Misalignment cases on machine trains of CC Power Plants

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Abstract

This paper discusses the vibration patterns found in some real cases of machinery external misalignment belonging to the Power Generation industry. It does not address internal alignment, which is the alignment of the stationary components (diaphragms, oil deflectors, etc.) relative to the rotors The machines analyzed are the turbine-generator trains found in the Combined Cycles Power plants; the article focuses on the single shaft, 8 bearings machine train layout, which is composed by a Heavy Duty Gas turbine, two bodies of Steam turbine (HP/IP and LP) and the Electric Generator. The alignment of this type of machine trains can be difficult due to the big dimensions and masses involved, moreover the relative position of the machines is strongly affected by the significant thermal growth which typically occurs on steam and gas turbines. These main features, together with the fact that rigid couplings are used, introduce a high level of complexity which must be faced by diagnostic engineers in order to define the alignment condition of this type of units – the cases presented in this article are meant to be a good example of this complexity.

1 Introduction

The main purpose of the alignment process is to determine the relative alignment of the rotors at installation such that the bearing loads and rotor stress will be optimized in the hot alignment condition and within acceptable levels for any transient alignment conditions. Ideally, but not necessarily, this is done by removing the shear and bending moment interaction between the rotors in the hot alignment condition as much as possible. In this condition, the coupled rotor bearing loads will be equal to the uncoupled bearing loads.

The goal of shaft alignment is to have two or more shafts collinear, rotating around a common centerline of rotation. Coupling alignment is equivalent when it is assumed the coupling bore is centered to the shaft. External misalignment between two machines occurs when, operating at thermal equilibrium, the shaft centerlines are not collinear at the coupling and/or the shafts do not operate in their design axial positions within each machine [1].

Shaft alignment can be defined in a 3-dimensional perspective of how two machines are positioned relative to each another. For simplification, we break the alignment into their respective planes:

- Vertical: offset and angularity;
- Horizontal: offset and angularity;
- Axial alignment (Z).

In Figure 1, the machine on the left is depicted as the fixed machine and the movable machine's shaft line is plotted with respect to the fixed machine. Shaft misalignment is the deviation of relative shaft position along a collinear axis of rotation measured at two points of power transmission when equipment is running under normal operating conditions.

Misalignment can have different effects on different machine types. Some machines may not display any symptoms of misalignment until there is fatigue or failure of the coupling. This is because the flex planes might be absorbing the energy and taking the articulation of misalignment. In other cases, the shafts and bearings may absorb or reflect the results of misalignment in the forms of vibration and bearing temperatures changes. High vibration is not always an indicator of misalignment.

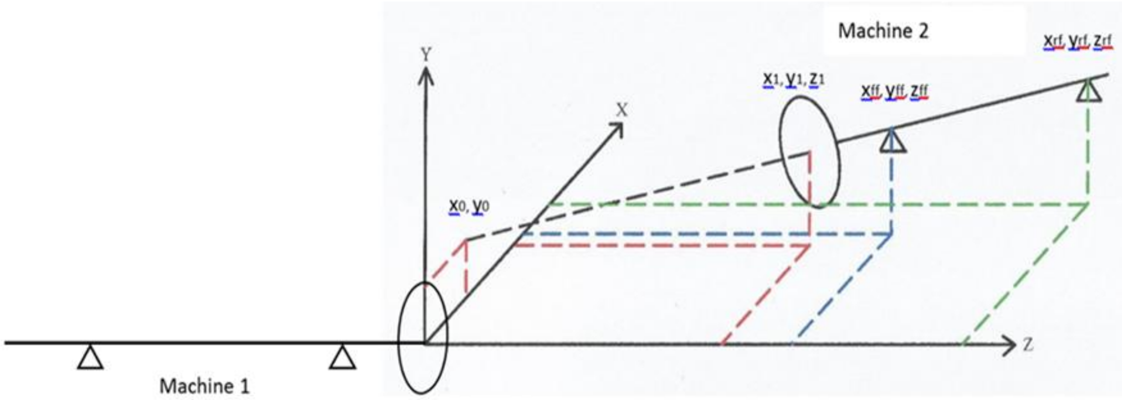

Figure 1. External misalignment scheme

- Bearing and coupling types will dictate the force transmitted to bearings and stationary components.
- Chronic broken coupling components or even cracked shafts may be an indicator of misalignment.
- Vibration can be higher or lower in radial and axial planes.
- Bearing metal temperatures are typically affected.
- A cold coupling alignment check is needed to fully evaluate the static alignment condition, but this may not quantify the dynamic alignment condition.

The detection of misalignment of turbomachines relies upon four different indicators:

- Shaft relative vibration (patterns and amplitudes);
- Shaft centerline position;
- Bearing metal temperature;
- Casing vibration (radial and axial).

In general, misalignment acts as a strong preload and forces the shaft to an abnormal position in the bearing (more loaded or unloaded, depending on the bearing of the machine train). Typically, the Shaft Centerline (SCL) Plot is examined during the start-up (SU) or shut-down (SD), to compare the real centerline path with respect to the theoretical one (this information is easy to find in the technical literature - [1] and [2]).

Moreover, this preload will typically restrain the shaft relative vibration in the direction of the preload itself, causing flattened orbits and in some case introducing a 2X component (which can produce the "banana shape" or "eight shape" orbits in some specific conditions). When a preload acts on the shaft this one will rotate in the range of deflection where the stiffness of supports behaves nonlinearly. The region of deflection is determined by the specific region of the radial preload. The rotor response is not the same in vertical and horizontal directions. The stiffness non linearity causes not only changes in the 1X response amplitude but is also responsible for the generation of higher harmonics. In that case the main vibration excitation is unbalance. The preload and the nonlinearity modify the response to this unbalance.

Another important aspect to consider is that the restriction of the shaft vibration, together with the extreme shaft position, will make the energy transfer from the shaft to the casing more efficient, therefore an increase of casing vibration is expected.

Finally, heavily loaded and unloaded bearings will cause respectively higher and lower than normal bearing metal temperature.

With respect to the combined cycle single shaft machine trains, which are the object of this article, it is important to mention that the couplings used on these trains are almost always rigid couplings, therefore there is no chance to absorb part of the misalignment in the coupling itself, as instead happens with flexible couplings.

Moreover, these machine trains are difficult to align, as they require the movement of big masses, and the alignment in series of three shafts – this is a very time consuming procedure and sometimes during the outages a slightly misaligned condition is preferred to a very time consuming alignment within tolerance.

Finally, gas and steam turbines are machines subjected to a strong thermal transient, which affects the relative position and alignment. Therefore the specifications for cold alignment take into account an estimated growth of pedestals and machine casing. Nevertheless, sometimes the real growth is different from the expected one (for example, there can exist problems of structural looseness or weakness due to cracks, wrong mounting, incorrect anchoring, etc.) therefore the hot condition may be misaligned even if the machines are aligned within tolerances in the cold condition.

All the above points introduce a high level of complexity which must be faced when diagnostic engineers have to define the alignment condition of these type of units – the cases presented in this article are meant to be a good example of this complexity.

The machine train layout which will be analyzed in the present article is shown in Figure 2; as already mentioned, it is composed by a Gas Turbine, two bodies of Steam Turbine (HP/IP and LP) and an Electric Generator, all coupled in a single shaft train. This machine train rotates at 3000 rpm and produces 450 MW at base load.

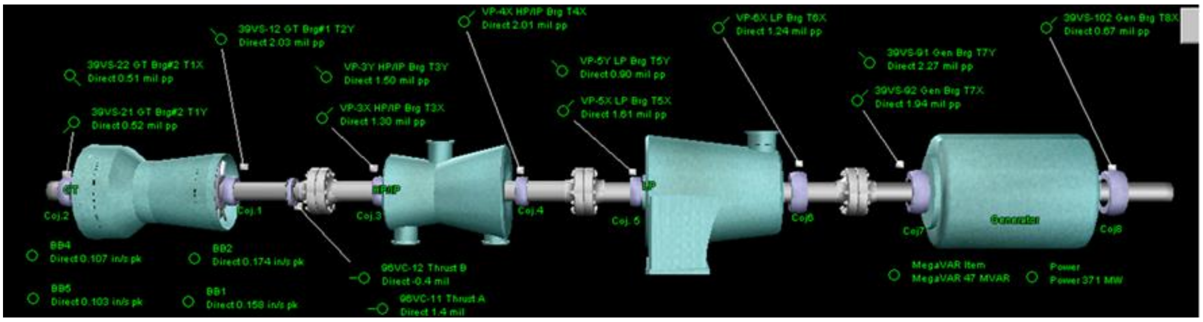

Figure 2. Single shaft machine train, including from left to right: GT (Brgs 1 and 2) - coupling A - ST HP/IP (Brgs 3 and 4) - coupling B - ST LP (Brgs 5 and 6) – coupling C - Electric Generator (Brgs 7 and 8)

2 Real cases

2.1 Typical case of external misalignment detection

This first case treats a misalignment between ST HP/IP and ST LP (coupling B). It is a good example of how to apply the indicators above mentioned to detect a typical case of misalignment in turbomachinery.

After a major outage of the unit, since the first SU's it was observed that the metal temperature at Bearing 4 (ST HP/IP) was significantly high (up to 107 ºC). This observation triggered a specific analysis to understand if this could be due to a misaligned condition.

The SCL plots of the SU (from turning gear speed up to nominal speed) was analyzed; this plot, configured with the real diametrical clearances, revealed that actually Bearing 4 was more loaded than normal, as the position of the shaft at the end of the SU was significantly lower than expected for a tilting pad bearing [2] (Figure 3). This information indicates a possible misalignment between ST HP/IP and ST LP, but other aspects should be taken into consideration to diagnose external misalignment.

In fact, external misalignment can be confirmed when a heavily loaded bearing is associated with a lightly loaded bearing [1]. For this reason, the SCL plot of Bearing 5 was analyzed. This bearing is an elliptical bearing, and the final position of the shaft at the end of the start-up appeared to be too high, confirming that this bearing was abnormally unloaded.

Interesting enough, the orbit patterns totally reflect this situation; at Bearing 4, the compensated orbit was very flattened (Figure 4), and its major axis almost horizontal, confirming the existence of a radial preload acting in the vertical direction. At Bearing 5 instead was observed a circular orbit pattern (Figure 5), which is indicative of a lightly loaded bearing; normally in fact the elliptical bearings exhibit a quite elliptical orbit when are properly loaded.

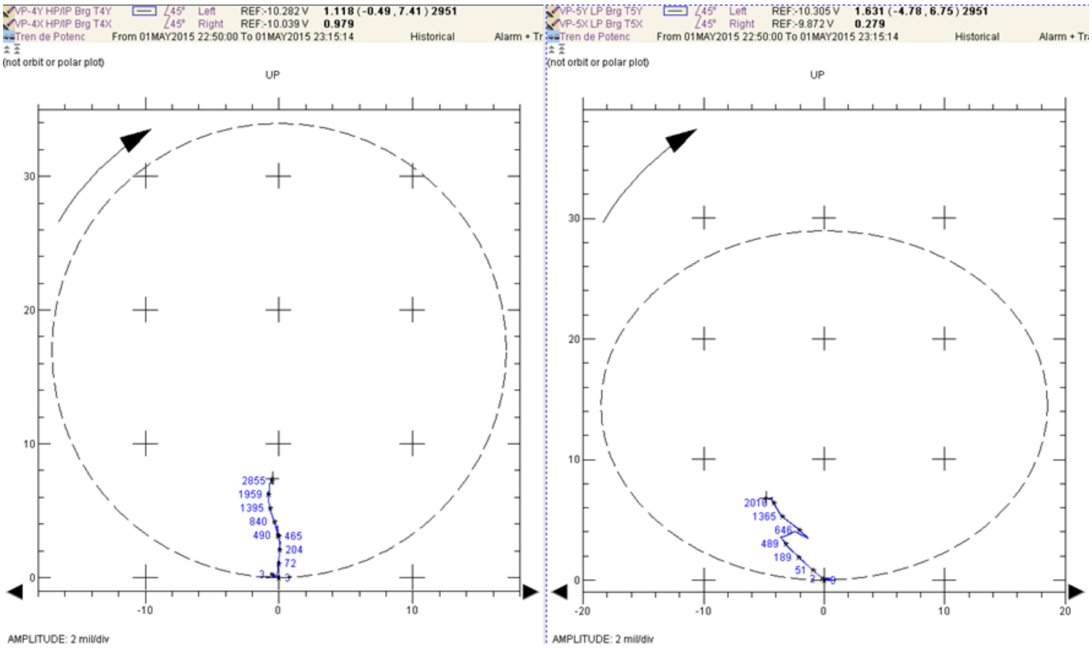

Figure 3. SCL plot measured at Bearings 4 and 5 during the SU

Unfortunately, these machines are not equipped with permanent accelerometers on the casing, therefore the absolute casing vibration could not be analyzed. Nevertheless all the others indicator (shaft position, shaft vibration and bearing temperature) were clearly indicating an external misalignment condition.

The maintenance team was asked to check the alignment between ST HP/IP and ST LP when possible. During the first available stop, the alignment condition was measured, and it was found a strong misalignment, as expected. Despite of the evidence, there was no time to improve the alignment condition, as the downtime of the unit had to be limited. As already mentioned, the alignment of these single shaft trains require several days, and the plant personnel took the decision to maintain the unit in a misaligned condition.

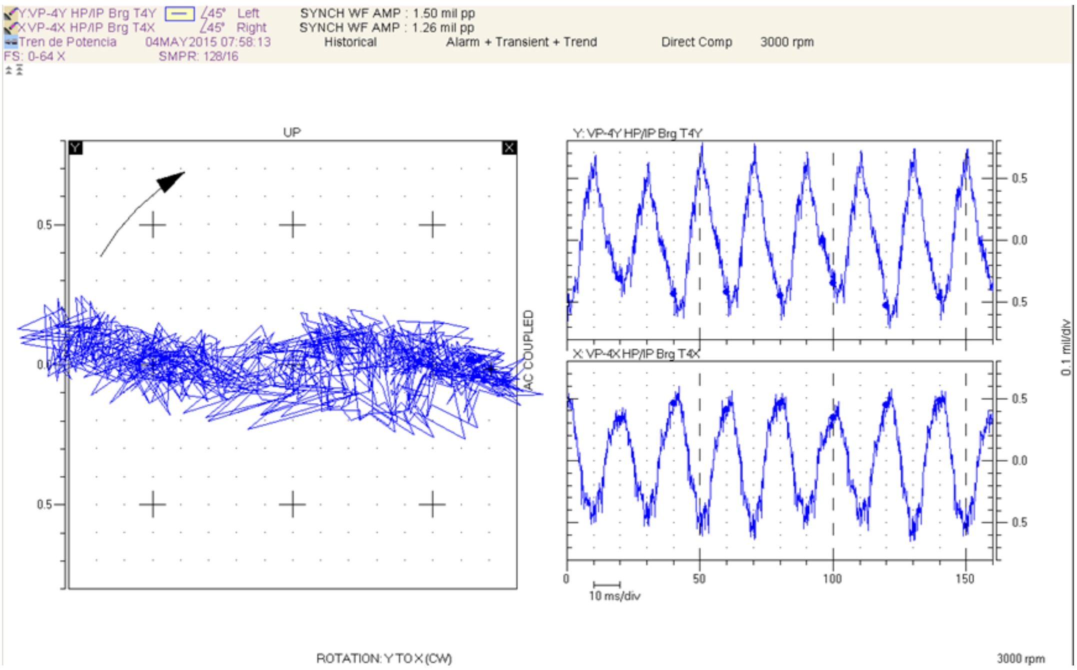

Figure 4. Direct compensated orbit at Bearing 4 on Load

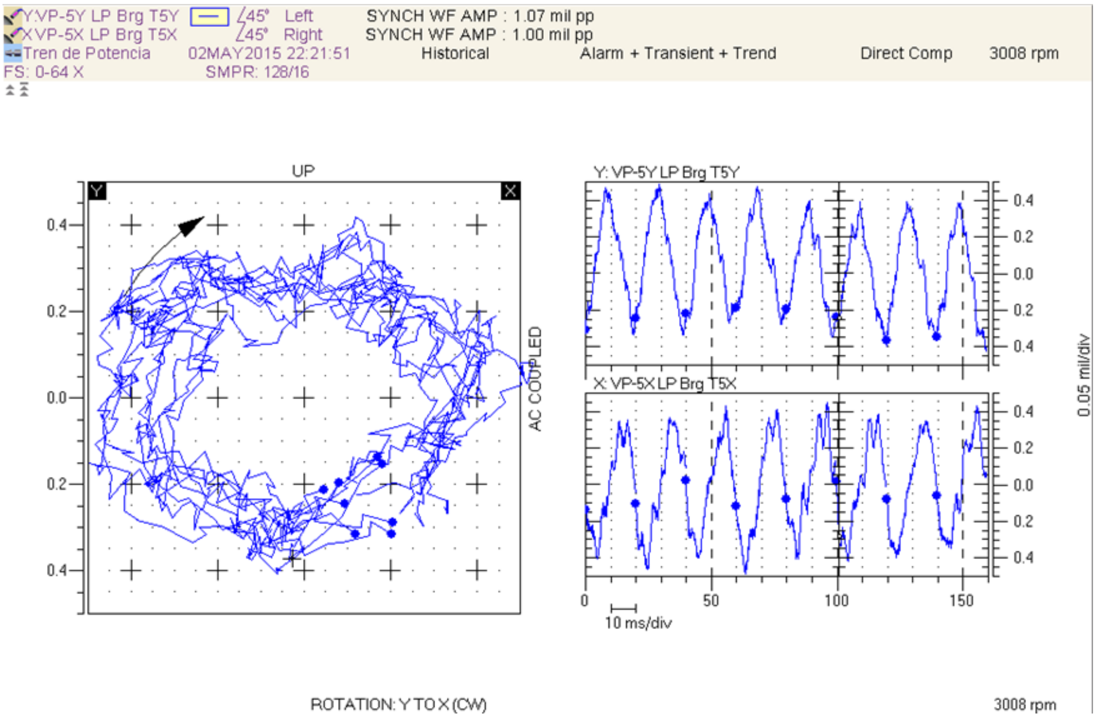

Figure 5. Direct compensated orbit at Bearing 5 on Load

2.2 Misalignment in hot condition leading to bearing damage

This second case is related to a misalignment between GT and ST HP/IP, belonging to the same type of machine train seen in the previous case, but located in another power plant.

After the overhaul of the unit, the orbit pattern at Bearing 3 (ST HP/IP) became very flattened when on load. There are several reasons why an orbit can be flat: high preload (due to external or internal misalignment), split resonance, mix mode, important anisotropy for example.

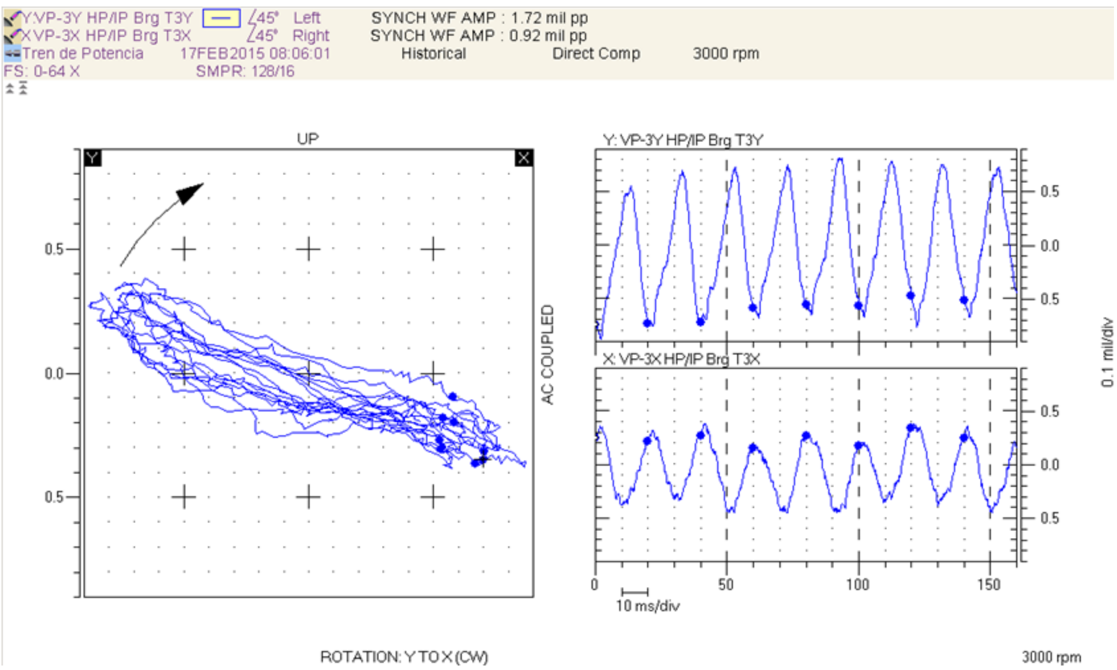

Figure 6. Direct compensated orbit measured on load at Bearing 3, showing a flattened shape.

To rule out some of the mentioned possible causes, it is important to observe that, just at the end of the speed increase (at 3000 rpm), the orbit pattern was rather circular. Then, during the thermal transient (which normally takes about 6-8 hours) the orbit pattern used to become more flattened, and even show an “eight shape” resulting finally in a very flattened shape after the thermal transient, as mentioned above (Figure 7).

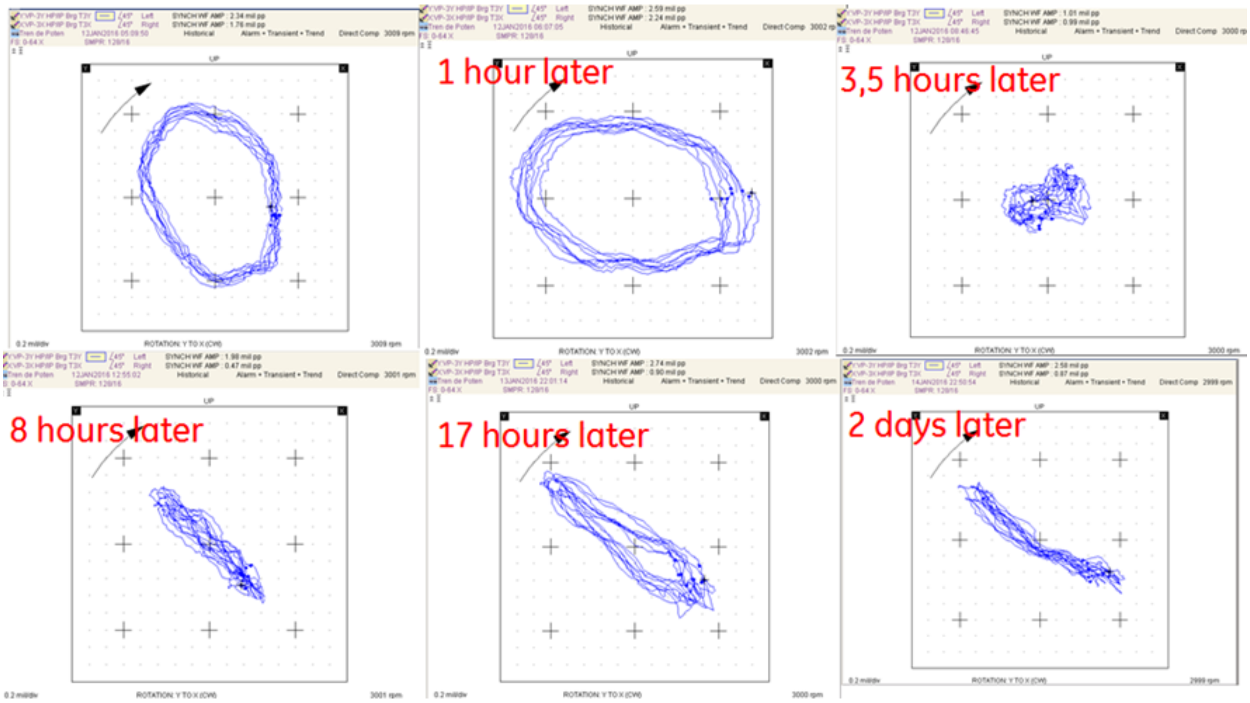

Figure 7. Evolution of the orbit shape at Bearing 3 from no load condition until the end of the thermal transient.

For this reason, it was suspected that the unit was affected by a problem of misalignment between GT and ST, magnified by the effects of the thermal gradient, which was causing their relative position to be more offset than in the cold condition, and therefore increasing the preload on Bearing 3.

To further investigate this hypothesis, the Shaft Centerline (SCL) Plot, configured with the real diametrical clearances, was observed. It was plotted the SCL path for the coupled Bearings 2 and 3 during a start-up (in blue in Figure 8) overlaid with the path during a shut-down (in orange). The position of the SCL at the end of the start-up can be considered normal taking into account these are both tilting pad bearings [2].

On the other hand the comparison between SU and SD gave a hint of the effect of the thermal transient on the SCL position. The plot suggested that Bearing 2 becomes more unloaded after the thermal transient, and in fact the bearing metal temperature used to decrease from 95º to 90º in few hours, remaining then steady.

The analysis for Bearing 3 is instead more complicated. The curves seem to suggest that also this bearing was becoming more unloaded, which does not really make sense, as in case of high preload between bearings, an unloaded bearing is always associated to a loaded one [1].

The doubt was solved looking at the SCL path of the shut-down in low speed range, from 600 rpm up to turning gear speed. In this speed range, the SCL path is highly influenced by the temperature variations affecting the bearing. The proximity probes are mounted on the bearing and this introduces a source of error in the SCL position measurements all the times a strong temperature gradient is affecting the bearing.

One good way to avoid this unwanted effect is to compensate the SCL of the shut-down curve at a speed which is not yet affected by temperature (of course the SU curve must be compensated at that same speed to compare). This trick reveals that actually after the thermal transient Bearing 3 becomes more loaded, as the SCL goes downwards and toward the left. The left movement can be quantified in around 5 mils (Figure 9).

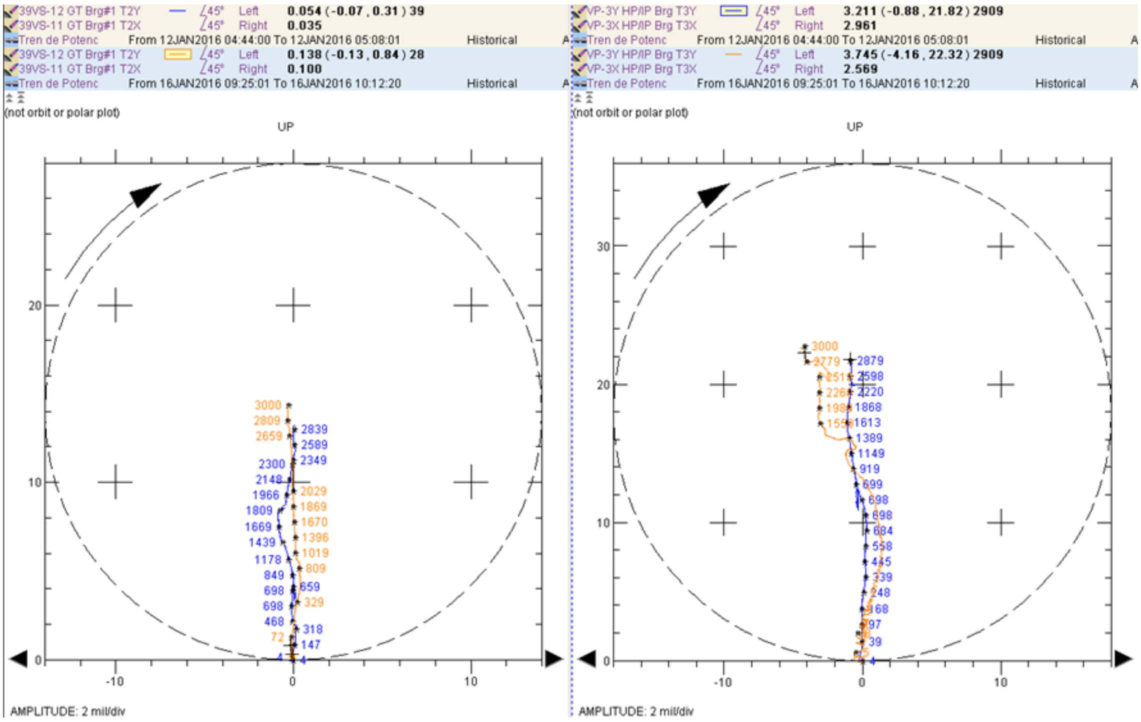

Figure 8. SCL plot measured at Bearings 2 and 3 during the SU (in blue) and during the SD (in orange). Compensation at slow roll speed.

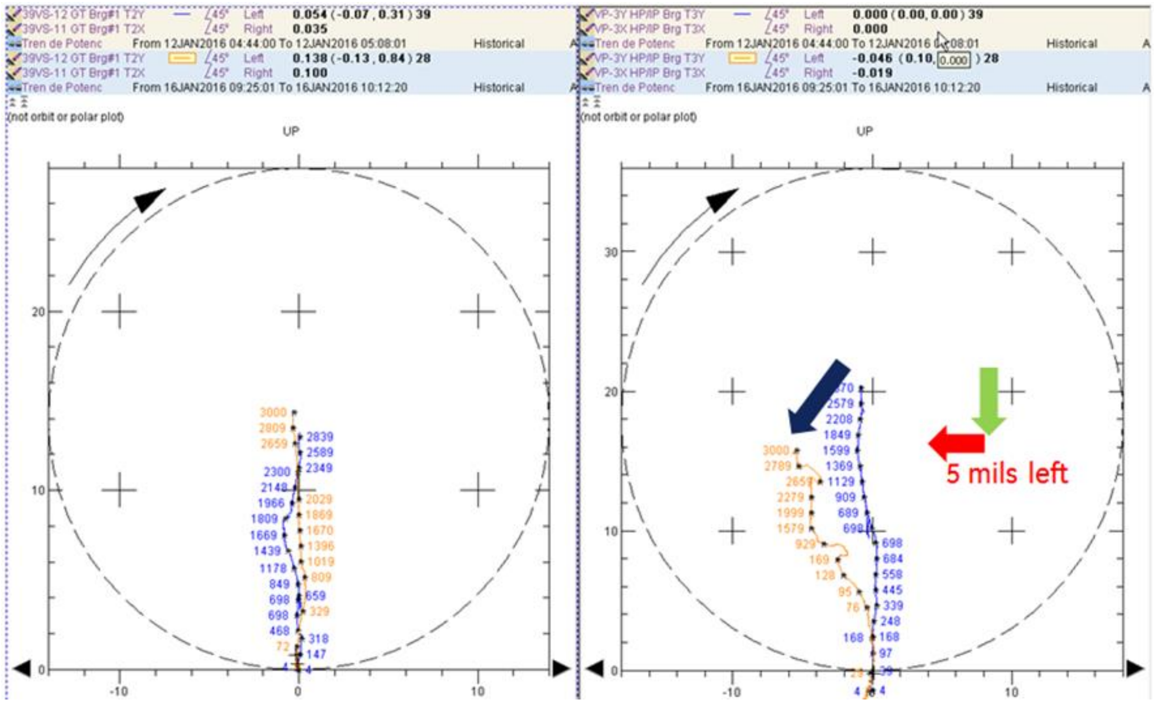

Figure 9. SCL plot measured at Bearings 2 and 3 during the SU (in blue) and during the SD (in orange). Compensated to avoid the effects of temperature on position measurement.

Overlaying the SCL and the orbit, the complete picture can be obtained (Figure 10). At FSNL, the shaft operates in the center of the bearing (slightly unloaded) and the orbit has a rather circular orbit. After the thermal transient instead, the increase of load with a strong component in the left direction causes a restriction of the shaft vibration in the same direction, and therefore the mentioned flattened pattern. It is important to observe that at Bearing 3 the metal temperature is not indicative of any bearing load change, as remains around 80 ºC during the whole cycle (SU+ thermal transient + load condition). A possible explanation of this phenomenon is that the temperature transducer is not located in the load area of the bearing.

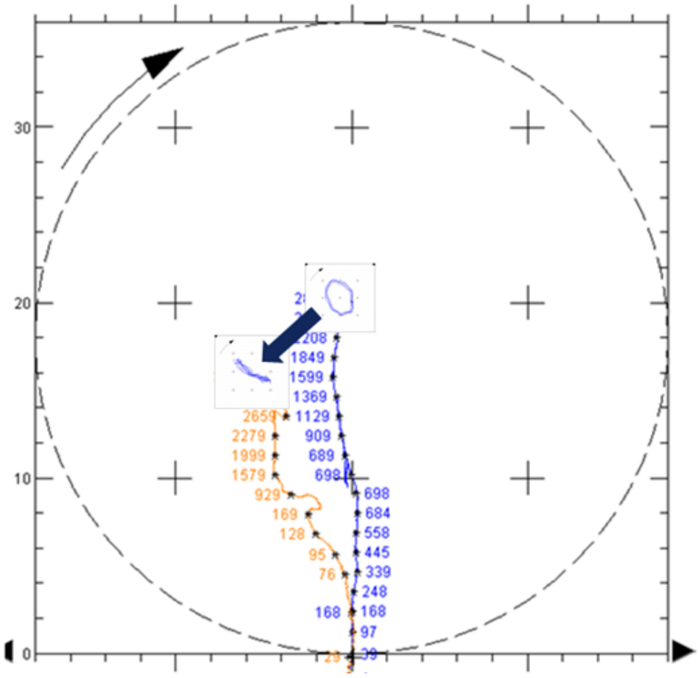

Figure 10. The complete picture: shaft position and dynamic movement at Bearing 3 before (blue) and after (orange) the thermal transient.

A comparison with the behavior of this bearing before the overhaul, revealed that the progressive loading (=downward movement) of the bearing was also observed in the past, nevertheless the movement towards the left direction became significantly bigger after the overhaul (it went from around 1,5 mils to around 5 mils). The data measured on a sister unit also confirmed that a leftward movement of about 1-1,5 mils is normal for this bearing.

All this led to the conclusion that the leftward movement was indicative of a misalignment condition between GT and ST HP/IP, being this misalignment strongly influenced by the thermal growth of bearing supports and pedestals. The root cause of this misalignment could be due to a wrong (= out of tolerance) alignment in the cold condition and/or to an excessive thermal growth.

After more than one year of misaligned condition, a new phenomenon started to appear. During the thermal transients at Bearing 3 it started to be observed an oscillation of the direct levels of vibration (Figure 11). As observed in the spectra, these oscillations were due to very low frequency vibration – the phenomenon used to last few minutes at the beginning, but after several months it used to last about 4 to 6 hours, always during the thermal transient period.

This behavior was diagnosed as a worsening of the bearing condition, possibly a looseness between the bearing and the casing, due the action of high preloads during many months of operation.

For all the above, it was recommended to check the alignment condition and Bearing 3 condition.

The maintenance operations done during the first available outage found a misalignment between GT and ST HP/IP. The rim-face readings indicated a strong misalignment, which was improved up to a level considered acceptable, but still slightly out of specs. This compromise decision was taken in order to minimize the downtime of the unit.

The inspection of the bearing revealed that there was a significant looseness in the bottom part of Bearing 3, between the bearing itself and the casing. This looseness is not considered normal (this bearing is normally mounted with interference), and is indicative of a wear process which was developing with time.

The analysis carried out above indicates that this wear process was caused by the abnormally high preloads acting on the bearing. The bearing was repaired and then properly mounted.

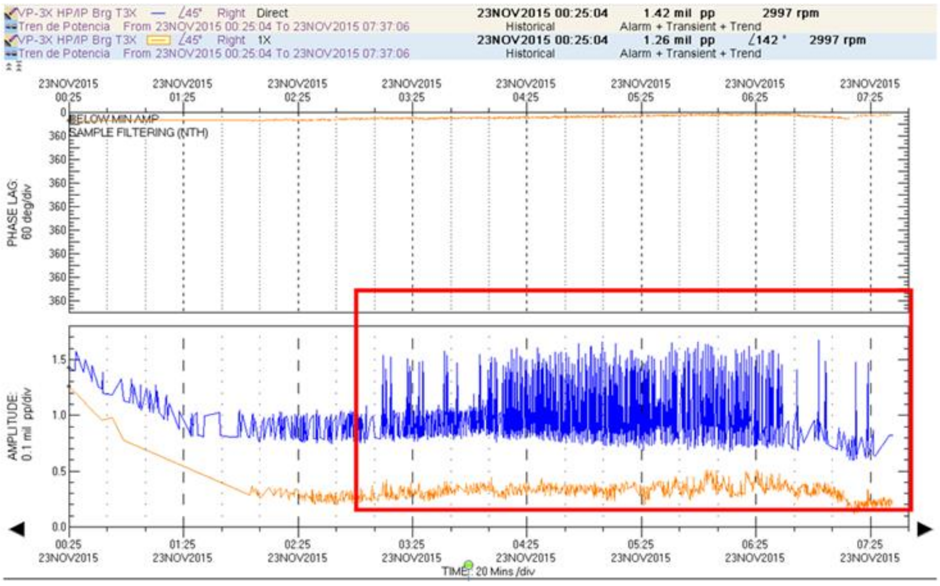

Figure 11. Direct vibration and 1X trends measured at Bearing 3 during the thermal transient. The direct amplitudes oscillate during the thermal transient.

After the overhaul, the new SUs showed that the direct vibration oscillations had disappeared, confirming the relationship between this pattern and the looseness found at Bearing 3.

Moreover, the orbit pattern on load at this bearing after the thermal transient became much less flattened than before the outage (Figures 12 and 13), indicating a reduction of the preloads in the new mechanical condition. This analysis was confirmed by the SCL plots: the comparison between the new SU and SD curves at Bearing 3 indicated that the leftward movement was reduced up to 1 mil, a value considered normal for this bearing (Figure 14).

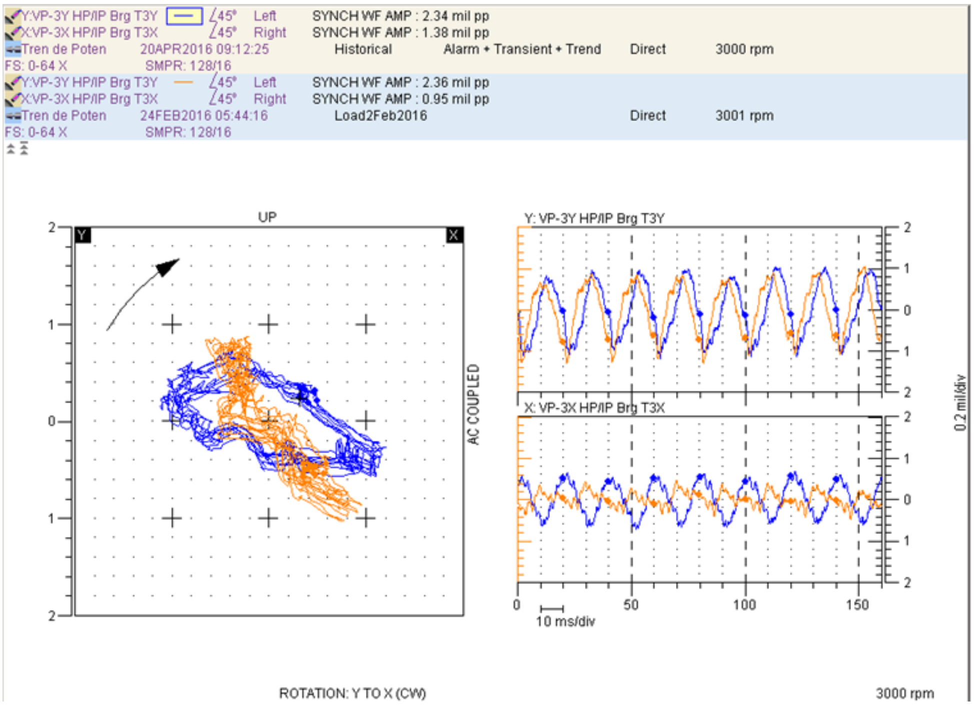

Figure 12. Direct uncompensated orbits overlay – Orbits measured at Bearing 3 on load before (orange) and after (blue) the alignment.

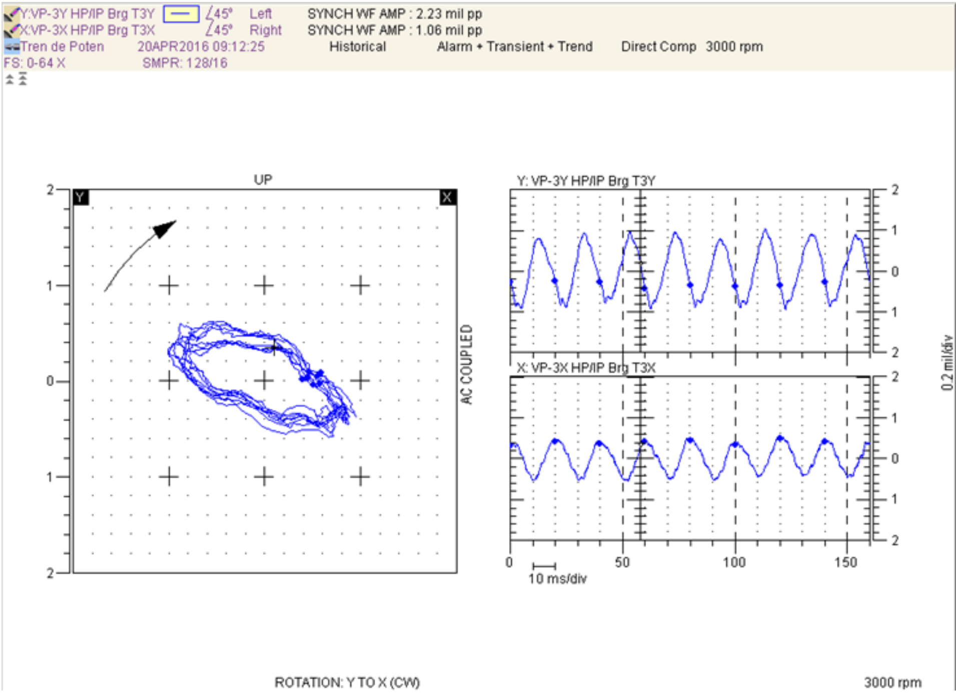

Figure 13. Direct compensated orbit on Load at Bearing 3 after the alignment.

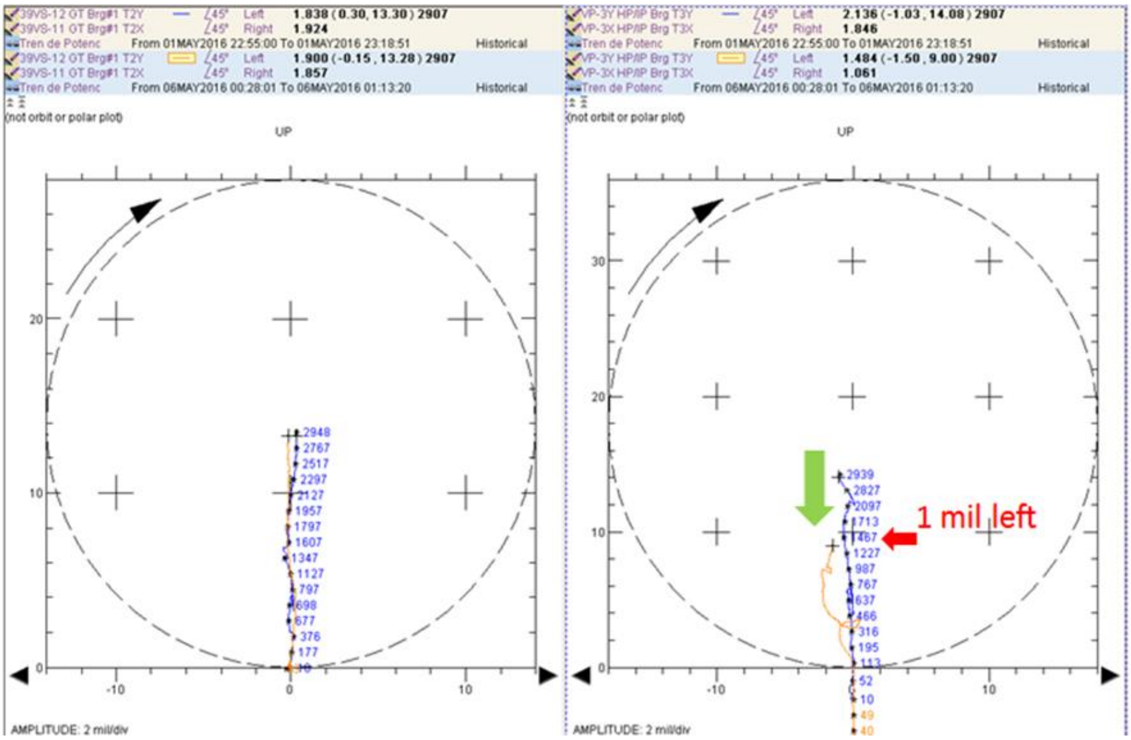

Figure 14. SCL plot measured at Bearings 2 and 3 during the SU (in blue) and during the SD (in orange) after the alignment. Compensated to avoid the effects of temperature on position measurement.

3 Conclusions

The cases presented in this article show that the detection of external misalignment involves the analysis several indicators. Case 2 also illustrates that sometimes diagnose misalignment can be challenging, being complicated by the dynamic of the rotor system and even by the unreliability of the shaft centerline position measurement, which is heavily affected by the influence of temperature on the probes supports.

Case 2 also shows that the thermal gradients themselves, can complicate the achievement of a good alignment condition after the thermal transient, on load condition.

Many engineers think that the 2X component is the only clear indicator of misalignment, but the diagnostic method and the real cases demonstrate that this theory is wrong, and can lead to big misunderstandings. Such 2X component could be the consequence of different malfunctions such as misalignment, generator rotor stiffness asymmetry, bearing support resonance, abnormal asymmetry of rotor stiffness, journal ovalization, etc. So the existence of the 2X component in the rotor spectrum can mainly be due to two factors:

- shaft misalignment and resulting radial preload, or radial preload generated by fluid flow;
- shaft asymmetry (such as cracked rotor) together with radial preload (from misalignment, fluid flow, gravity or other origins).

In the first situation the shaft rotates in the range of deflection where the stiffness of supports behaves nonlinearly. The region of deflection is determined by the specific region of the radial preload. The rotor response is not the same in vertical and horizontal directions. The stiffness non linearity causes not only changes in the 1X response amplitude but is also responsible for the generation of higher harmonics. In that case the main vibration excitation is unbalance. The preload and the nonlinearity modify the response to this unbalance.

In the second situation the preload plays also a role but the second component playing an important role is not the system stiffness nonlinearity but stiffness anisotropy (asymmetry) and only of those elements which are involved in rotative motion (mainly the rotor). The moderate preload together with the rotating anisotropy of shaft stiffness causes another excitation force. As the stiffness changes twice per one rotation of the shaft, this excitation has a 2X component. The mechanism in that case is purely linear.

It is important to look at the shape and precession of the 2X orbit. Indeed from the standard course of Bently Nevada it is known that "banana" and “figure eight” shapes are created when 2X precession is close to “undetermined”, meaning it can be forward or can be reverse but the 2X orbit is flat. The action of 2X component is “along the line”, which should be related to properties of stationary system. The internal loop in the orbit is created by the component with sufficiently forward precession, the asymmetry “rotates with shaft” and the stationary force (preload) causes variation in rotor bending, two times per revolution. This distinction is nicely visible in orbits. This can be noticed when looking at spectrum, but only full spectrum.

For sure, the orbit pattern is a very powerful indicator of preloads/misalignment, but this data must always be supported by evidences of wrong shaft position (SCL plot), bearing metal temperature, casing vibration. More than that, the behavior of the machine must be deeply understood, and several conditions must be analyzed (start-up, thermal transient, load, etc.). Only such a deep analysis, which includes the use compensated data, and the overlay of curves can bring reasonable certainty to the diagnose of misalignment.

Acknowledgments

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