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MONITORING AND IDENTIFYING VIBRATION SOURCES OF OF ROLLING BEARING AND HYDRODYNAMIC BEARING TURBOCHARGERS

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Abstract: *The monitoring of rotor systems has a relevant importance because it can supply useful data over the proper functioning conditions of the rotating machine. Also it can provide relevant data in order to identify the eventual malfunctioning defects that can be reported in the incipient phase when the rotating machine is not totally damaged and the repairing cost are less higher than the costs that involve the change of the entire rotating machine. The study highlights some aspects related to monitoring of general vibration amplitudes of turbochargers equipped with hydrodynamic and rolling bearings also highlighting aspects related to vibration source identification that occur during turbocharger functioning at high rotating speeds. The measurements accomplished in this study present the general vibration amplitudes of two different types of turbochargers one with hydrodynamic bearings and one with rolling bearings, the signals being gathered from two different directions.*
Keywords: *rotordynamics, vibration, turbocharger, vibration source.*

1. INTRODUCTION

Turbochargers are constantly subjected to high centrifugal forces that occur during functioning at high speed that in some cases can exceed 250000 rpm. The monitoring of the unbalance that occur during functioning because of the deposit of unburned fuel on the turbine impeller can provide useful data on the proper functioning of the turbocharger. The deposit of the unburned fuel on the turbine impeller creates additional forces that the rotor sustaining system in this case rolling and hydrodynamic bearings have to support during functioning. These forces modify the unbalance plane in such manner that a removal of the turbocharger is needed in order to remove the unburned fuel from turbine impeller [1], [7].
The unbalance of the turbocharger rotor can be identified using two means that involve measurements on dynamic turbocharger test rig or using high tech vibration data acquisition.
This study presents data measurements gathered using both methodology present above, aiming the real time monitoring of turbochargers sustained by rolling and hydrodynamic bearings. The measurements were accomplished in such manner that the data sets can be used for a pertinent comparison between the general vibration amplitudes of the two turbochargers subjected for study.

2. CONTENT

In order to identify the turbocharger unbalance it were accomplished several tests at stabilized regime of 30000, 55000 and 90000 rpm, performed on the dynamic turbocharger test rig Schenck MBS 110 presented in figure 1.

Figure 1. Schenck MBS 100 test rig

The test rig uses for turbocharger driving mechanism compressed air at high pressures. The oil is supplied by a hose which can also be observed in figure 1. The turbocharger is mounted on a seismic platform that has a accelerometer which measurers the unbalance vibration amplitude in units of $\mu m / s$.

The unbalance measurements for the rolling bearing turbocharger are presented in figure 2 a,b,c.

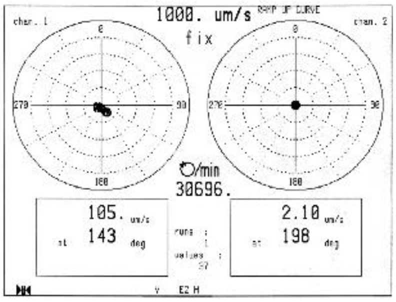

Figure 2.a Rolling bearing turbocharger unbalance for 30000 rpm

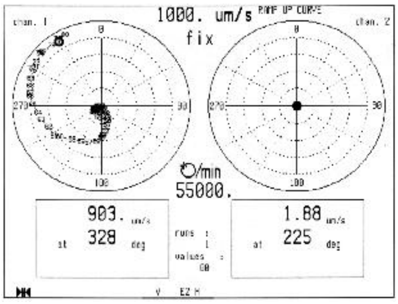

Figure 2.b Rolling bearing turbocharger unbalance for 55000 rpm

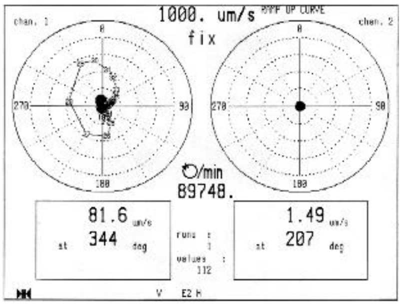

Figure 2.c Rolling bearing turbocharger unbalance for 90000 rpm

Using the same three stabilized regimes several measurements of turbocharger rotor unbalanced were performed for the hydrodynamic turbocharger. The unbalance measurements for the hydrodynamic turbocharger are presented in figure 3 a, b, c.

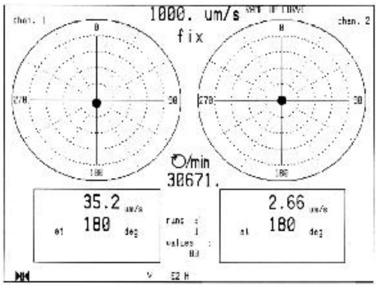

Figure 3.a Hydrodynamic bearing turbocharger unbalance for 30000 rpm

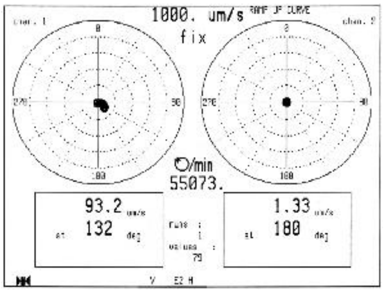

Figure 3.b Hydrodynamic bearing turbocharger unbalance for 55000 rpm

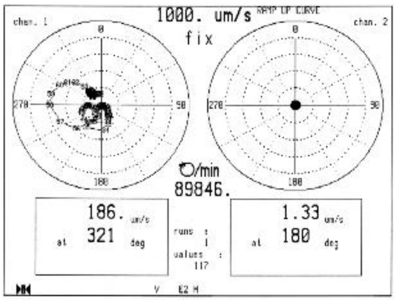

Figure 3.c Hydrodynamic bearing turbocharger unbalance for 90000 rpm

In order to identify the vibration sources generated by the functioning of the rolling bearing turbocharger several tests had been accomplished using pulse 12 vibration acquisition platform. There were used 6 accelerometers mounted on the turbocharger intermediate housing next to the bearing. The channel set up and the platform settings are presented in figure 4.

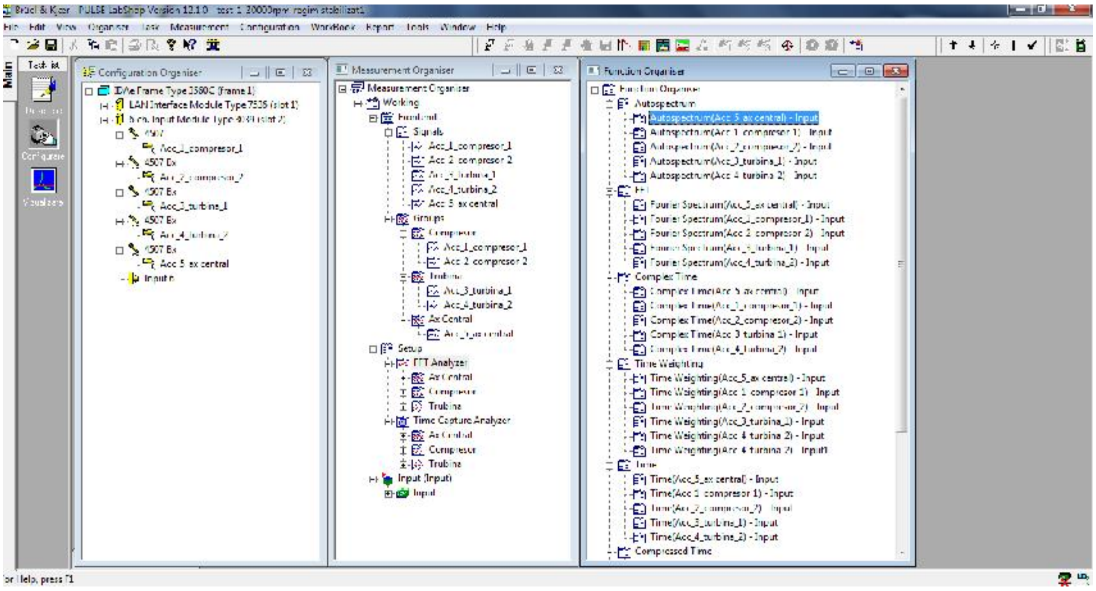

Figure 4. Platform and accelerometer settings

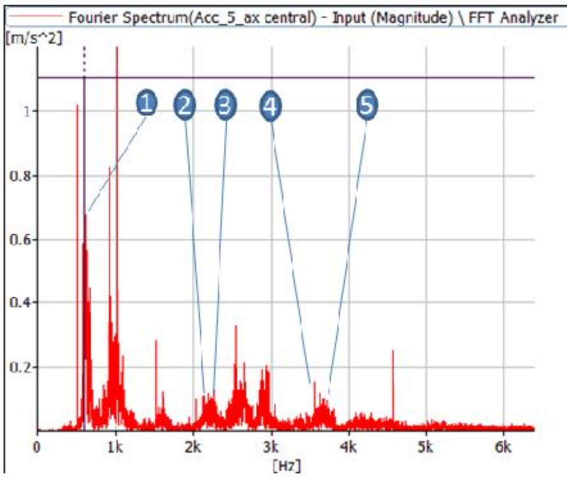

Figure 5. Measurements performed at 30000 rpm stabilized regime on rolling bearing turbocharger

Form figure 5 we can denote the principal 5 amplitudes that are related to some possible vibration sources besides the fundamental frequency given by [2], [8]:

$$f_n = \frac{n}{60} = \frac{30000}{60} = 500\text{HZ} \tag{1}$$

In order to be precise with data interpretation we have excluded from the amplitudes presented in figure 4 the frequencies related to f_n=500 Hz with multiples and submultiples like: $1f_n$; $2f_n$; $3f_n$; $5f_n$; $9f_n$ that correspond to 500 Hz; 1000 Hz; 1500 Hz; 2500 Hz; 4000 Hz.

The remaining amplitudes noted with 1 to 5 are related to the following possible vibration sources:

- 1- 595 Hz rotor vibration frequency;
- 2- 2119 Hz rotor or bearing vibration frequency;
- 3- 2264 Hz bearing vibration frequency;
- 4- 3558 Hz fundamental vibration frequency which also corresponds to one of the bearing vibration frequency;
- 5- 3661 Hz bearing frequency vibration frequency.

The signals gathered at stabilized regimes for 55000 rpm and 90000 rpm are presented in figures 6 and 7.

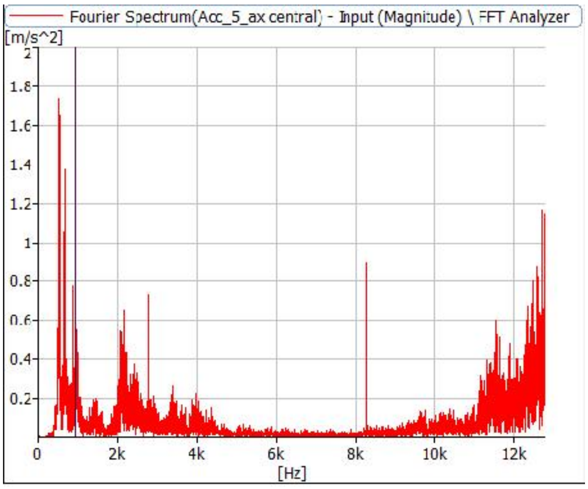

Figure 6. Measurements performed at 55000 rpm stabilized regime on rolling bearing turbocharger

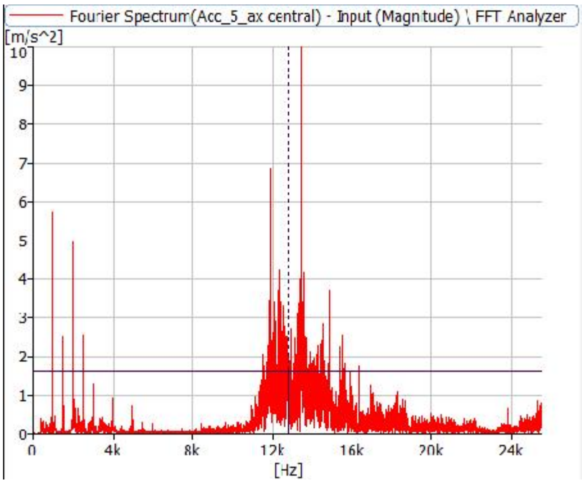

Figure 7. Measurements performed at 90000 rpm stabilized regime on rolling bearing turbocharger

It is to be mentioned that before performing the turbocharger dynamic tests there was accomplished an analytical calculus of bearing vibration frequencies and also several simulations performed on the components of the turbocharger entire assembly, simulations which highlighted the possible vibration frequencies of the turbocharger assembly.
Maintaining the same measurements parameters as for the rolling bearing turbocharger it were performed some dynamic hydrodynamic bearing turbocharger tests in order to identify the possible vibration sources.
The measurements highlighted the following facts:

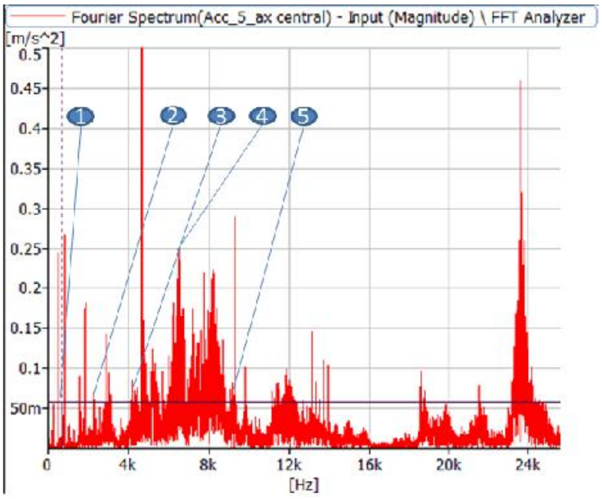

Figure 8. Measurements performed at 30000 rpm stabilized regime on hydrodynamic bearing turbocharger

In the first stage there were excluded the vibration that correspond to multiples or submultiples of the fundamental frequency f_n=500 Hz, frequencies that can be observed in figure 9 and correspond to the values of 500 Hz, 1000 Hz, 2500 Hz, 4500 Hz and 8000 Hz.

Excluding the frequencies mentioned above the frequencies noted with 1 to 5 in figure 9, have as possible vibration sources:

- 1- 672 Hz rotor frequency vibration;
- 2- 2260 Hz rotor frequency vibration;
- 3- 4328 Hz intermediate housing vibration frequency;
- 4- 6540 Hz intermediate housing vibration frequency;
- 5- 9224 Hz intermediate housing vibration frequency;

Maintaining the same aspects as for the rolling bearing turbocharger in figures 10 and 11 there are presented the signals gathered from the hydrodynamic bearing turbocharger at 55000 rpm and 90000 rom stabilized regimes.

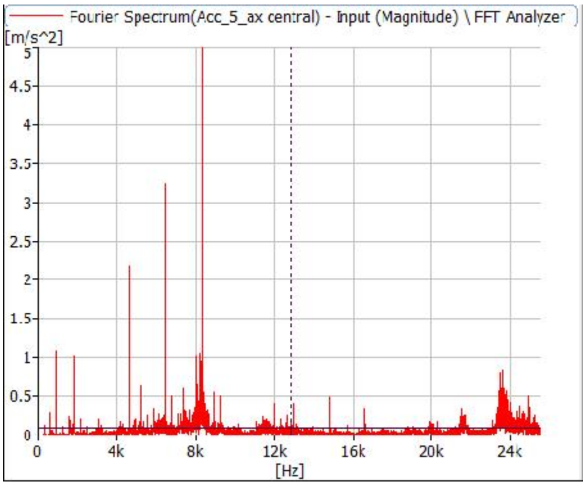

Figure 9. Measurements performed at 55000 rpm stabilized regime on hydrodynamic bearing turbocharger

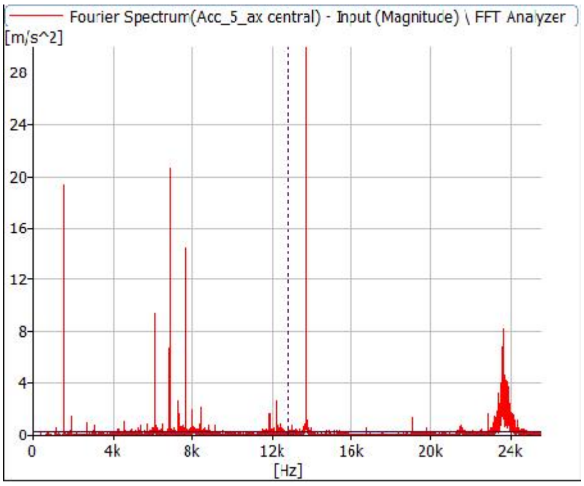

Figure 10. Measurements performed at 90000 rpm stabilized regime on hydrodynamic bearing turbocharger

The signals presented in this study of identifying the vibration sources of rolling and hydrodynamic bearing turbochargers have been gathered from the two accelerometers mounted on the intermediate turbocharger housing near the bearing.
The turbocharger subjected to study were considered from the same mass and same geometrical point of view the only difference being the rotor sustaining system which in one case is represented by rolling bearing and in the other case is represented by a hydrodynamic bearing.
The analytical calculus performed in order to identify the bearing frequencies was accomplished a mathematical methodology which takes into consideration the number of the rolling elements (balls) and the revolution speed of the bearing, the simulations regarding the rotor and the turbocharger intermediate housing being accomplished using specialized finite element method software ANSYS.

3. CONCLUSION

The measurements highlighted the fact the simulations accomplished were appropriate to the values obtained by effective measurements on dynamic turbocharger test rig.
In order to identify all of the vibrating sources of the turbocharger assembly it is necessary to develop multiple tests considering all of the turbocharger functioning conditions that involve the consideration of the transient regimes of acceleration and deceleration, functioning regimes which are dominant in the real functioning of the turbocharger.
The vibrational behavior of the turbocharger assembly is a complex one identifying the exact vibration sources being a difficult task to accomplish [6], [5].
The study highlighted the fact that in the case of the rolling bearing turbocharger the general value of vibration amplitudes are higher than the general value of the amplitudes gathered from the turbocharger with hydrodynamic bearings. This fact can be explained considering the influence of the oil film which in the case of the hydrodynamic bearing turbocharger acts also like additional damping device which diminish the vibration amplitudes [3].
In the case of the rolling bearing turbocharger the higher vibration amplitudes are justified by the vibration of the rolling elements (balls) which increase the vibration amplitudes with the increase of the rotor revolution speed.
In some cases it is to be considered that also the air which is transported on the turbine and compressor impellers generated certain vibrations which are more commonly founded at very high frequencies which in some cases can be called sound vibrations.

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