1 – 7000 Hz). This result confirms the statement, known from literature, that rubs can be identified in real machines by acceleration measurements in the high frequency range.

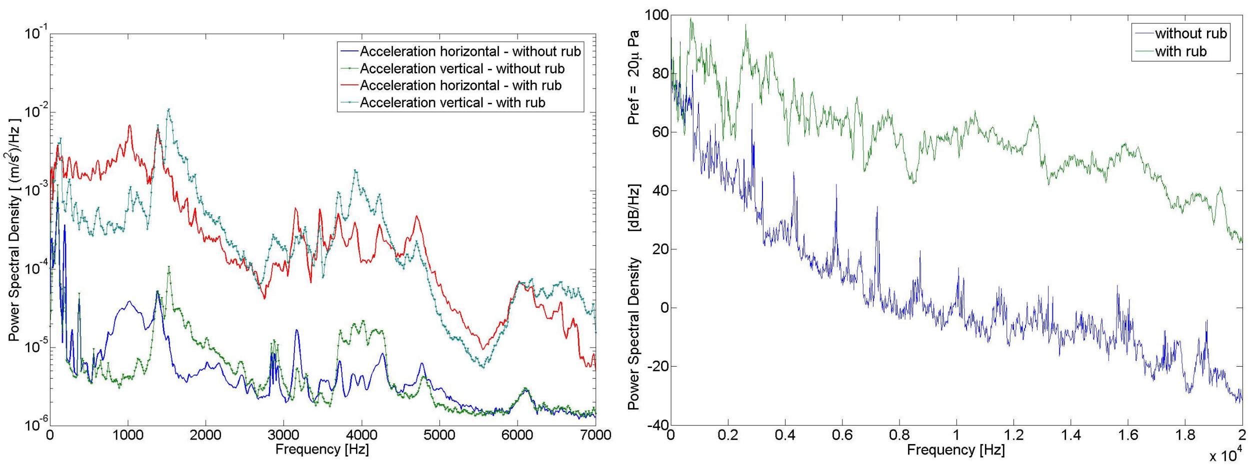

Figure 12 – Power spectral density of acceleration of support 3 (left) and of the sound pressure (right).

The same conclusion can be drawn from noise emission measurement, where still higher frequency components are excited by the rub , with amplitudes which are up to 60 dB higher with respect to the no-rub condition. This result is obviously related to the particular situation of the test rig where the rub occurs in open air conditions; in real machines the rub occurs inside the casing of the machine, the vibrations which are excited by the rub will be in general smaller, due to the stiffness of the casing, and the noise will be much less, due only to structural noise generated by the vibration of the casing. But if the environment where the machine is installed is not too noisy also noise measurements could identify an impending rub.

For giving an idea of the force transmitted to the supporting structure by the seal ring support during light rub conditions also the spectra of the force sensors in the frequency range 0-100 Hz are shown in Fig. 13. These force spectra which represent only roughly the contact forces because the measured forces are affected by the inertia forces of the seal ring support, indicate that the main components are 1x and 2x and that the overall amplitude is in between 10 and 20 N (for light rub). The contact force modifies also the equilibrium conditions of the shaft, therefore also the bearing forces will be modified during rub conditions. This is shown in Fig. 14 : the modifications due to rub are mainly 2x and 3x, as could be expected analyzing the shaft orbits in the bearings. Fig. 15 shows instead the strong modifications in vertical bearing force harmonic components due to a severe rub. Also a frequency shift is recognizable in the spectra: the severe rub reduces slightly the rotating speed.

These forces are generally not measured in industrial machines, but sometimes the inlet oil pressures in the bearings are monitored. Therefore being the pressure related to the forces transmitted by the shaft to the bearing, the modification of the oil pressure frequency components could indicate the occurrence of the rub.

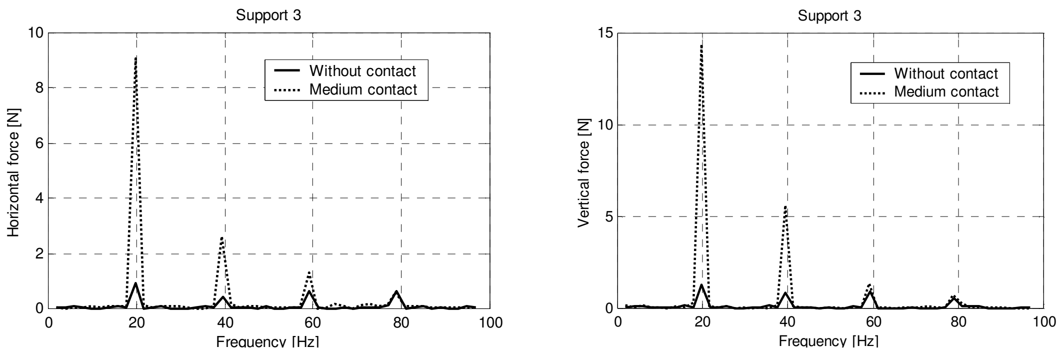

Figure 13 – Horizontal and vertical forces measured in support 3 during light rub

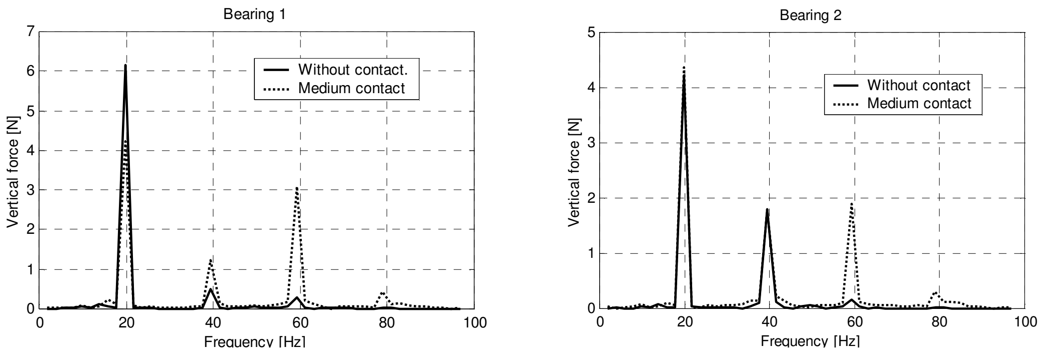

Figure 14 – Vertical forces measured at bearings 1 and 2 during light rub

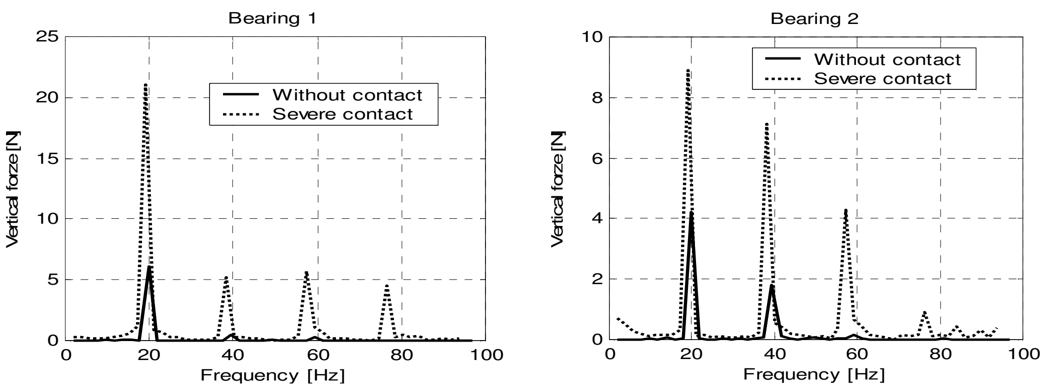

Figure 15 – Vertical forces measured at bearings 1 and 2 during severe rub

As previously specified also the rotational speed fluctuation has been measured with a two laser beam rotational vibrometer, which by time integration evaluates the amplitudes of angular oscillations with a sensitivity of 0.01 °/ V. The expected behaviour was that impact on the seal and friction during contact would create a measurable modification in the speed spectrum, and therefore also in the torsional oscillations. The measurement was taken at the external tail end of the shaft, and when the shaft was driven by the motor fed by the inverter all the expected frequency components in the speed were masked completely by the noise generated by the inverter. Therefore a test has been made with the motor directly connected to the net (therefore running at 3000 rpm), but surprisingly the residual noise in the torsional vibration spectrum (more or less "white noise" randomly distributed components with amplitudes of max. 25 mV) was slightly reduced by the rub, and the expected modification in 1x and higher harmonics was unrecognizable. The only effect of the rub was the excitation of one torsion resonance during the run down

CONCLUSIONS

The test rig as proposed is capable to simulate the rubbing phenomena which may occur in real rotating machinery. Since the test rig has from dynamic point of view a behaviour which is similar to full size rotating machines, also its sensitivity to contact forces during rubs can be assumed to be similar, considering the scaling factor. Therefore the analysis has been finalized to check all measurable modifications due to rub in the signals which are normally available in monitored machines (such as shaft displacements in bearings measured by proximity probes, oil pressures related to static and dynamic loads on bearings), and in the signals furnished by additional instrumentation which can easily be installed on the machine (such as accelerometers on the casing, microphones for measuring sound pressures and torsional vibrations measured by a rotational vibrometer). The goal of this analysis is to identify all possible symptoms of a rubbing rotor which could be measured on a real machine.

The main results are following:

1) higher harmonic components in proximity probe signals in bearings should be able to identify even light rubs
2) higher harmonic components in bearing oil pressure spectra should also be able to identify rubs
3) the increase of vibrations in the rubbing casing in the high frequency range, as well as the increase of sound pressure also in the high frequency range are clear symptoms of a rubbing condition.
4) The torsional vibrations seem to be rather unaffected by the partial rub: its modification resulted smaller than the noise in the signal.

Further investigations are required to analyse deeply all different conditions and related behaviour starting from light intermittent rub up to severe continuous rub, and considering different rotating speeds.

ACKNOWLEDGMENTS

The VICONDIA project of the ALFA II program which allowed the participation to the experimental tests of the of the authors, is gratefully acknowledged.

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