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SUBSYNCHRONOUS RADIAL VIBRATION INDUCED BY AXIAL VIBRATION

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Abstract. *The paper presents a presence of subsynchronous radial vibration during mechanical running tests of High Pressure Compressor following API 617-7th ed. requirement. During the tests it was investigated how was the lateral vibration behavior while the rotor speed, oil pressure and flow to thrust bearing, load and torque were varied. After some tests, vibration analysis and literature revised, it was concluded that the main cause of radial subsynchronous vibration was due to axial subsynchronous vibration of rotor.*

Keywords: *subsynchronous vibration, axial vibration, compressor, mechanical running test*

1. INTRODUCTION

Whenever a subsynchronous vibration is found in a spectrum of a rotating machine, one becomes a concern because of its potential to induce instability exciting the damped natural frequency of the rotor system.

During mechanical running tests in vacuum of high pressure compressors, designed to work in off-shore oil production, subsynchronous vibration (SSV) were observed in radial direction using proximitor probes. Three different types of compressors were tested and in all of them the SSV were observed with different frequencies and amplitudes.

The tests were carried out as specified by API617 - 7th ed. and the limit of the amplitude of any discrete, nonsynchronous vibration (limited to 5micra) was sometimes exceeded by this SSV.

During investigations, one possible cause for the SSV was the possibility of axial vibration since there wasn't any axial load in thrust bearing. Although axial vibration is not usually monitored in industrial area, in order to check the veracity of this proposition, it was proposed measuring axial vibration while the following parameters were varied:

- Oil pressure
- Oil flow rate
- Rotor Speed
- Load on thrust bearing
- Torque

2. TEST PROCEDURE AND COMPRESSOR DATA

In this paper is presented the data of tests carried out with two different compressors model which will be named hereinafter “A” and “B”. The tests were performed following these main steps:

a. Run up the rotor up to overspeed with no load, since the rotor was running in vacuum.
b. Running the compressor during four hours in maximum continuous operating speed (MCOS).
c. Changed the oil pressure and flow to bearings as rotor was running in MCOS.
d. Coast down of rotor without torque.
e. Running the compressor with load, during performance test, but in speed lesser than MCOS, otherwise, the compressor and test bench could be damaged.

A dynamic signal processing instrument was used to record the vibration data. The software provides frequency spectrum, orbit, trend and waveform. As show in figure 1, two radial probes were mounted in each housing journal bearing, 45 degrees off top dead center, and an axial proximity was mounted at end of the shaft. All tests were run with ISO VG 32.

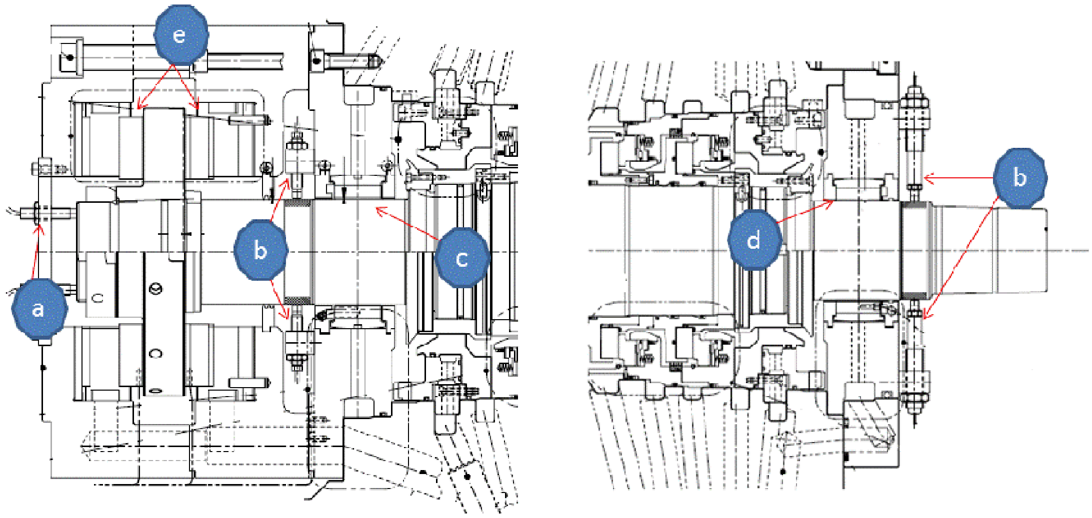

Fig. 1. Diagram of probes assembly
(a:axial probe; b: Radial probes; c: no drive end bearing; d: drive end bearing; e: thrust bearing)

Measurements include rotor speed, oil supply pressure, bearing pad temperature, oil flow for each individual bearing, axial displacement, suction and discharge pressure of compressor.

Information of compressor design and representative data from the tests are provided in the following sections to relay the main points of the tests and observations.

2.1 Compressor A

The compressor A, which has two sections, was designed to be submitted the following operation conditions: first suction pressure (=47.39bar), first suction temperature (=40°C), first discharge pressure (136bar), first discharge temperature (=123.7°C), second suction pressure (=135bar), second suction temperature (=40°C), second discharge pressure (=250bar), second discharge temperature (=86°C), mass flow (=107,400kg/h), range of critical speed (=5,250-5,550rmp), rotor weight (350kg), maximum continuous speed (=12,978rpm) and overspeed (=13,367rpm).

The both journal bearings are five-pad tilting pad type with 40mm length, 90mm diameter, load on pad, offset pivot of 60% , diametral clearance of 0.13mm and intake pressure of 1.5bar. And the thrust bearing is double acting self-equalizing with eight pads, offset pivot of 60%, total clearance of 0.3mm and intake pressure of 0.65bar. Figure 2 shows these bearing:

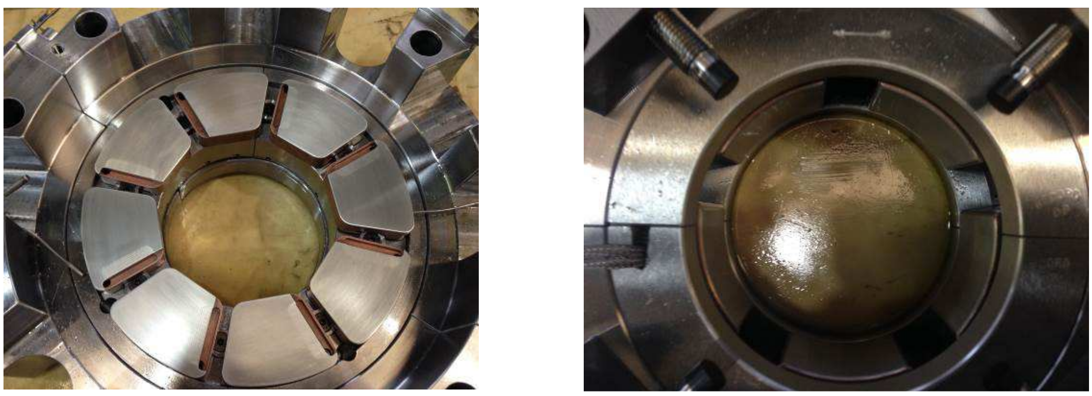

Fig. 2. Left: Thrust bearing. Right: Journal bearing

As shown in figure 3, at first run up of the rotor was already observed a 142.5Hz subsynchronous radial vibration indicated by probe X-NDE, which is mounted in no-drive-end bearing. . Although the subsynchronous vibrations were measured by the others probes, the amplitude was too low. So, just the data acquired by sensor X-NDE and axial probe was analyzed.

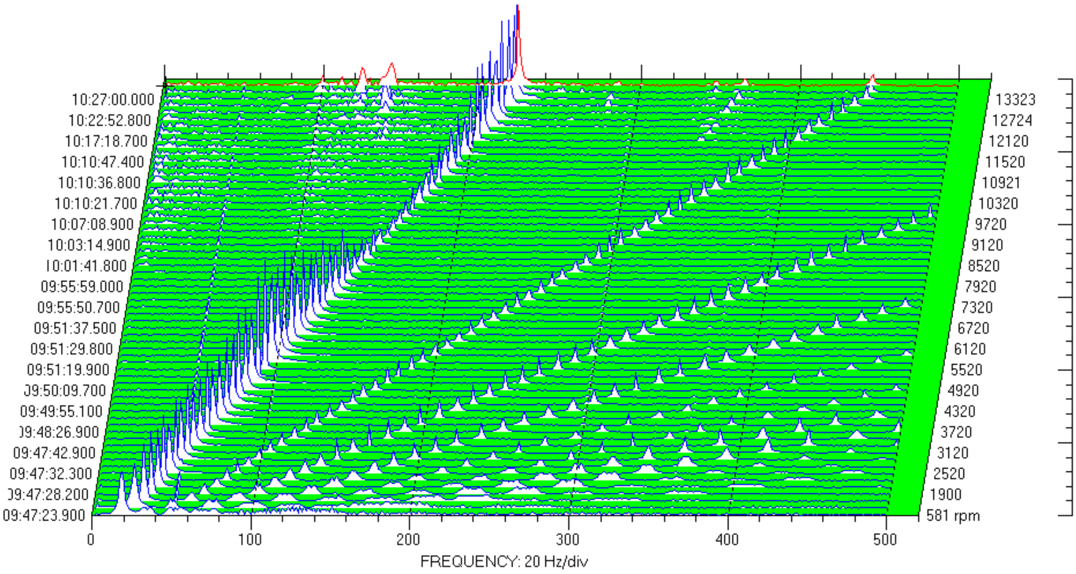

Fig. 3. Waterfall of the radial probe X-NDE during run up

When the compressor is running in overspeed, it is clearly seen, in figure 4, that the two frequencies, 125 and 142.5Hz, excite the rotor in axial direction with amplitude varying from 28µm-pp to 86µm-pp while the radial probe X-NDE records vibration varying from 0.93µm-pp to 1.86µm-pp. The average synchronous radial vibration (1N) was 5.1µm-pp, so the subsynchronous was near of 30% of 1N. During this period, the average oil pressure and flow in NDE bearing are 1.29bar and 28.1l/min, respectively. At the same moment, the average oil pressure, flow and displacement of thrust bearing are 0.85bar, 122.5l/min and 79.56µm. The displacement of thrust bearing refers to axial static equilibrium position of rotor that means the gap between active thrust bearing and thrust collar.

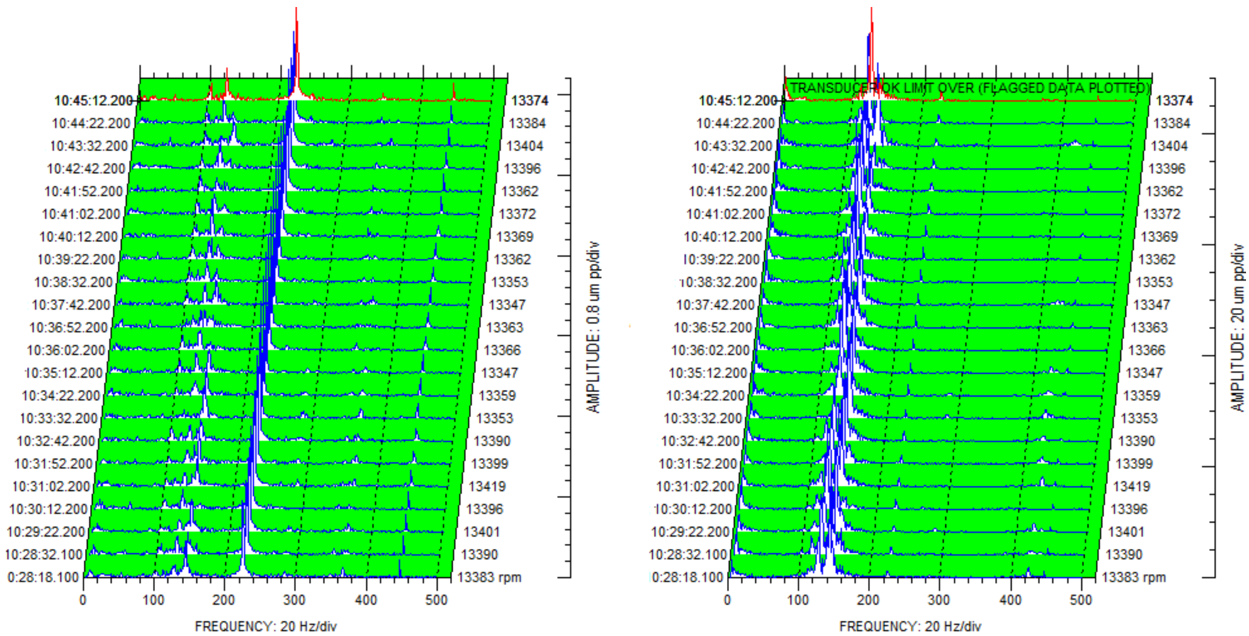

Fig. 4. Waterfall during overspeed – 13,367rpm. Left: X-NDE bearing / Right: Axial bearing

In order to identify which synchronous speed the subsynchronous beginning, the rotor speed was coasted down and run up as shown in figure 5. It is noted in both waterfall that the threshold of subsynchronous frequencies, 125 and

142.5Hz, is around 186Hz, with and without torque. Also it was identified the low frequency excitation in 2.5Hz that starts around 128.75Hz synchronous speed. During this period, the average oil pressure and flow in NDE bearing are 1.18bar and 26.97l/min, respectively. At the same moment, the average oil pressure, flow and displacement of thrust bearing are 0.85bar, 0122.4l/min and 81.25µm.

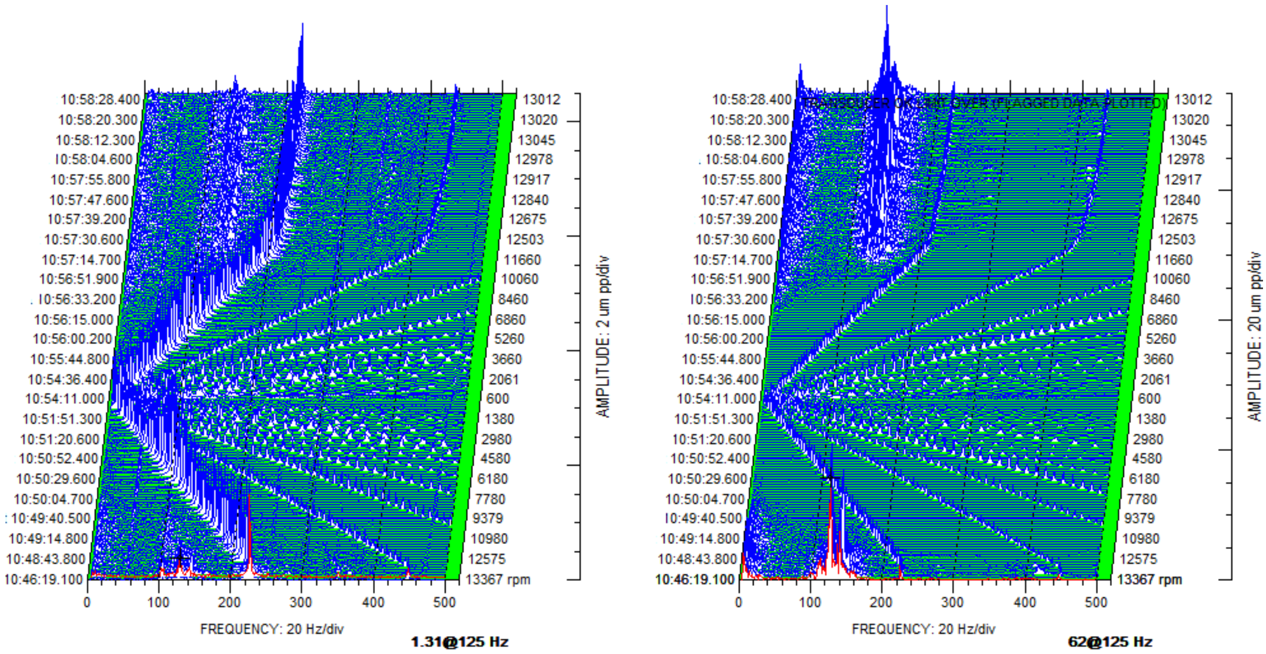

Fig. 5. Waterfall – Coast down and run up of rotor. Left: X-NDE bearing / Right: Axial bearing

The next step was to change the oil pressure and flow to the bearings whilst the rotor speed was kept near 12.978rpm without load. The figure 6 and 7 were split in region in order to easily link the operation condition and forced response. The following average operation conditions at bearings were established:

Table 1. Bearing Operational Conditions of the Bearing

		Thrust Bearing			Radial Bearing	
	Period	Oil Flow (l/min)	Oil Pressure (bar)	Axial Disp. (µm)	Oil Flow (l/min)	Oil Pressure bar
A	11h01m46s– 12h01min10s	135	0.96	82	27.8	1.24
B	12h01min10s-12h41min10s	170	1.51	75.9	33.4	1.59
C	12h41min10s-13h51min10s	92.4	1.52	83	29.12	1.37
D	12h21min10s-12h31min-10s	124.5	0.8	85.3	NA	NA

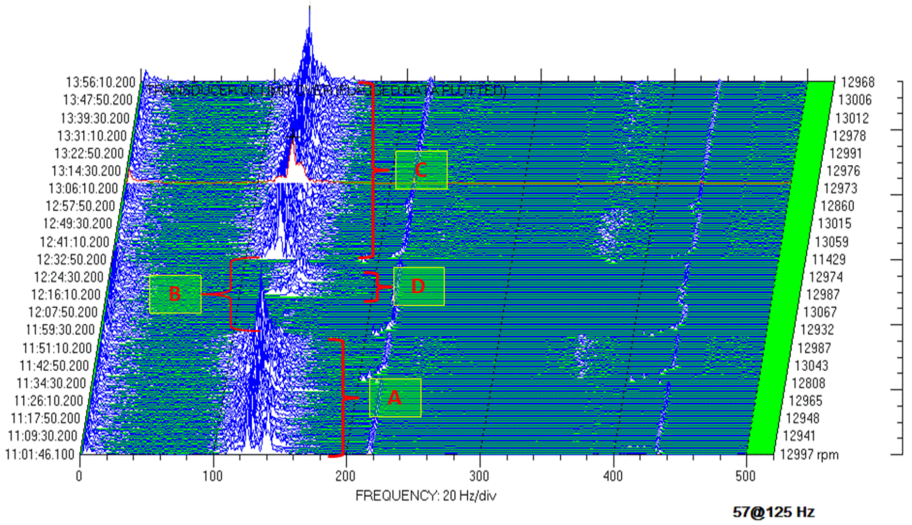

Fig. 6. Waterfall axial bearing – Rotor speed in steady state condition

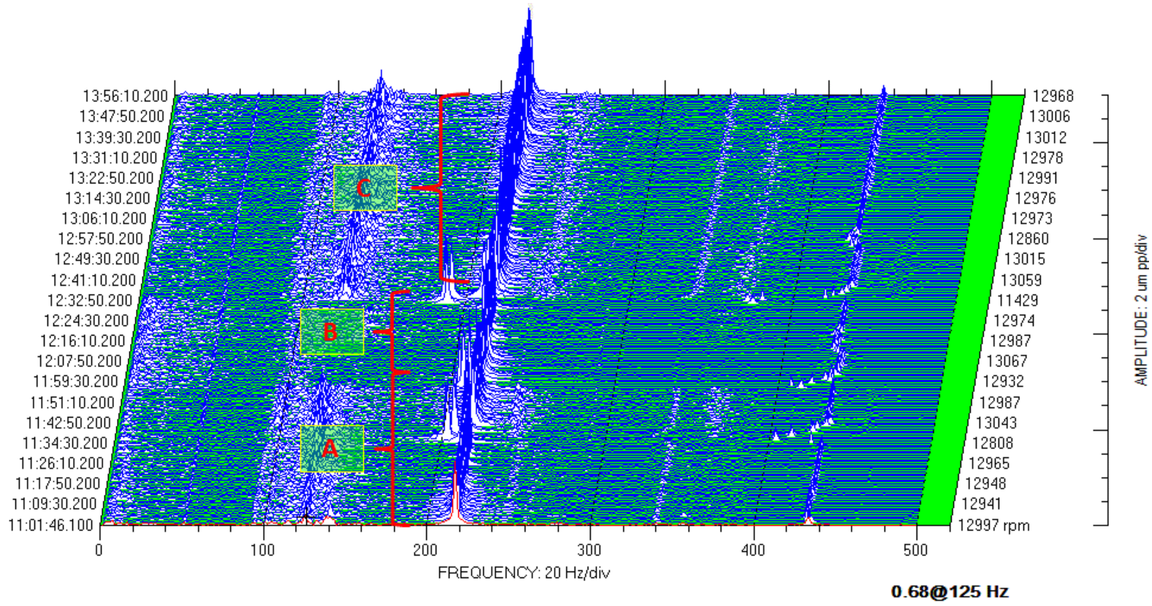

Fig. 7. Waterfall radial bearing (X-NDE) – Rotor speed in steady state condition

Analyzing the results above, it is possible to conclude that increasing thrust bearing stiffness by increasing oil flow and pressure and reducing axial displacement, the subsynchronous (125/142.5Hz) disappears. Therefore, it is expected that this subsynchronous shall not occur when the compressor work in actual condition on site because the load condition also increases thrust stiffness. The SSV at 2.5Hz remained present even though thrust bearing stiffness was increased. Since the amplitude at 2.5Hz is too low in radial direction, this is not a concern.

Tests described above were carried out with of rotor in vacuum (-0.95bar) so in no load condition. During the test with load varying the differential pressure in first stage between 6.3 and 9.5bar and the second stage from 2bar to 7.0bar at 9,000rpm, it was just noted that the 2.5Hz frequency appeared near synchronous speed of 128Hz with lower amplitude. The rotor speed did not reached the 11,160rpm, threshold limit of 125/142.5Hz subsynchronous frequency, because of there is a temperature limit on the casing which did not allow this test. Even if the subsynchronous (125/142.5Hz) appears during actual operation conditions of the compressor, it is expected low amplitude vibration since this happened with the 2.5Hz frequency which had similar behavior.

2.2 Compressor B

The compressor B, which has one section, was design to be submitted the following operation conditions: suction pressure (=249bar), suction temperature (=40°C), discharge pressure (549bar), discharge temperature (=96.6°C), mass flow (=215,100kg/h), range of critical speed (=6,100-6,250rpm), rotor weight (=300kg), maximum continuous speed (=11,400rmp) and overspeed (=11,750rpm). The bearings were similar with the ones used in compressor A.

Starting mechanical running test, it was seen the occurrence of 142.5Hz subsynchronous vibration when rotor speed was around 10,260rpm record by probe Y-NDE. – see figure 8. During this period, the average oil pressure and flow in NDE bearing were 1.14bar and 18.22l/min, respectively. At the same moment, the average oil pressure and flow in thrust bearing were 0.62bar and 76.5l/min. Moreover, the amplitude of radial vibration was 5.35µm-pp at 142.5Hz against 3.72µm-pp at 1N. Beyond the subsynchronous vibration was higher than synchronous, it exceeded the criteria limit proposed by API 617-7th which is 5 µm-pp.

One of the probable cause of higher radial SSV in compressor "B" than "A" is due to oil flow to thrust bearing being lower in the compressor B.

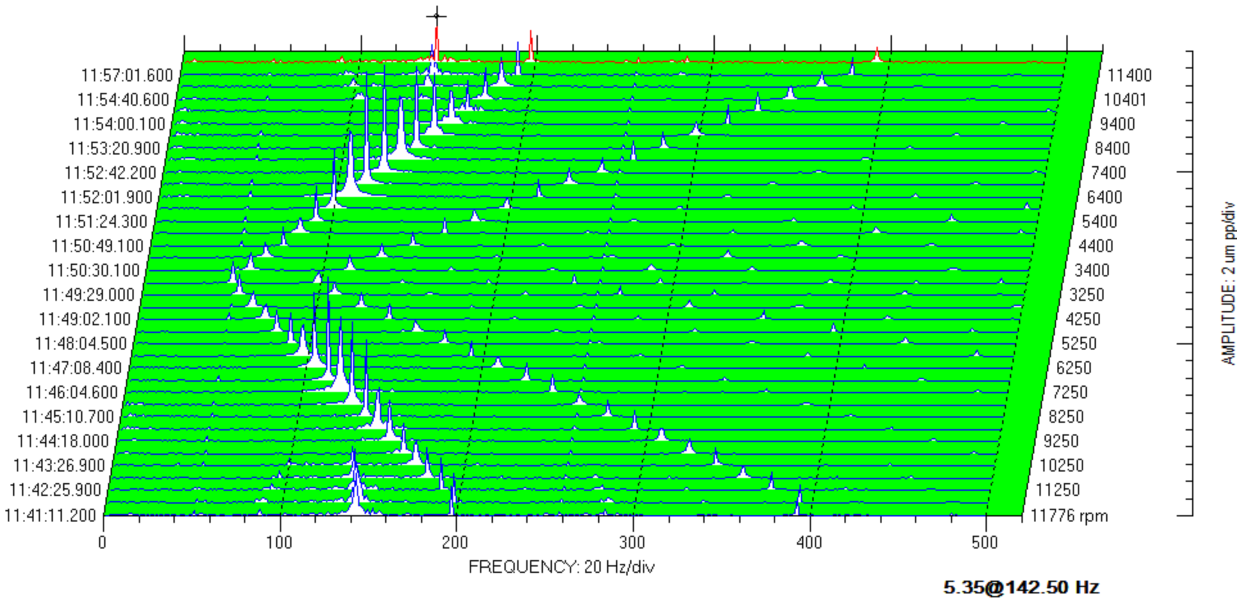

Fig. 8. Coast down and run up of the rotor speed – Radial vibration (Y-NDE)

This subsynchronous excitation seems to have backward direction, that is, opposite the whirl of synchronous frequency – see figure 9. While the vibration in 142.5Hz is present, the orbit of the shaft in NDE bearing looks like a "nest" and is too hard to analyze. But, when this SSV disappears, the whirl synchronous frequency is easily noted as displayed in figure 10.

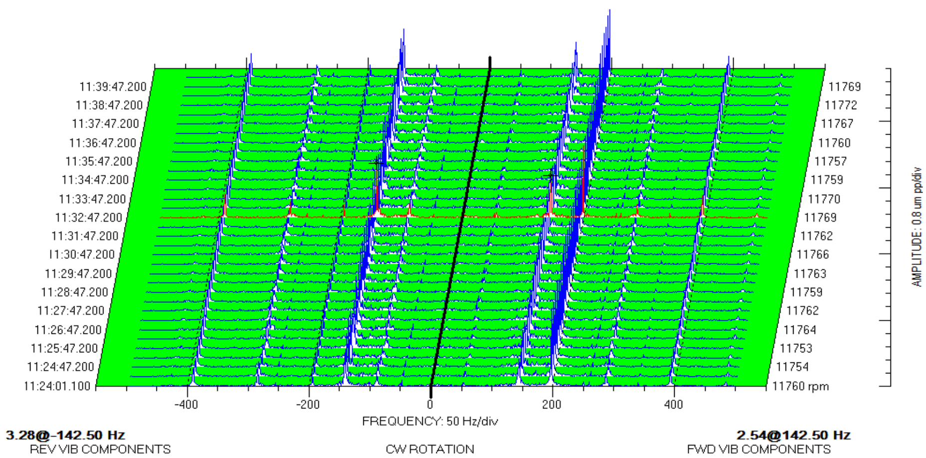

Fig. 9. Subsynchronous backward whirl – Y-NDE bearing

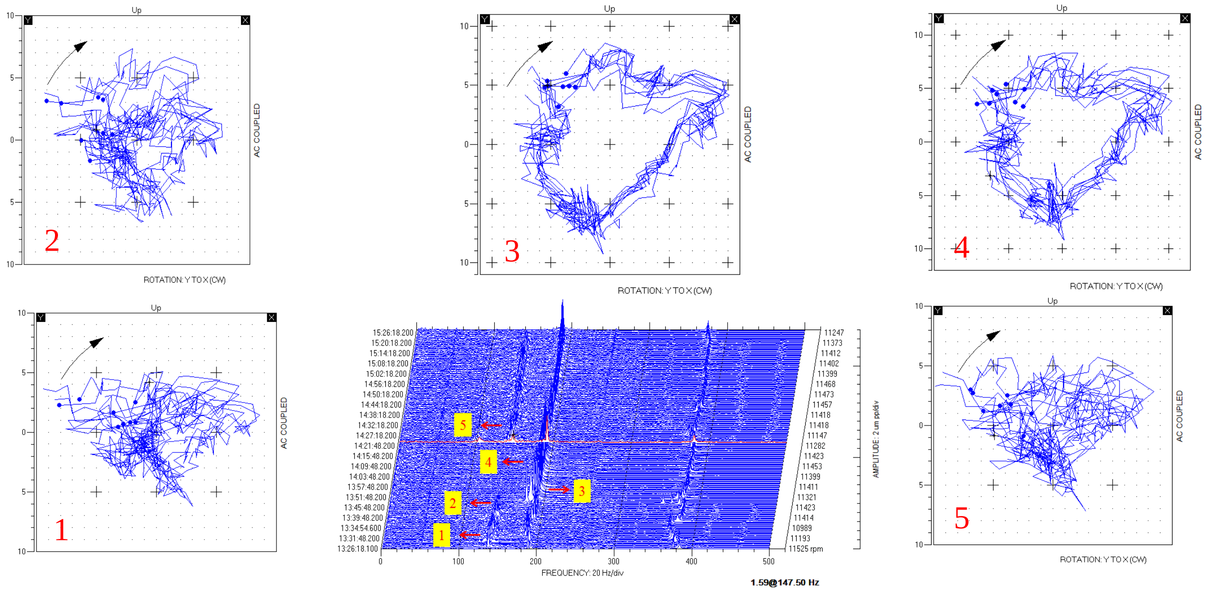

Fig. 10. Orbit of the shaft at no drive end bearing

This likely backward subsynchronous frequency present during the mechanical running test of compressor "B" may be the cause of decreasing of the synchronous vibration as revealed by figure 11.

However, despite the differences about the compressors, both tests presented similar results although the amplitudes of the SSV are different. Furthermore, the main variable responsible by subsynchronous frequency in radial vibration was the axial vibration in frequency lower than synchronous speed. In addition, the axial SSV displayed more sensitivity to change in oil flow than load, speed and torque.

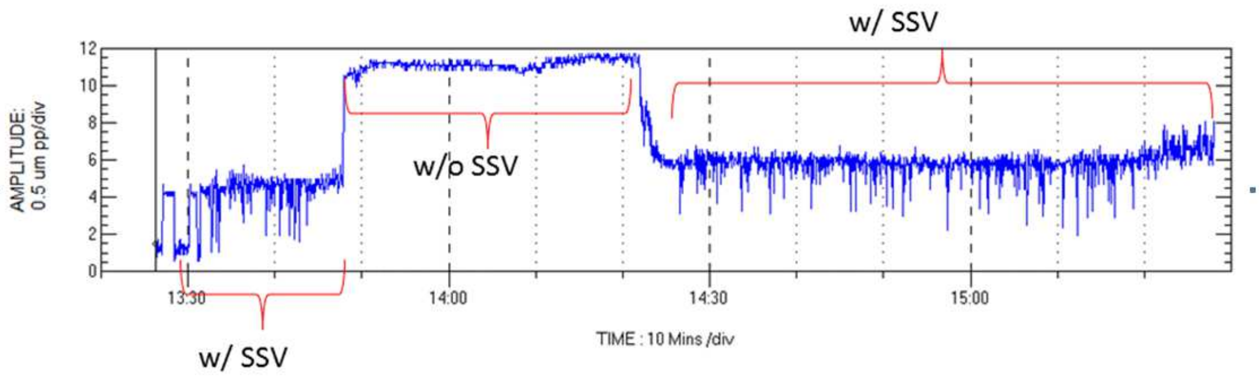

Fig. 11. Trend of Synchronous Vibration at Y-NDE bearing

2.3 CONCLUSIONS

The main cause of radial subsynchronous vibration, in these analyses, is due to axial SSV. That is, the dynamic of thrust bearing is coupled with the rotor-journal bearing system. A model which could be explaining this behavior is described by Lie, et al [3].

Oil flow rate displayed being the principal variable responsible for axial SSV. One possible cause of SSV is that the insufficient oil to completely fill the pad film thickness may trigger pad flutter behavior.

As mentioned above, this radial SSV could excite in backward direction decreasing synchronous whirl. But, it is not a rule because the same kind of radial SSV in forward direction has been found.

If these both compressor present this radial SSV during operation on site, the solution will be increase the oil flow rate to thrust bearing.

In spite of not being a API 617 requirement and not usual monitored in industrial applications, these test showed the importance of set the axial probe to measure vibration instead of just check rotor displacement during mechanical running test and operation on site.

Unfortunately, it was not possible to make more analysis in order to put the integrality of machinery in risk. This is the disadvantage of doing test in real machinery. On the other hand, it is possible to closer the operation test condition to actual operation condition.

3. REFERENCES

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