International Journal of Latest Engineering Research and Applications (IJLERA) ISSN: 2455-7137

Volume – 07, Issue – 03, March 2022, PP – 10-19

Steam turbine seal rub. History Case

B. Fraga De Cal[1], P. Fraga López[2]

[1] *Ph D. Mechanical Engineer. A Coruña University, Spain.*
Renewable Energy MSc. (Queen Mary University of London)
[2] *Ph D. Prof. Mechanical Engineer. A Coruña University, Spain*
Higher Polytechnic School. Department of industrial and naval engineering

Abstract: This history case describes how vibration data provided by an advanced online analysis system helped identify the severity of vibrations experienced by a ship's propulsion steam turbine. In their machine plant they have a very advanced monitoring, but as far as vibrations are concerned, they only have an alarm system for vibration amplitude.

A few months after its commissioning, after a scheduled shutdown for preventive maintenance, in which the optimal conditions of that steam turbine were rehabilitated, it began to present brief periods of high vibration. These high vibration peaks appeared little by little, more and more frequently and with greater amplitude. In August 2002, the observed vibration approached the turbine's safety firing limits. Considering the indicated measures, the operators and maintenance personnel decided to wait until the next arrival at port to turn off the turbine and face the solution of these problems. The turbine continued in operation for about a month, in which they continued to observe occasional periods of high vibration.

In October 2002, already in port, the turbine was dismantled and inspected. The findings indicated that the reason for the vibration spikes was due to the oil injected into the sealing shutter, which was carburized and caused a friction of the turbine shaft sporadically. The oil had seeped through the oil mazes of that shutter, which had worn unevenly due to misalignment.

Keywords: Vibrations, rotating systems, misalignment, rubs.

1. Introduction

The basic technical characteristics of the steam turbine are:[1]

- Shaft power, nominal/normal: 26100/24850 Kw
- Shaft speed, nominal/normal: 9250/9466 rpm
- Steam flow, nominal/normal: 24020/18660 kg/h
- Steam inlet pressure/temperature: 185 bar, 540º C
- Steam exhaust pressure/temperature: 30 bar, 210º C.

Photography 1. Steam turbine rotor

The turbine Westinghouse was designed to produce 35,000 shaft horsepower (26,000 kW), using steam provided by 2 Babcock & Wilcox Modified "D" Super-heated boilers, to reach the designed ship speed of 27 knots (50 km/h; 31 mph).

The drawing in Figure 1 shows a cross-section of the part where the anomaly occurred. The lubrication oil of the bearings is contained by the labyrinth of oil, which sprouted beyond the closing joints of that

labyrinth. Normally, this liquid oil flows into the drain indicated in Figure 1, and any other excess oil, already evaporated by the high temperature, is evacuated from the space between the oil and the labyrinth by the vacuum chamber separation system.[1]

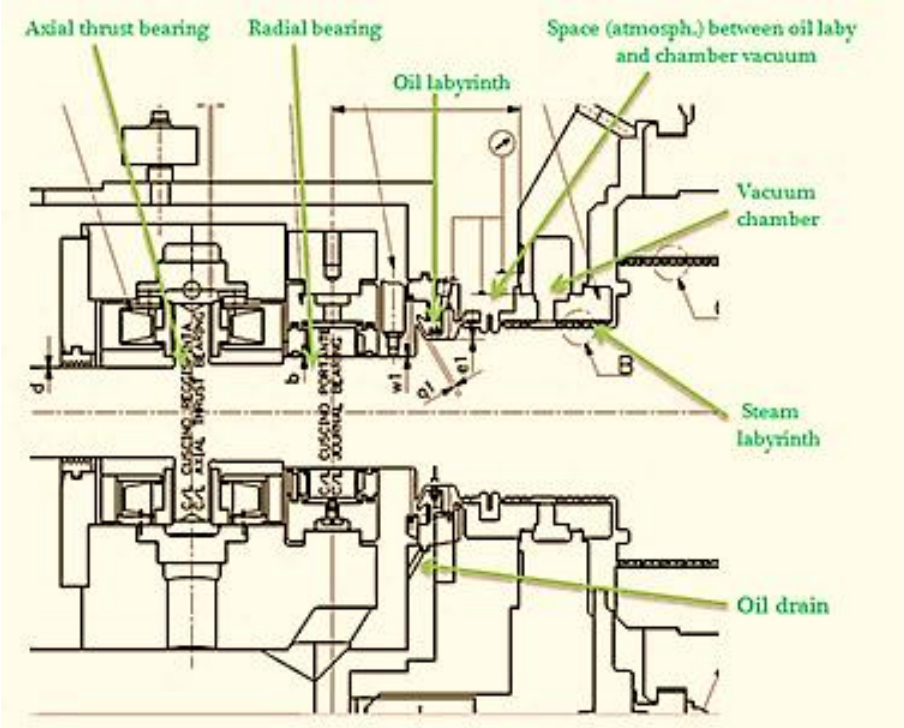

Figure 1. Turbine cross-section drawing.

Therefore, the vacuum chamber collects flow suction ducts from two directions: the steam that escapes from the joints of the steam labyrinth closure, and the evaporation, where appropriate, of the oil leaks from the joints of the steam labyrinth.

2. Method of Analysis and Instrumentation

The movement of the rotor can be measured as displacements, according to the Standard-670/2014 recommendation of A.P.I. (American Petroleum Institute), using proximity or induction transducers in horizontal and vertical position. The phase angle, i.e. the position of the rotor at the moment the sensor measures the maximum displacement of vibration, may also be measured by making a reference mark consisting of a longitudinal groove on the surface of the shaft, and by installing another displacement sensor in a transverse plane containing this mark, in such a way that each time the rotation passes the aforementioned reference, in front of the sensor, once each turn, there will be a change in the output signal of this transducer and a peak will appear in the signal, as can be seen in the scheme of Figure 2, whose response in voltage (mV) will be proportional to the displacement measured by the transducer, depending on the variation of the interlock between it and the shaft.

The movement of the rotor is generally measured, therefore, using two proximity transducers, positioned on the horizontal and vertical axis with a lag angle of 90º between them, in order to detect the displacement according to the two degrees of freedom of the axis, in x-y coordinates.

The signals of each of these transducers can be observed in the oscilloscope, as sinusoids out of phase 90º as a function of time, or orbits, if the output of the reference transducer (Kp, Kφ) is connected to the "z" axis of the cathode ray tube of an oscilloscope, where a bright spot and a discontinuity on the trace will occur, as shown in this Figure 2.

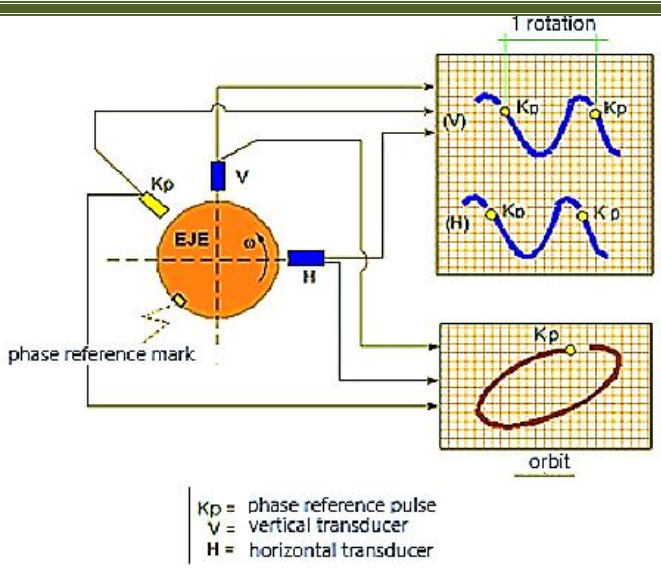

Figure 2. Transducer installation

The bright spot (Kφ) indicates the moment when the rotor groove passes in front of the reference sensor (Kp), and this shaft position is considered to be the zero angular position of the rotor. The distance, in a horizontal direction, between the bright spot and the next positive peak of the curve, indicates the phase angle of the maximum amplitude of the vibration.

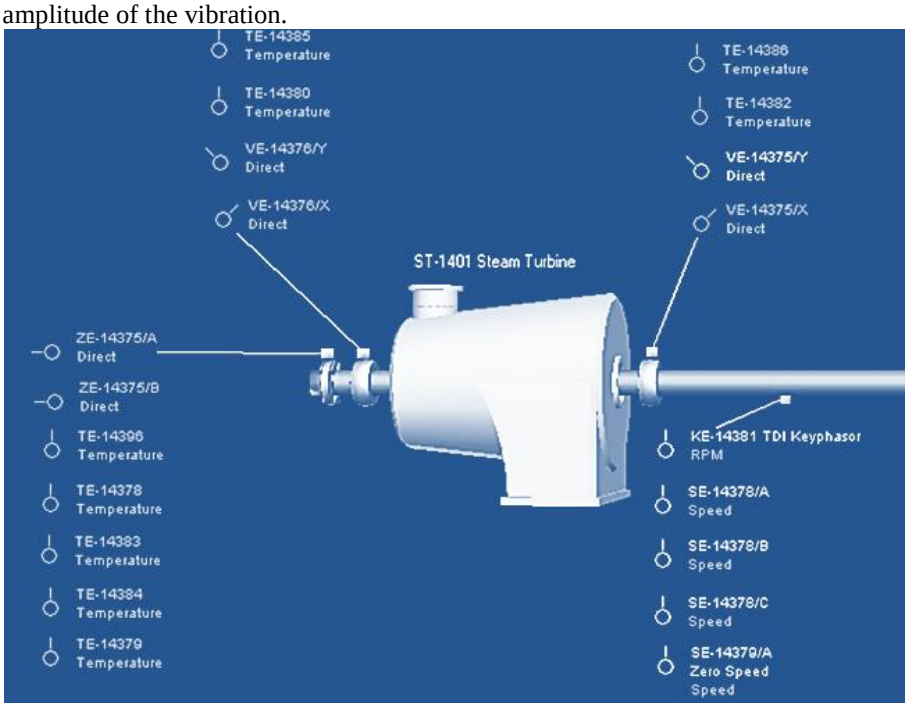

Figure 3. Instrumentation vibrations of the turbine communicates with the software system.

The Figure 3 shows a view of the monitoring and instrumentation scheme used for the steam turbine of this case analysed. In addition to monitoring temperature and speed, proximity and phase reference probes ("Keyphasor") are also installed. These non-contact proximity transducers measure the relative vibration of the turbine rotor inside its oil film bearings, according to the arrangement indicated in Figure 3, marked with the legend VE-Direct.

3. Sequence of Events

3.1. Trouble-Free Period

Since the initial start of navigation of the ship, approximately 3 months before the alarm of October 2002, the operation of the turbine was normal. Vibration levels were consistently low and no wide-ranging peaks were observed. Levels in the range of 5 to 15 microns were measured, which were fine and within the normal operating parameters of the machine. The alert alarm had been set at 65 microns and the danger alarm at 85 microns.

3.2. High Vibration Intermittency Observed.

From February 2002 to the end of August 2002, brief periods of high vibration were observed, with peak levels reaching 40 microns. On August 30 and 31, and again on September 2, 2002, the observed peaks rose in large numbers, reaching 75 microns, which was almost the maximum level of danger alarm, of 85 microns (Figure 4). A review of the vibration data revealed that 90% of the observed high vibration could be attributed to a 1X response (synchronous).

This Figure 4 shows occasional peaks of high vibration that occurred over a two-week period. The reference point of the alert alarm (57 microns) is indicated by a dashed line.

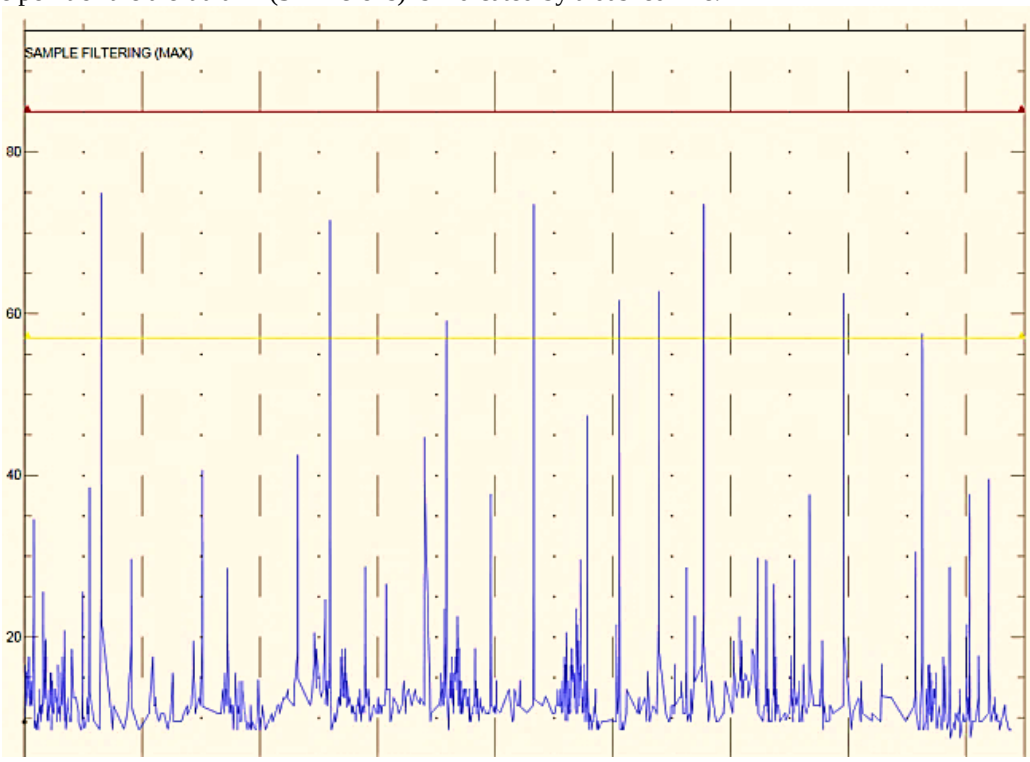

Figure 4. Vibration tendency.

A closer evaluation showed that brief recurrent periods of high vibration started from normal levels of 10 to 15 microns, quickly rose to such a high peak value of 50 to 75 microns, and then quickly fell back to normal levels. Each event lasted from about 5 to 15 minutes (Figure 5). The peaks appeared at intervals ranging from 8 hours to 3 days. During some of the high vibration events, operators who were near the turbine reported that they could feel and hear significant increases in vibration from the machine.

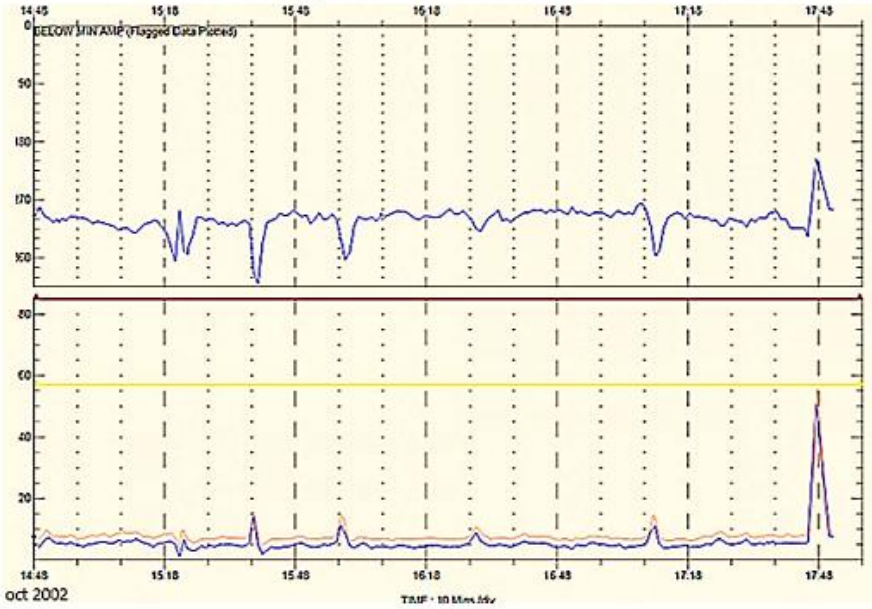

Figure 5. High vibration intervals.

Figure 6 and Figure 7 show the phase and amplitude of the vibration (both filtered and unfiltered-1X). The shape of the orbit shows the behaviour of a typical short period of high vibration. In this example, the vibration amplitude was increased from normal levels (8 microns) to a maximum value of 74 microns in about 4 minutes. The orbit in Figure 6 shows the global amplitude of broadband, while the orbit in Figure 7 indicates its already filtered shape at 1X response.

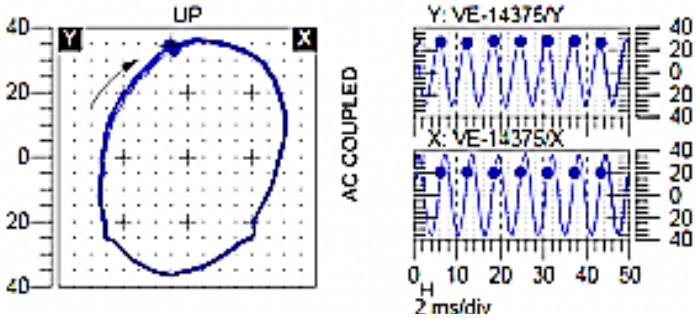

Figure 6. Unfiltered 1X response.

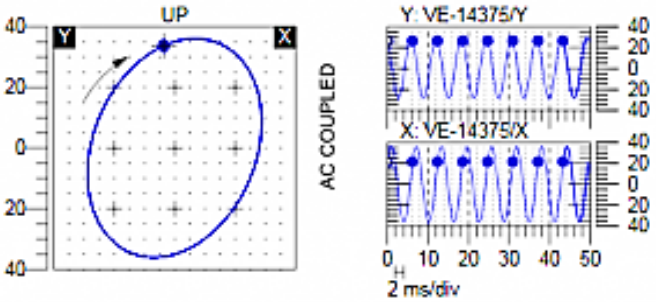

Figure 7. Filtered 1X response

4. Initial Research and Actions

Defects such as imbalance, misalignment and tightening/rubbing can often produce low frequency vibrations corresponding to the operating speed of the rotor (1X response). However, in the analysis of the high vibration of this turbine, these defects were ruled out, directly, because they would have caused a vibration

constantly, instead of the observed intermittent periods of high vibration. The shapes of the orbit were then revised (Figure 8).

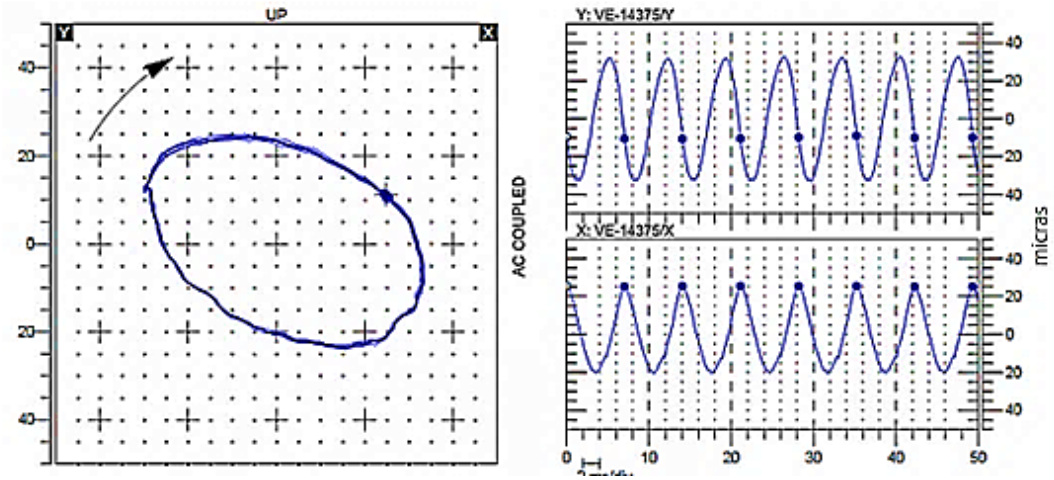

Figure 8. Orbit and amplitude/time diagram.

The shape of the orbits indicated the presence of an oscillation and change of situation of the phase reference mark, as indicated in Figure 9 and Figure 10. These orbits, during intermittent vibration peaks, therefore show unstable phase angles.

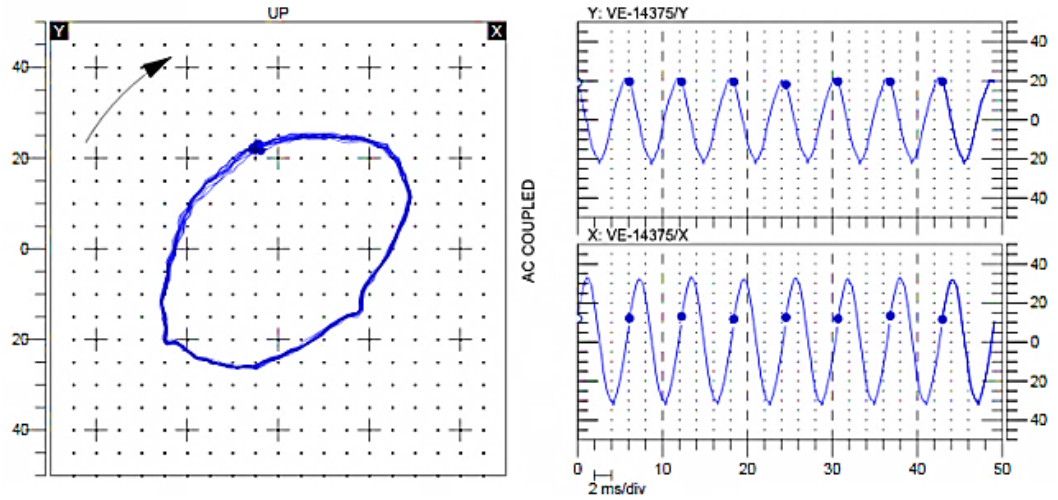

Figure 9. Orbit and amplitude/time diagram.

In order to minimize the possibility of damage caused by the continued operation, the following actions were decided:

1. Analyse and evaluate the lubricating oil of bearings and seals for possible replacement: The oil was analysed and found to be in good condition for use. There was no need for replacement.
2. Perform oil refining maintenance and cleaning: This routine maintenance was performed without major findings.
3. Check the oil pressure, upstream of the flow control holes: the oil pressure was checked and verified to be normal.
4. It was observed that the pressure in the vacuum chamber was slightly higher than that of the nominal characteristics. It was reduced to match the specification.

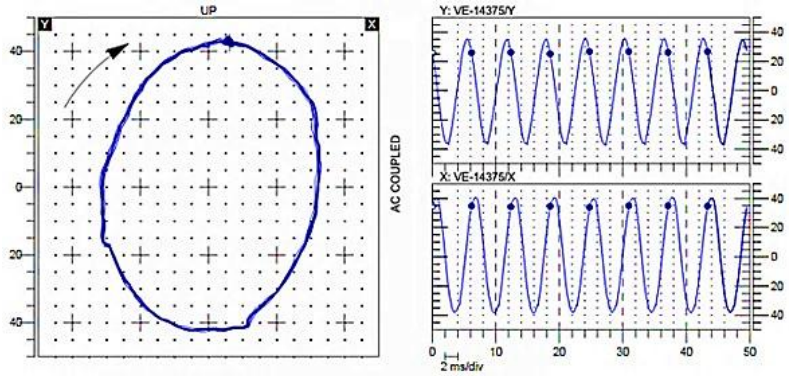

Figure 10. Orbit and amplitude/time diagram.

The intermittency of the signal of high vibration peaks made us suspect, in principle, of a friction since this phenomenon, when it is partial, causes this type of signals. It was therefore decided to carry out tests and collection of transient turbine start-up data, which resulted in:

- As the shaft speed increases, the amplitude of the lateral vibration also increases.
- The amplitudes of the 1X synchronous vibrations of the shaft, in the section next to the left support, in the steam inlet, are higher and almost constant.

Responses such as those in Figure 11 and Figure 12 were obtained. In Figure 11, it is observed that the resonance frequency has increased with respect to the original, in addition to the fact that the amplitude of that 1X response, once the resonance region is exceeded, increases and presents an almost constant amplitude and phase angle.

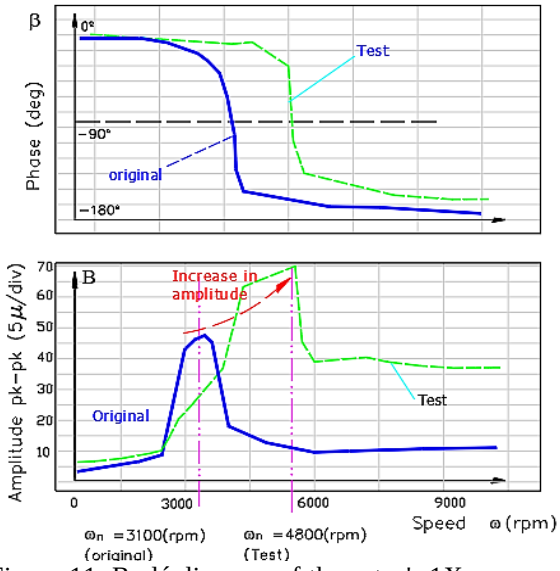

Figure 11. Bodé diagram of the rotor's 1X response.

In Figure 12, it is also observed that the resonance frequency has increased with respect to the original, in addition to the fact that the amplitude of that 1X response, once the resonance region is exceeded, increases and presents an almost constant amplitude, during a certain period of time.

These symptoms, in addition to the intermittency of high vibration, can constitute a partial rubbing, with great probability.

Orbit shapes during intermittent vibration peaks show shifting phase angles, a classic indication of friction caused by thermal arc with a transient hot spot.

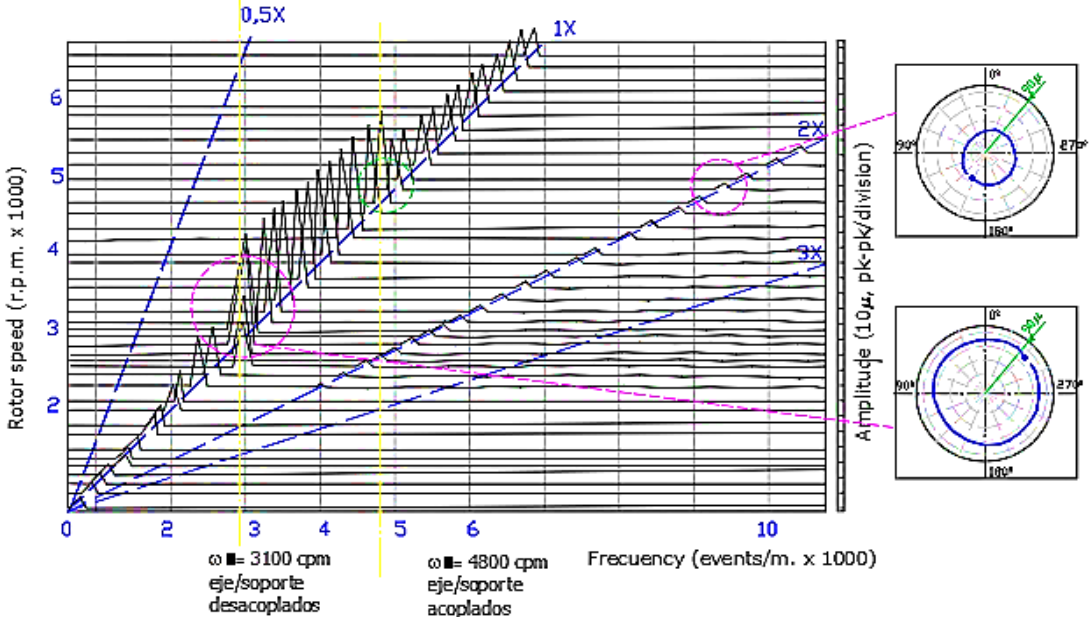

Figure 12. Cascade and orbit spectrum.

In this spectrum of Figure 12, which corresponds to the start of the machine, it is observed how when the resonance speed is reached, the amplitude of the synchronous response 1X increases, that is, the movement and displacement of the shaft becomes greater. This situation could be related to the shaft initiating a friction with the mechanical support or seal of seal, and not a bearing, since the defects in bearings provide vibration signals at very high frequencies, which did not occur in this case.

The results experienced and related to synchronous forced vibration, 1X response, can be summarized as follows, for partial rubbing:

- When the shaft comes into contact with the bracket, the latter restricts the movement of the rotor which acts as a "third bearing". The rotor increases its natural frequency, and new natural frequencies appear in the spectrum, as the shaft-coupled support contributes to the flexibility and/or stiffness of the system. In this case, it increases its dynamic stiffness.
- The dynamic response of the coupled shaft/support system depends largely on the strike between both elements. The support introduces a geometric nonlinearity to the system. The high amplitudes are maintained in a significant spectrum of the rotational speed, for the duration of that friction, and correspond to the difference between the system of decoupled and coupled natural frequencies (Figure 12).
- The maximum actual amplitude depends on the radial clearance of the shaft/bracket and the sealing damping characteristics. The strike also affects the period of rotation, for which shaft and support remain in contact.
- The response of the rotor depends on whether the rotational speed is increased or decreased, and if there is an absence of external impulses to drive the rotor out of the initial almost stabilized regime. The range of the rotational speed, while the shaft is kept in contact with the support, begins and ends for different values of that speed.

The resonance frequency of the shaft/support or seal, coupled (^{cou}), is given by the equation:

$$\omega_i^{n(cou)} \sim (-1)^i \sqrt{(K_2 + K_3)M} \quad ; \quad \omega_{i+2}^{n(cou)} \sim (-1)^i \sqrt{(K_1 + K_2 + K_s)/M_s} \quad ; \quad i = 1, 2$$

M = rotor mass; K_1, K_2, K_3 are coefficients of rotor stiffness, determined by the longitudinal location of the support; Ms, Ks: are the mass and rigidity of the support/seal. Contrasted with the decoupled shaft/support/seal system (a), there is an increase in the value of that natural frequency. The "mechanical seal", therefore, behaves as "a new bearing" added, and increases the stiffness of the system as the most prominent factor:

$$\omega_2^{n\,(un)} = (-1)^i \sqrt{K/M} < \omega_2^{n\,(cou)} = (-1)^i \sqrt{(K_2 + K_3)M}$$

5. Turbine. Maintenance and Findings

In November 2002, the steam turbine was dismantled for general inspection during a scheduled shutdown.

5.1. Inspection Results

It was found during the inspection, once the turbine was dismantled:

- Carbonized lubricating oil, which had accumulated in the vacuum chamber of the seal, and which had come into contact with the turbine rotor (Figure 13). Confirms partial rubbing.
- Unilateral wear of the oil labyrinth rings was discovered (Figure 14). It can mean a misalignment of the shaft.
- No significant damage had occurred to the rotor shaft, steam mazes or bearings.

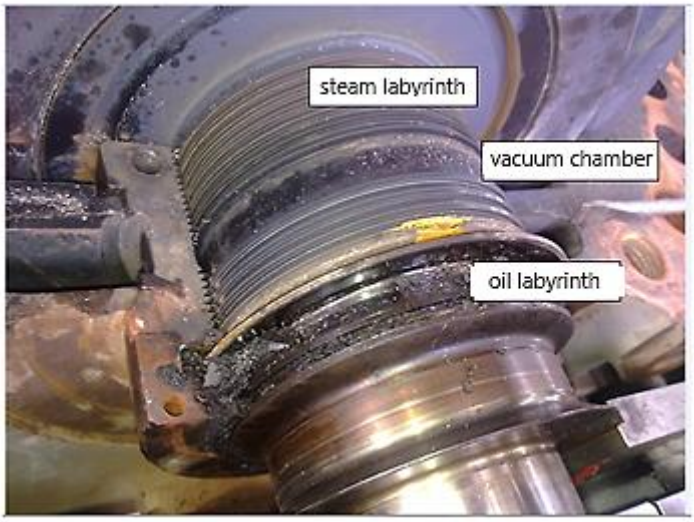

Figure 13. Charred oil, visible in the vacuum chamber, immediately after the closing of the labyrinth.

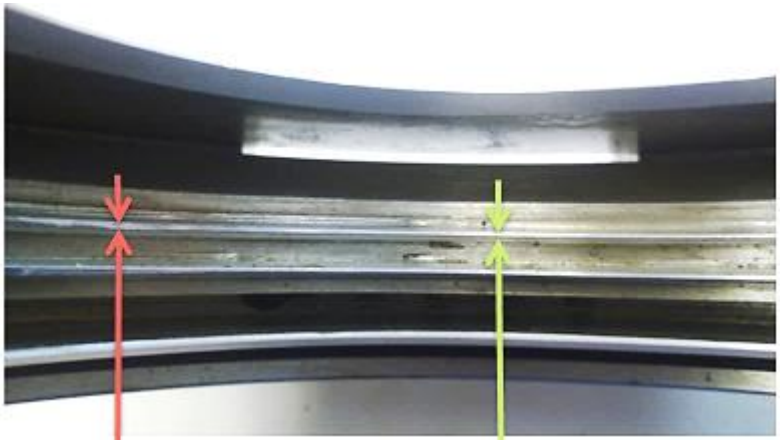

Figure 14. This photo shows uneven (one-sided) wear of oil labyrinth rings

6. Conclusions and Future Actions

With the checks made, the diagnosis of this high vibration of intermittent character was found:

- A high misalignment between the turbine supports, up to 1 mm. Misalignment caused uneven wear on one side of the oil maze rings or seal, causing a partial friction of the shaft on the seal.
- Excess oil seeped into the space between the oil rings and the vacuum chamber.
- Oil vapor mixed with high-temperature steam was introduced into the vacuum chamber, immediately after the labyrinth closure.
- The hot oil formed a hard charred "residue", which caused a partial friction in that sector of the seal.

Within a few minutes, the coal deposits were polished until the problem disappeared, but the oil leak would form more coke and the cycle could be repeated over time.

In order to avoid, then, the repetition of a similar incident in the future, the misalignment of the turbine was investigated, in response to that uneven wear of the labyrinth closure that was reflected in Figure 14.

The turbine had been properly aligned at the ship's previous scheduled shutdown, and was installed in its already closed and assembled foundation. No apparent causes of the subsequent misalignment were found, but the turbine manufacturer reported that, despite the stiffness of the foundation, mismatches similar to the one analysed can sometimes occur during the first months of the operation of a new or regenerated turbine. To avoid similar incidents in the future, it is considered necessary to carry out the hot alignment of the equipment, after 3 to 6 months of operation.

Misalignment is often imperceptible in the commissioning of many machines, and it can be said that it constitutes a "latent" defect that over time can have an impact on very serious breakdowns.

7. Bibliografía

[1]. J. Garcia Mesa y P. Fraga, «Informe avería Turbina de vapor,» Navantia, Ferrol (La Coruña), 2002.

[2]. L. Tam, A. Przekwas, R. Hendricks y M. Braun, «Numerical and Analytical Study of Fluid Dynamic Forces in Seals and Bearings.,» *Journal of Vibration, Acoustics, Stress, and Reliability en Design. Trans of the ASME.*, vol. 110, nº 3, pp. 315-325, 1988.

[3]. F. Sorge, «Preventing the oil film instability in rotor-dynamics,» *Journal of Physics. Conference Series 744(1):012153*, 2016.

[4]. E. Jang, Y. Park, C. Kim y A. Muszyinska, «Identification of the quadrature resonances using modal nonsynchronous perturbation testing and dynamic stiffness approach for an anisotropic rotor system with fluid interaction,» *International Journal of Rotating Machinery*, vol. 2, nº 3, pp. 187-199, 1996.

[5]. A. Muszynska, «Whirl and whip-Rotor/bearing stability problems,» *Journal of Sound and Vibration*, vol. 110, nº 3, pp. 443-462, 1986.

[6]. L. Tam, A. Przekwas y R. Hendricks, «Numerical and Analytical Study of Fluid Dynamic Forces in Sealsand Bearing,» *Rotating Machinery Dynamics. ASME*, nº ASME Publ. H0400B, 1987.

[7]. Eungu, Park, Ypoung-Pil, Kim y A. Muszynska, «Identification of the quadrature resonances using modal nonsynchronous perturbation testing and dynamic stiffness approach for an anisotropic rotor system with fluid interaction,» *International Journal of Roating Machinery*, vol. 2, nº 3, pp. 187-199, 1996.

[8]. B. Fraga, «Study of stability margin ans its connetion with the dynamic stiffness applied to a tidal turbine,» *International Journal of Latest Engineering Research and Applications (IJLERA)*, vol. 04, nº 01, 2019.

[9]. Y. Wang, X. Ren, X. Li y G. Chunwei, «Numerical investigation of subsynchronous vibration in floating ring bearings,» *Proceedings of the Institution of Mechanical Engineers Part J. Journal of Engineering Tribology 1994-1996*, Vols. %1 de %2208-210, 2018.

[10]. R. Hendricks y A. Muszynska, «Turbomachine sealing and secondary flows, part 2 – review of rotordynamics issues in inherently unsteady flow systems with small clearances,» de *Proceedings if the Second International Symposium on Stability Control of Rotating Machinery (ISCORMA-2)*, Gdansk, Poland, 2003.

[11]. P. Goldman y A. Muszynska, «Application of full spectrum to rotating machinery diagnostics,» *Orbit BNC*, vol. 20, nº 1, 1999.

[12]. A. Muszynska, «Vibrational diagnostics of rotating machinery,» vol. I, nº 3-4, 1995.

[13]. P. Goldman, J. Yu y D. Bently, «Full Annular Rub in Mechanical Seals, Part.1.,» *International Journal of Rotating Machinery*, nº April, pp. 319-328, pp. 26-32, 2002.

[14]. C. Warda, «Effect of ring misaligment on the fatigue life of the radial cylindrical roller bearing,» vol. 111, nº March, pp 1-11.

[15]. S. Li, W. Zhang y B. Hou and Guo, «Research on dynamic stiffness of vibration,» Binggong Xuebao, 2017.

[16]. B. Fraga De Cal, «The importance of fault prediction in a tidal turbine. Misalignment and cracks in the shaft.,» Edimburgo, 2018.

