

# Condition monitoring of Pump-turbines. New challenges.

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## **1. Introduction**

Pumped storage hydropower plants are of paramount importance for the stability of the electrical grid. They can store huge amounts of energy by pumping water from a lower to a higher reservoir (pump operation) converting the surplus of electrical energy into potential energy [1]. At peak hours or in case of emergency this potential energy is converted again into electrical energy and delivered to the grid.

The machines used in these power plants are generally reversible pump-turbines (RPT). RPT are high performance machines that have to change operation from pump to turbine mode (reversing runner rotation and flow direction) in a short time. Due to their special design characteristics (large power concentration) and operating conditions, they generate large dynamic forces when in operation. In the last years, due to the massive entrance of wind power, the number of start/stop cycles of RPT has increased dramatically as well as the operation at off-design conditions subjecting the machine to large dynamic forces. The RPT rotating train and structure has to resist all these forces for a lifespan of several decades. In this new scenario, more cases of damage have been reported and a more advanced and effective vibration monitoring is necessary.

Compared with a conventional hydro turbine these machines have less number of blades (6 to 9) and higher rotating speeds operating at high pressure. The main excitation phenomenon is the well-known rotor-stator interaction (RSI) [2-5]. This excitation occurs in other kind of turbomachinery [2], but is critical in RPT due to the low number of blades and to the high head (difference in altitude between the upper and the lower reservoir). In some studies [3, 6, 7], RSI is supposed to be the cause of the failures found.

In this paper the vibration monitoring in RPT is analysed. The characteristics of the main excitation forces and the procedure used to select the monitoring parameters (spectral bands) are presented. An analysis of the vibration signatures measured in different machines as well as the main types of damage found after several year of monitoring are introduced and discussed.

## **2. Vibration based condition monitoring system**

### **2.1 Description of the monitoring system**

Since 1992 several pump-turbine units have been monitored. A reversible pump-turbine unit is a vertical shaft machine with a hydraulic turbine in the bottom and an electrical generator in the top. Typical RPT have three radial bearings, one in the turbine and two in the electrical generator, as well as one axial thrust bearing. For monitoring, sensors were located in all the bearings and in other positions depending on the machine (Figure 1). The signals from accelerometers and pressure sensors were acquired as well as the signals related to the operating conditions of the machine (head, distributor opening, pump/turbine operation).

In the beginning off-line monitoring using data collectors were used. In 1998 remote on-line monitoring systems were installed in many of them (Figure 2). Acquisition systems MVX from Acoem were installed in each machine. The MVX systems are connected to the power plant internal LAN and then to the company network. Signals are acquired continually and from time to time sent to a Diagnostic Center where all the power plants are supervised. Overall levels and spectral bands are calculated and trended. In case of alarm, spectra are analysed and if necessary more advanced signal

treatments like wavelets are used as well as numerical simulation techniques. A huge data base is now available for analysis.

## 2.2 Main excitation force (RSI)

Some important aspects have to be understood when setting up a monitoring system. The most important is to know what forcing frequencies are generated by the machine components (turbine, fluid film bearings, generator, etc.). The excitation forces in hydraulic machinery can be classified according their origin as hydraulic, mechanical and electromagnetic.

The mechanical forces generated are the typical of any rotating system like unbalance and misalignment [8, 9] but the hydraulic forces are different [10, 11]. Comparing a RPT unit with other hydraulic machines, the hydraulic forces (especially the Rotor-Stator Interaction) are much larger since the runner of these machines have fewer number of blades than in a Francis turbine and has to withstand high pressure.

The highest vibration level in RPT units comes from the pressure fluctuation induced by the rotor stator interaction (RSI) [3,6,7,12]. Basically, RSI arises from the interference between the rotating blades of the runner and the stationary guide vanes.

When the rotating blades of the rotor pass in front of the static vanes of the stator (see [4] for figure details) the pressure field in the gap between blades and vanes can be described as the superposition of all the combinations  $m, n$  [3, 4]:

$$p_{mn}(\theta, t) = A_{mn} \cdot \cos(mZ_v\theta_s + \phi_m) \cdot \cos(nZ_b\theta_r + \phi_n) \text{ for } m = 1, 2, \dots, \infty \quad n = 1, 2, \dots, \infty \quad (1)$$

In Eq.(1),  $\theta$  is the angular coordinate (index  $s$  is for stationary coordinate and  $r$  for rotating coordinate),  $Z_v$  is the number of guide vanes in the stationary part and  $Z_b$  the number of rotating blades in the runner.  $m, n$  are entire numbers that represent the order of harmonic. The amplitude of the excitation  $A_{mn}$  depends on several parameters like the head (difference in altitude between the upper and the lower reservoir), position of the distributor (operating point), design of the machine and order of harmonics  $m, n$ .

Analysing this equation in detail and rewriting it in the stationary reference frame [3, 4], the following main conclusions can be extracted:

- Due to this interaction, several harmonic frequencies are excited. Viewed from the stationary frame, these frequencies depend on the rotating speed of the rotor (runner)  $f_f$ , the number of rotating blades  $Z_b$  and the order of harmonic  $n$

$$f_n = n \cdot Z_b \cdot f_f \quad (2)$$

- The blade passing frequency is defined as  $f_{n=1} = f_b = Z_b \cdot f_f$
- The excitation shape corresponding to one excited frequency  $f_n$  is the superposition of several excitation modes that can be calculated with the following expression:

$$k = mZ_v - nZ_b \quad (3)$$

- Higher amplitudes are expected for lower  $|k|$  and lower harmonics  $m, n$ . Therefore and due to hydraulic design conditions of RPT's (number of rotating blades and guide vanes), usually the highest value  $A_{mn}$  is obtained for  $m=1$  and  $n=2, 3$ , which usually gives a  $k=2, 3$ .

- The sign of  $k$  indicates the rotation of the frequency pulsation. If it is positive the pressure pulsation rotates in the same direction than the runner and in the opposite when it is negative.

### 2.3 Analysis of the vibration signatures

For an effective condition monitoring the vibration signatures have to be known. RPT are not standard radial turbines (Francis), compared to them, RPT have less number of blades in the runner, higher heads and higher rotating speeds. Depending on the head and on the discharge, different phenomena can occur with significant changes in the vibration signatures.

In Figure 3, two vibration signatures are compared, one belonging to a standard Francis turbine (375rpm,  $Z_b=14$ ,  $Z_v=20$ ) and one belonging to a RPT (500rpm,  $Z_b=9$ ,  $Z_v=20$ ). In both cases vibration was measured in the turbine bearing. It can be seen that while the hydraulic excitation in the Francis turbine appears at the blade passing frequency  $f_b$  ( $k = 6$ ), in the RPT appears at  $2 \cdot f_b$  ( $k = 2$ ) with a much higher amplitude.

The main characteristics of the machines analysed in this study and predicted predominant frequency for each machine are indicated in Table 1. The predicted predominant frequency is calculated according to the RSI characteristic analysis (see chap. 2.2).

In Figure 4, the theoretical predominant frequency according to the RSI analysis is validated experimentally for RPT-1 and RPT-3. The higher vibrations occur at 140 Hz ( $2f_b$ ) in RPT-1 and at 168.75 Hz ( $3f_b$ ) in RPT-2. The amplitudes depends very much on the operating mode and on the power delivered by the machine Figure 4a. These amplitudes are at maximum for full power turbine operation and higher than in pump operation mode (Figure 4b). This makes the monitoring of RPT more complex, because it is necessary to distinguish between changes due to operating conditions and changes due to damage.

In the generator bearing (Figure 4c), vibration amplitudes are lower and signatures are different. In this case the predominant frequency is  $f_f$  (unbalance), although the hydraulic excitation due to RSI can also be detected.

### 2.4 Selection of spectral bands

Like in other rotating machinery properly specified spectral alarm bands are one of the most critical tools to detect potentially serious problems in hydraulic turbines. Spectral bands were selected for the monitoring of RPT after a detailed analysis of the excitation forces and of the dynamic characteristics of the machines. To properly set up the spectral bands it is necessary to understand what problems are detectable by vibration analysis in each measuring position. It is known that mass unbalance, misalignment, bearing wear, hydraulic forces and some electrical problems can be detected rather easily treating the vibration signals.

RPT are vertical shaft machines with three radial fluid film bearings connected to foundation and one vertical thrust bearing. The turbine and the generator are connected by a rigid coupling. The turbine bearing withstand mechanical and hydraulic forces while the two generator bearings withstand mechanical and electromagnetic forces. Spectral bands in the generator bearings and in the turbine bearing are selected according to each type of forces.

### 2.4.1 Generator bearing

A sub-harmonic band is selected to detect bearing damage. Damage in fluid film bearings may generate sub-synchronous vibration as well as multiple harmonics in the latter stages of bearing wear [13, 14]. The  $f_f$  band was selected to control the unbalance [13, 15].  $f_f$  can be calculated according to Eq.(4)

$$f_f = N_{rpm} / 60 \quad (4)$$

The  $2f_f$  and  $3f_f$  band was introduced to detect misalignment of the shaft [16]. The next band, ( $f_{pad}$ )-band was chosen to monitor damage in the bearing itself. The predicted frequency in this case can be calculated with the following expression (Eq.(5)) [13].

$$f_{pad} = z_{pad} \cdot f_f \quad (5)$$

where  $z_{pad}$  is the number of bearing pads. Therefore the frequency range of this band depends on the type of bearing. The N-harmonics band was used to control possible bearing wear or excessive clearance on the bearings [13]. A band containing  $f_{pol}$  [13] was selected to detect electrical damage in the generator.

$$f_{pol} = 2 \cdot f_{line} \quad (6)$$

where  $f_{line}$  is the frequency of the electrical grid. Moreover, a band containing  $2f_{pol}$  is set to control the first harmonic of  $f_{pol}$ . Finally a band containing the RSI main frequency (usually  $2f_b$  and  $3f_b$  in RPT) was selected to monitor the hydraulic excitation. Nevertheless, this band can be easily controlled from the turbine bearing. The spectral band selection of this bearing can be summarized in Table 2.

### 2.4.2 Turbine bearing

For the turbine bearing a sub-harmonic band to detect bearing damage and cavitation rope was chosen [17-19]. The cavitation rope may appear in the draft tube (outlet of the turbine in generating mode) at low loads and generates low frequency pressure pulsations  $f_{vr}$ . In conventional hydraulic turbines many types of cavitation are possible but one of the most characteristics is the draft tube vortex rope

$$f_{vr} = (0.2 \div 0.3)f_f \quad (7)$$

The  $f_f$  band and the  $2f_f$ - $3f_f$  are selected again to control unbalance and misalignment of the shaft. If the machine is working around the best efficiency point the main hydraulic forces are generated by the RSI which is much more important than unbalance. The  $f_b$ - band contains the fundamental harmonic frequency of the RSI excitation. Although for conventional Francis turbines this is the predominant hydraulic frequency, for RPT machines sometimes this frequency is almost not detected.

The most important band to detect hydraulic problems and changes in the machine operating conditions is the  $n \cdot f_b$ - band. The calculation of  $n$  that gives the predominant frequency is explained in [3,4, 20] and in chap. 2.2 of this paper. As mentioned before this  $n$  is usually 2 or 3 for a RPT. Higher harmonics of the  $f_b$  band ( $f_b$ -harmonics) was also selected to control changes in the RSI characteristic.

Furthermore, a middle band is considered to detect turbulence or possible excited natural frequencies of the rotor[13]. Also, a final band is taken into account to monitor possible cavitation frequencies[13, 19]. The band selection for the turbine bearing can be summarized in Table 3.

In Figure 5 an example of band selection based on these criteria for the RPT-1 is shown.

### **3. Vibration levels**

#### **3.1 Overall levels**

The specification of overall vibration alarm levels in rotating machinery is not a simple task. This work is carried out by international committees where professionals with expertise are establishing the reference criteria. ISO-10816-5 Standards “Machine sets in hydraulic power generating and pumping plants Group 3” are the ones more related to RPT, because RPT are vertical shaft machines with bearing housings normally all braced against the foundation. To be in zone A and B of these standards (unlimited operating time) overall levels should be below 2.6 mm/s rms. If vibration level is more than 4 mm/s RMS would be in zone D what is normally considered to be of sufficient severity to cause damage to the machine. Compared with Standards some of the machines analysed had larger amplitudes due to RSI with overall levels higher than 2.6mm/s rms. Even with these high levels machines have been in operation for many years without serious problems. Therefore the overall alarm and trip levels have to be selected according to the machine design and operating mode.

Before selecting levels it is convenient to know the vibration characteristics of these machines. One of the characteristics is that vibration levels are generally larger in the turbine bearing than in the generator bearings because the main excitation RSI acts on the runner and the maximum response is in this bearing. In Figure 6, the trending of overall vibration levels in all bearings of two RPT with different design is shown. Comparing the amplitudes it can be seen that vibration levels are much higher in the turbine bearing than in the other bearing. Another characteristic is that vibration levels depend on the operating mode (as a turbine or as a pump) and on the delivered power (Figure 7).

Consequently when a new unit starts to run, before selecting the alarm and trip levels some preliminary measurements have to be performed both in turbine and in pumping mode. Unless the machine operates at very low load, the reference values have to be extracted from the measurements taken when machine is at maximum power. This is because the main excitation force (RSI) is at maximum when the machine is at full power. Only vibration levels measured with the machine operating at the same power can be compared and trended.

#### **3.2 Spectral band levels**

For a more accurate monitoring the spectral bands indicated above (Section 2.4) were used. When selecting the alarm levels in each band it has to be known if any change in levels is due to operating conditions or to damage. Therefore band vibration levels have to be trended for a determined machine power. Normally one data base is for vibrations measured at full power, one for vibrations measured around nominal power and another for vibrations measured at minimum power. For measurements taken with the machine operating with the same power, no significant changes in band levels are expected except for the RSI band. Changes in the other bands would be indicative of damage. Some examples are provided in the following paragraphs.

Vibration levels selected at first were of 0.2mm/s rms for the sub-synchronous bands and for unbalance 0.5mm/s rms in the turbine and 0.8mm/s rms in the generator. The RSI band had to be adapted to on-site measurements. In Figure 8, the vibration amplitudes at the RSI band for four machines of the same power plant have been represented. There are differences in vibration levels between machines even if they have the same design and installed in the same place. This is due to the fact that vibrations depend much on the structural response of the rotor; there are several natural frequencies around the excitation frequencies what may change the response

amplitude. Values between 0.3mm/s rms and 0.5mm/s rms were selected for the other bands.

#### **4. Analysis of vibration signatures after overhauls.**

The analysis of vibration signals before and after a repair is a good way to calibrate the symptoms and the alarm limits. Some typical cases of damage found are indicated in Table 4 and commented in the following paragraphs.

##### **4.1 Damage in generators**

The typical symptom of damage is vibration at two times the line frequency (100Hz). In Figure 9, the evolution of damage in the generator of a RPT has been represented. Changes in the vibration signature (spectra) and in the spectral bands can be observed. It can be seen that the band around two times the line frequency (100Hz) is increasing while the amplitudes of the other bands do not change.

##### **4.2 Wear in Bearings**

Wear in bearings is another typical damage. Families of sub-harmonics and harmonics appear in the spectrum (Figure 10). It can be seen that sub-harmonic and harmonic bands increase when damage worsens while the amplitude of the 100Hz spectral band remains the same. After repair all levels go down.

##### **4.3 Damage in bearing support**

Due to the large hydraulic excitation forces in the turbine all the machine bearings withstand large forces. In some cases these forces produce damage in bearings or in structure. In Figure 11 the vibrations generated by a reduction of stiffness in a radial bearing due to grouting damage can be observed. This is a very dangerous situation because the rotor natural frequencies decrease to the rotating frequency.

##### **4.4 Draft tube cavitation vortex detection**

When the turbine is operating off-design at part loads a cavitation vortex is generated at the exit of the turbine in the draft tube. This vortex generates low frequency pressure pulsations that propagate through the whole hydraulic system. These pressure pulsations can be amplified by hydraulic resonance generating large fluctuations in the turbine discharge producing instabilities in the operation of the turbine (Figure 12).

##### **4.5 Thrust bearing damage**

Damage in thrust bearing is another type of damage found. The symptom is the appearance of a vibration at the pad passing frequency. In Figure 13 the vibration signature measured axially in the thrust bearing has been represented. The pad passing frequency can be detected in the signature ( $f_{pad} = Z_{pads} \cdot f_r = 8 \cdot 6.25 = 50$  Hz) as well as the symptoms of damage in the generator.

##### **4.6 Typical unbalance after overhaul**

One of the advantages of remote vibration monitoring is to see if the machine is in good operating condition after an overhaul. In Figure 14 it can be seen that after a repair of the generator the unbalance level had an important increase. After balancing the levels decreased as expected.

## 4.7 Damage in runners

Some typical types of damage have been easily detected but a few of them, especially the ones occurring in the runner, were hardly detectable [6].

There are several reasons that make the runner prone to have damage. First, the runner is the part of the turbine that receives directly the strong pressure pulsations generated by RSI, second the maximum deformations in the runner occur at the external diameter in axial direction and third the excitation frequencies of RSI are in the range of the impeller natural frequencies. The runner vibrations can hardly be detected by monitoring vibrations in the bearing because natural frequencies of the runner do not produce important deformations in the rotor. In Figure 15, the deformations in a runner and rotor when excited by RSI have been represented. The numerical simulation done with FEM including the mass of water was checked experimentally. It can be observed that while the runner has large deformations, the rotor is very little affected. The strong deformations generate large stresses in the fillets of the crown/blade connections where fatigue damage can be caused. Some cases have been found during monitoring.

As indicated above, new renewable power plants (NRE) like wind generate intermittent energies what can destabilize the electrical grid. With rapidly expansion of NRE generation large hydro has to ensure grid stability. With this new scenario, in the last ten years the number of start-stops of these machines has increased dramatically. The consequence is an increase in damage in the motor/generator and in the turbine runner due to fatigue.

The cracks are not detectable until the crack causes a part of the runner to break. Even then the symptoms (changes in unbalance and RSI excitation) can be so small that can go unnoticed by the monitoring system. In Figure 16 the change in unbalance produced by a broken runner is shown. The unbalance level with the damaged runner is still under the alarm limit. More sophisticated methods are required to detect runner vibration.

## 5. Conclusions

In this paper the vibration monitoring of large reversible pump-turbines (RPT) units have been analysed. The results obtained after 15 years of monitoring several power plants with this type of machines have been used for the analysis. The machines were monitored locating accelerometers in all the bearings and in some cases pressure transducers and proximity probes. Several acquisition systems were installed in each power plant (one for each machine) which are remotely connected to a diagnostic center.

The main excitation forces were analysed. The vibration behaviour of RPT is quite different compared with conventional hydraulic turbines generating much larger vibration levels in the whole machine. The cause is the high pressure pulsations generated in the turbine by the interaction between the rotating blades of the impeller and the stationary vanes of the distributor named rotor/stator interaction (RSI). The method used to calculate the main excitation frequency that depends on the combination of runner blades and guide vanes is explained in the paper.

For a deeper analysis of the RSI effects, vibration signatures measured in machines of different design (manufacturers) are introduced and commented. It is shown that the vibration levels of RPT are very dependent of the operating conditions. Vibration levels are at maximum when the machine is operating in turbine mode at maximum load.

The procedure for the selection of the spectral bands and alarm levels used in RPT monitoring is indicated. The band upper frequency and the band lower frequency are



indicated depending on the rotating speed and on the number of runner blades and guide vanes.

Some of the most typical types of damage found in these machines during these years of monitoring have been represented. Damage in generator, in radial and thrust bearings, cavitation, damage in structure and unbalance are introduced. The symptoms and the changes in the spectral band levels for all these types of damage are included.

Due to the strong entrance of wind power supplying intermittent energy, the number of start-ups in these machines has increased dramatically affecting especially the runner where fatigue damage is not uncommon. The runner vibrations can hardly be detected by monitoring vibrations in the bearing because runner vibrations do not produce important deformations in the rotor. The number of breakdowns has increased and better vibration monitoring procedures are necessary especially to monitor the runner.

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## Figures

Figure 1: Monitoring positions and system.

Figure 2: Monitored RPT units.

Figure 3: Comparison of vibration signatures a) Francis turbine. b) RPT-2.

Figure 4: Vibration signatures measured in two RPT for the Turbine bearing in turbine operation (a) for Turbine bearing in pump operation (b) and for Generator bearing in Turbine operation(c).

Figure 5: Spectral bands selection for RPT-1.

Figure 6: Trending of overall and band levels of vibrations measured in all machine radial bearings.

Figure 7: Variation of vibration levels with distributor opening (machine power)

Figure 8: Variation of levels with time at constant power.

Figure 9: Damage in generator.

Figure 10: Wear in bearings.

Figure 11: Damage in bearing support.

Figure 12: Draft tube cavitation.

Figure 13: Damage in thrust bearing pads.

Figure 14: Unbalance detected after an overhaul.

Figure 15: Deformation in a RPT runner in resonance conditions.

Figure 16: Runner damage.

## **Tables**

Table 1: Main characteristic of two PT units and predicted predominant frequency.

Table 2: Spectral band selection for the generator bearing

Table 3: Spectral band selection for the Turbine Bearing

Table 4: Main types of damage and symptoms

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The highest vibration level in RPT units comes from the pressure fluctuation induced by the rotor stator interaction (RSI) [3,6,7,12]. Basically, RSI arises from the interference between the rotating blades of the runner and the stationary guide vanes.

When the rotating blades of the rotor pass in front of the static vanes of the stator (see [4] for figure details) the pressure field in the gap between blades and vanes can be described as the superposition of all the combinations  $m, n$  [3, 4]:

$$p_{mn}(\theta, t) = A_{mn} \cdot \cos(mZ_v\theta_s + \phi_m) \cdot \cos(nZ_b\theta_r + \phi_n) \text{ for } m = 1, 2, \dots, \infty \quad n = 1, 2, \dots, \infty \quad (1)$$

In Eq.(1),  $\theta$  is the angular coordinate (index  $s$  is for stationary coordinate and  $r$  for rotating coordinate),  $Z_v$  is the number of guide vanes in the stationary part and  $Z_b$  the number of rotating blades in the runner.  $m, n$  are entire numbers that represent the order of harmonic. The amplitude of the excitation  $A_{mn}$  depends on several parameters like the head (difference in altitude between the upper and the lower reservoir), position of the distributor (operating point), design of the machine and order of harmonics  $m, n$ .

Analysing this equation in detail and rewriting it in the stationary reference frame [3, 4], the following main conclusions can be extracted:

- Due to this interaction, several harmonic frequencies are excited. Viewed from the stationary frame, these frequencies depend on the rotating speed of the rotor (runner)  $f_f$ , the number of rotating blades  $Z_b$  and the order of harmonic  $n$

$$f_n = n \cdot Z_b \cdot f_f \quad (2)$$

- The blade passing frequency is defined as  $f_{n=1} = f_b = Z_b \cdot f_f$
- The excitation shape corresponding to one excited frequency  $f_n$  is the superposition of several excitation modes that can be calculated with the following expression:

$$k = mZ_v - nZ_b \quad (3)$$

- Higher amplitudes are expected for lower  $|k|$  and lower harmonics  $m, n$ . Therefore and due to hydraulic design conditions of RPT's (number of rotating blades and guide vanes), usually the highest value  $A_{mn}$  is obtained for  $m=1$  and  $n=2, 3$ , which usually gives a  $k=2, 3$ .

- The sign of  $k$  indicates the rotation of the frequency pulsation. If it is positive the pressure pulsation rotates in the same direction than the runner and in the opposite when it is negative.

### 2.3 Analysis of the vibration signatures

For an effective condition monitoring the vibration signatures have to be known. RPT are not standard radial turbines (Francis), compared to them, RPT have less number of blades in the runner, higher heads and higher rotating speeds. Depending on the head and on the discharge, different phenomena can occur with significant changes in the vibration signatures.

In Figure 3, two vibration signatures are compared, one belonging to a standard Francis turbine (375rpm,  $Z_b=14$ ,  $Z_v=20$ ) and one belonging to a RPT (500rpm,  $Z_b=9$ ,  $Z_v=20$ ). In both cases vibration was measured in the turbine bearing. It can be seen that while the hydraulic excitation in the Francis turbine appears at the blade passing frequency  $f_b$  ( $k = 6$ ), in the RPT appears at  $2 \cdot f_b$  ( $k = 2$ ) with a much higher amplitude.

The main characteristics of the machines analysed in this study and predicted predominant frequency for each machine are indicated in Table 1. The predicted predominant frequency is calculated according to the RSI characteristic analysis (see chap. 2.2).

In Figure 4, the theoretical predominant frequency according to the RSI analysis is validated experimentally for RPT-1 and RPT-3. The higher vibrations occur at 140 Hz ( $2f_b$ ) in RPT-1 and at 168.75 Hz ( $3f_b$ ) in RPT-2. The amplitudes depends very much on the operating mode and on the power delivered by the machine Figure 4a. These amplitudes are at maximum for full power turbine operation and higher than in pump operation mode (Figure 4b). This makes the monitoring of RPT more complex, because it is necessary to distinguish between changes due to operating conditions and changes due to damage.

In the generator bearing (Figure 4c), vibration amplitudes are lower and signatures are different. In this case the predominant frequency is  $f_f$  (unbalance), although the hydraulic excitation due to RSI can also be detected.

### 2.4 Selection of spectral bands and levels

Like in other rotating machinery properly specified spectral alarm bands are one of the most critical tools to detect potentially serious problems in hydraulic turbines. Spectral bands were selected for the monitoring of RPT after a detailed analysis of the excitation forces and of the dynamic characteristics of the machines. To properly set up the spectral bands it is necessary to understand what problems are detectable by vibration analysis in each measuring position. It is known that mass unbalance, misalignment, bearing wear, hydraulic forces and some electrical problems can be detected rather easily treating the vibration signals.

RPT are vertical shaft machines with three radial fluid film bearings connected to foundation and one vertical thrust bearing. The turbine and the generator are connected by a rigid coupling. The turbine bearing withstand mechanical and hydraulic forces while the two generator bearings withstand mechanical and electromagnetic forces. Spectral bands in the generator bearings and in the turbine bearing are selected according to each type of forces.

In order to distinguish between a normal operation and damage a properly specification of levels is necessary. The specification of band alarm levels is even more complex. Some alarm levels have to be set up and modified after some time of monitoring.

### 2.4.1 Generator bearing

A subharmonic band is selected to detect bearing damage. Damage in fluid film bearings may generate subsynchronous vibration as well as multiple harmonics in the latter stages of bearing wear [13, 14]. The  $f_f$  band was selected to control the unbalance [13, 15].  $f_f$  can be calculated according to Eq.(4)

$$f_f = N_{rpm} / 60 \quad (4)$$

The  $2f_f$  and  $3f_f$  band was introduced to detect misalignment of the shaft [16]. The next band, ( $f_{pad}$ )-band was chosen to monitor damage in the bearing itself. The predicted frequency in this case can be calculated with the following expression (Eq.(5)) [13].

$$f_{pad} = z_{pad} \cdot f_f \quad (5)$$

where  $z_{pad}$  is the number of bearing pads. Therefore the frequency range of this band depends on the type of bearing. The N-harmonics band was used to control possible bearing wear or excessive clearance on the bearings [13]. A band containing  $f_{pol}$  [13] was selected to detect electrical damage in the generator.

$$f_{pol} = 2 \cdot f_{line} \quad (6)$$

where  $f_{line}$  is the frequency of the electrical grid. Moreover, a band containing  $2f_{pol}$  is set to control the first harmonic of  $f_{pol}$ . Finally a band containing the RSI main frequency (usually  $2f_b$  and  $3f_b$  in RPT) was selected to monitor the hydraulic excitation. Nevertheless, this band can be easily controlled from the turbine bearing. The spectral band selection of this bearing can be summarized in Table 2.

### 2.4.2 Turbine bearing

For the turbine bearing a subharmonic band to detect bearing damage and cavitation rope was chosen [17-19]. The cavitation rope may appear in the draft tube (outlet of the turbine in generating mode) at low loads and generates low frequency pressure pulsations  $f_{vr}$ . In conventional hydraulic turbines many types of cavitation are possible but one of the most characteristics is the draft tube vortex rope

$$f_{vr} = (0.2 \div 0.3)f_f \quad (7)$$

The  $f_f$  band and the  $2f_f$ - $3f_f$  are selected again to control unbalance and misalignment of the shaft. If the machine is working around the best efficiency point the main hydraulic forces are generated by the RSI which is much more important than unbalance. The  $f_b$ -band contains the fundamental harmonic frequency of the RSI excitation. Although for conventional Francis turbines this is the predominant hydraulic frequency, for RPT machines sometimes this frequency is almost not detected.

The most important band to detect hydraulic problems and changes in the machine operating conditions is the  $n \cdot f_b$ -band. The calculation of  $n$  that gives the predominant frequency is explained in [3,4, 20] and in chap. 2.2 of this paper. As mentioned before this  $n$  is usually 2 or 3 for a RPT. Higher harmonics of the  $f_b$  band ( $f_b$ -harmonics) was also selected to control changes in the RSI characteristic.

Furthermore, a middle band is considered to detect turbulence or possible excited natural frequencies of the rotor [13]. Also, a final band is taken into account to monitor possible cavitation frequencies [13, 19]. The band selection for the turbine bearing can be summarized in Table 3.



In Figure 5 an example of band selection based on these criteria for the RPT-1 is shown.

### **3. Vibration levels**

The specification of overall vibration alarm levels is not a simple task. This work is carried out by international committees where professionals with expertise are establishing the reference criteria. ISO-10816-5 Standards “Machine sets in hydraulic power generating and pumping plants Group 3” are the ones more related to RPT, because RPT are vertical shaft machines with bearing housings normally all braced against the foundation. To be in zone A and B of these standards (unlimited operating time) overall levels should be below 2.6 mm/s rms. If vibration level is more than 4 mm/s RMS would be in zone D what is normally considered to be of sufficient severity to cause damage to the machine.

In Figure 6, the trending of overall vibration levels in all bearings of two RPT with different design is shown. Comparing the amplitudes it can be seen that vibration levels are much higher in the turbine bearing than in the generator bearing. Some of the machines analysed have large amplitudes in the turbine bearing due to RSI with overall levels larger than 2.6mm/s rms. Even with these high levels machines have been in operation for many years without serious problems. Alarm and trip levels have to be selecting according to the machine design and data acquired.

When selecting the alarm levels it has to be known if any change in levels is due to operating conditions or to damage, so the influence of the operating conditions on the vibration amplitudes has to be known. In RPT vibration levels depend much on the power delivered by the machine what can be clearly observed in Figure 7. When the opening degree of the distributor is at maximum (maximum power) vibration levels are at maximum. Therefore for monitoring, vibration levels have to be related to machine operating conditions.

Vibration levels selected at first were of 0.2mm/s rms for the subsynchronous bands and for unbalance 0.5mm/s rms in the turbine and 0.8mm/s rms in the generator. The RSI band had to be adapted to experimental measurements. In Figure 8, the vibration amplitudes at the RSI band for four machines of the same power plant have been represented. There are differences in vibration levels between machines even if they have the same design and installed in the same place. This is due to the fact that vibrations depend much on the structural response of the rotor; there are several natural frequencies around the excitation frequencies what may change the response amplitude. Values between 0.3mm/s rms and 0.4mm/s rms were selected for the other bands.

### **4. Analysis of vibrations in overhauls.**

The analysis of vibration signals before and after a repair is a good way to calibrate the symptoms and the alarm limits. Some typical cases of damage found are commented in the following paragraphs. As indicated above, new renewable power plants (NRE) like wind generate intermittent energies what can destabilize the electrical grid. With rapidly expansion of NRE generation large hydro has to ensure grid stability. With this new scenario, in the last 10 the number of start-stops of these machines has increased dramatically. The consequence is an increase in damage in the motor/generator and in the turbine runner due to fatigue.

#### **4.1 Damage in generators**

The typical symptom of damage is vibration at two times the line frequency (100Hz). In Figure 9, the evolution of damage in the generator of a RPT has been represented.

Changes in the vibration signature (spectra) and in the spectral bands can be observed. It can be seen that the band around two times the line frequency (100Hz) is increasing while the amplitudes of the other bands do not change.

#### **4.2 Wear in Bearings**

Wear in bearings is another typical damage. Families of sub-harmonics and harmonics appear in the spectrum (Figure 10). It can be seen that sub-harmonic and harmonic bands increase when damage worsens while the amplitude of the 100Hz spectral band remains the same. After repair all levels go down.

#### **4.3 Damage in bearing support**

Due to the large hydraulic excitation forces in the turbine all the machine bearings withstand large forces. In some cases these forces produce damage in bearings or in structure. In Figure 11 the vibrations generated by a reduction of stiffness in a radial bearing due to grouting damage can be observed. This is a very dangerous situation because the rotor natural frequencies decrease to the rotating frequency.

#### **4.4 Draft tube cavitation vortex detection**

When the turbine is operating off-design at part loads a cavitation vortex is generated at the exit of the turbine in the draft tube. This vortex generates low frequency pressure pulsations that propagate through the whole hydraulic system. These pressure pulsations can be amplified by hydraulic resonance generating large fluctuations in the turbine discharge producing instabilities in the operation of the turbine (Figure 12).

#### **4.5 Thrust bearing damage**

Damage in thrust bearing is another type of damage found. The symptom is the appearance of a vibration at the pad passing frequency. In Figure 13 the vibration signature measured axially in the thrust bearing has been represented. The pad passing frequency can be detected in the signature ( $f_{\text{pad}} = Z_{\text{pads}} \cdot f_r = 8 \cdot 6.25 = 50 \text{ Hz}$ ) as well as the symptoms of damage in the generator.

#### **4.6 Typical unbalance after overhaul**

One of the advantages of remote vibration monitoring is to see if the machine is in good operating condition after an overhaul. In Figure 14 it can be seen that after a repair of the generator the unbalance level had an important increase. After balancing the levels decreased as expected.

#### **4.7 Damage in runners**

Some typical types of damage have been easily detected but a few of them, especially the ones occurring in the runner, were hardly detectable [6]. There are several reasons that make the runner prone to have damage. First, the runner is the part of the turbine that receives directly the strong pressure pulsations generated by RSI, second the maximum deformations in the runner occur at the external diameter in axial direction and third the excitation frequencies of RSI are in the range of the impeller natural frequencies. The runner vibrations can hardly be detected by monitoring vibrations in the bearing because natural frequencies of the runner do not produce important deformations in the rotor. In Figure 16, the deformations in a runner and rotor when excited by RSI have been represented. The numerical simulation done

with FEM including the mass of water was checked experimentally. It can be observed that while the runner has large deformations, the rotor is very little affected. The strong deformations generate large stresses in the fillets of the crown/blade connections where fatigue damage can be caused. Some cases have been found during monitoring.

The cracks are not detectable until the crack causes a part of the runner to break. Even then the symptoms (changes in unbalance and RSI excitation) can be so small that can go unnoticed by the monitoring system. In Figure 16 the change in unbalance produced by a broken runner is shown. The unbalance level with the damaged runner is still under the alarm limit. More sophisticated methods are required to detect runner vibration.

## 5. Conclusions

Hydraulic turbines used in pumped storage are reversible machines that can operate as a pump and as a turbine. They are fundamental for the stability of the electrical grid where consumption and generation have to be matched. They can start/stop rapidly absorbing the surplus of energy from the grid or supplying energy to the grid. Due to the strong entrance of wind power supplying intermittent energy, the number of start-ups in these machines has increased dramatically in the last ten years. The number of breakdowns has increased and vibration monitoring is more necessary than ever.

In this paper the vibration monitoring of large reversible pump-turbines (RPT) units have been analysed. The results obtained after 15 years of monitoring several power plants with this type of machines have been used for the analysis. The machines were monitored locating accelerometers in all the bearings and in some cases pressure transducers and proximity probes. Several acquisition systems were installed in each power plant (one for each machine) which are remotely connected to a diagnostic center.

The main excitation forces were analysed. The vibration behaviour of RPT is quite different compared with conventional hydraulic turbines generating much larger vibration levels in the whole machine. The cause is the high pressure pulsations generated in the turbine by the interaction between the rotating blades of the impeller and the stationary vanes of the distributor named rotor/stator interaction (RSI). This occurs because of the design characteristics of RPT that have low number of blades in the runner and operate under high heads. The method used to calculate the main excitation frequency that depends on the combination of runner blades and guide vanes is explained in the paper.

For a deeper analysis of the RSI effects, vibration signatures measured in machines of different design (manufacturers) are introduced and commented. It is shown that the vibration signatures of RPT are different from conventional hydraulic turbines and that the vibration levels are very dependent of the operating conditions. Vibration levels are at maximum when the machine is operating in turbine mode at maximum load; at minimum load vibrations levels can decrease to the half.

The procedure for the selection of the spectral bands used in RPT monitoring is indicated. The band upper frequency and the band lower frequency are indicated depending on the rotating speed and on the number of runner blades and guide vanes.

Some of the most typical types of damage found in these machines during these years of monitoring have been represented. Damage in generator, in radial and thrust bearings, cavitation, damage in structure and unbalance are introduced. The symptoms and the changes in the spectral band levels for all these types of damage are included.

Fatigue damage in the runners of these machines is not uncommon due to several reasons. The strong pressure pulsations of RSI are generated where the runner has

the maximum deformations with possible excitation of its natural frequencies. The runner vibrations can hardly be detected by monitoring vibrations in the bearing because natural frequencies of the runner do not produce important deformations in the rotor. New monitoring methods have to be used to monitor the runners of these machines.

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## **Figures**

Figure 1: Monitoring positions and system.

Figure 2: Monitored RPT units.

Figure 3: *Comparison of vibration signatures a) Francis turbine. b) RPT-2.*

Figure 4: *Vibration signatures measured in two RPT for the Turbine bearing in turbine operation (a) for Turbine bearing in pump operation (b) and for Generator bearing in Turbine operation(c).*

Figure 5: *Spectral bands selection for RPT-1.*

Figure 6: *Trending of overall and band levels of vibrations measured in all machine radial bearings.*

Figure 7: *Variation of levels with operating conditions.*

Figure 8: *Variation of levels with time at constant power.*

Figure 9: *Damage in generator.*

Figure 10: *Wear in bearings.*

Figure 11: *Damage in bearing support.*

Figure 12: *Draft tube cavitation.*

Figure 13: *Damage in thrust bearing pads.*

Figure 14: *Unbalance detected after an overhaul.*

Figure 15: *Deformation in a RPT runner in resonance conditions.*

Figure 16: *Runner damage.*

## **Tables**

Table 1: *Main characteristic of two PT units and predicted predominant frequency.*

Table 2: *Spectral band selection for the generator bearing*

Table 3: *Spectral band selection for the Turbine Bearing*

Figure 1  
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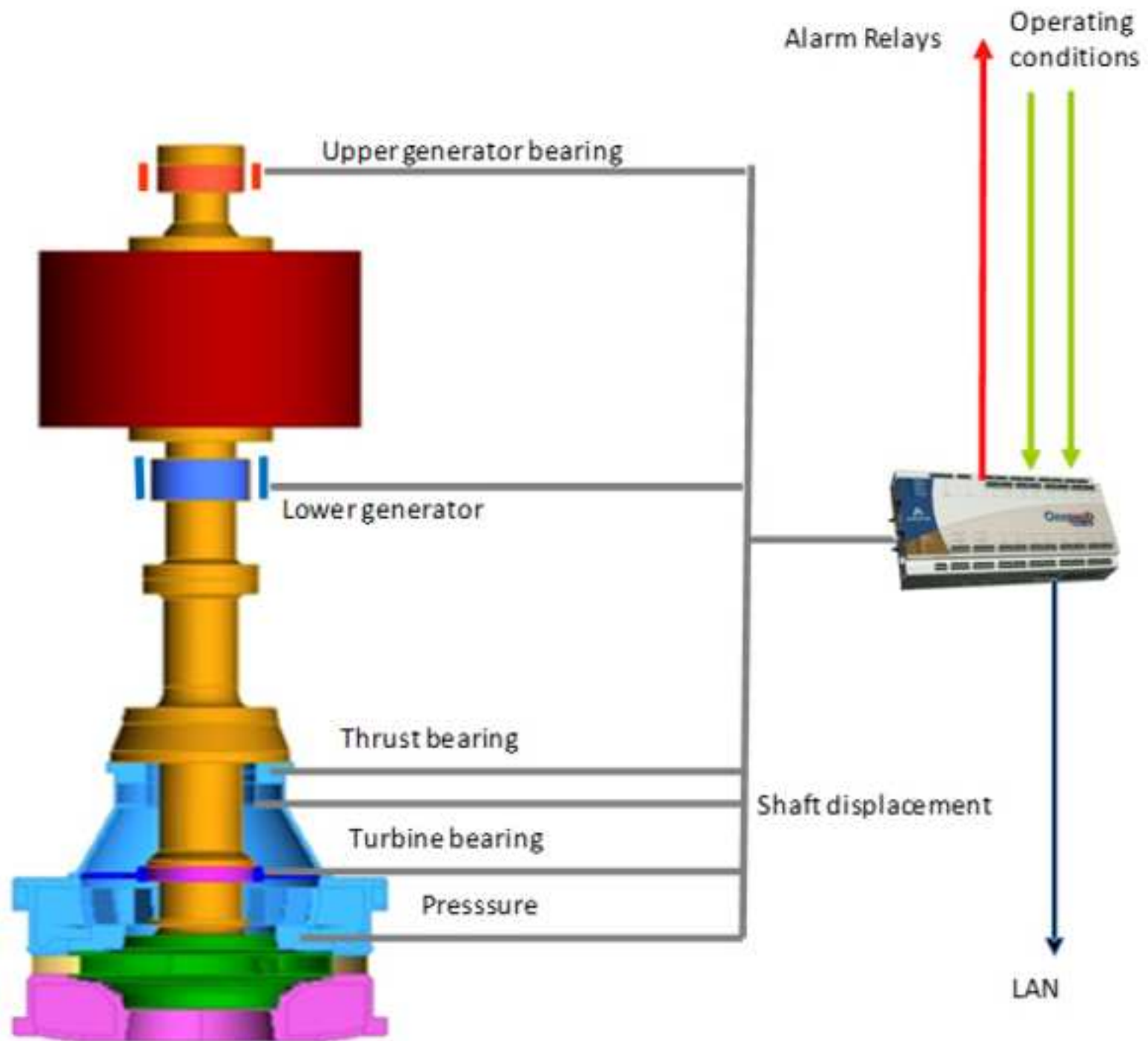


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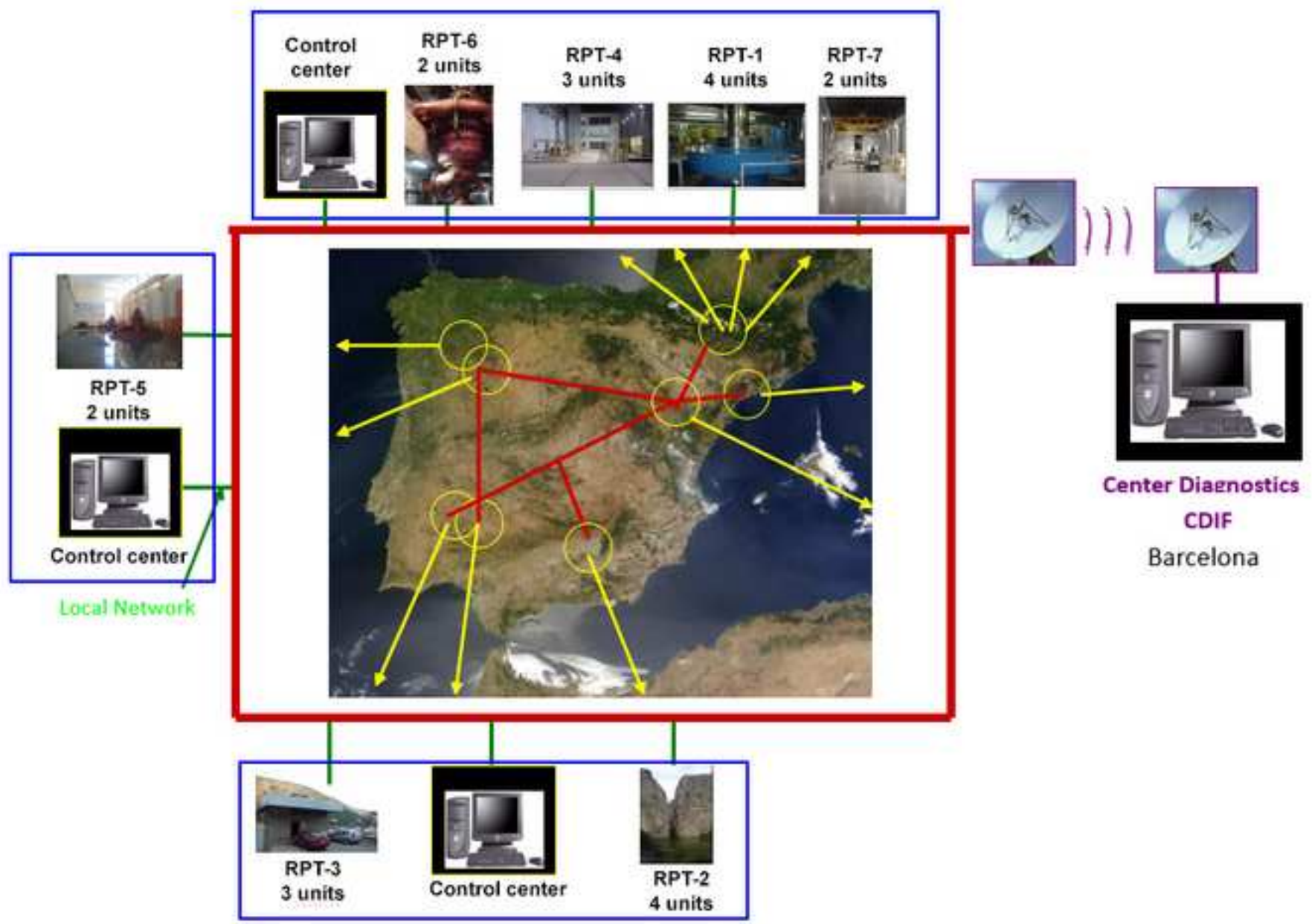


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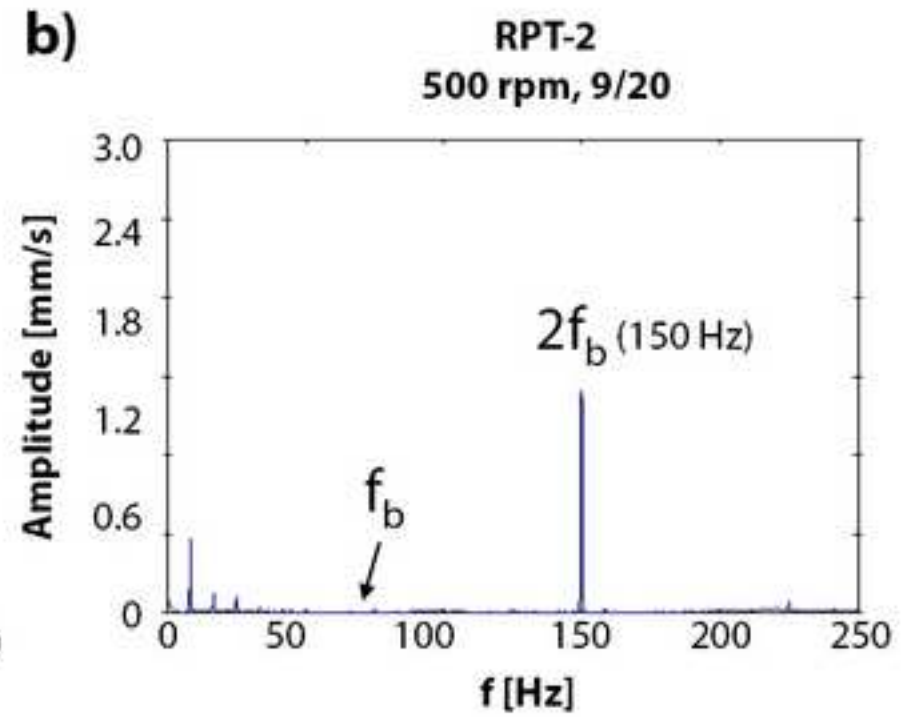
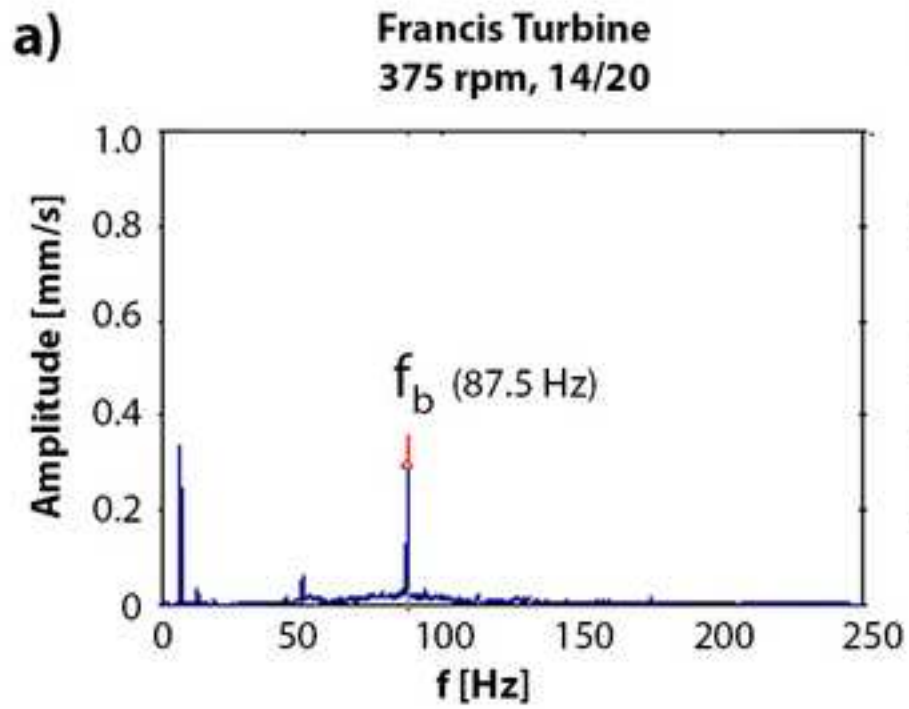




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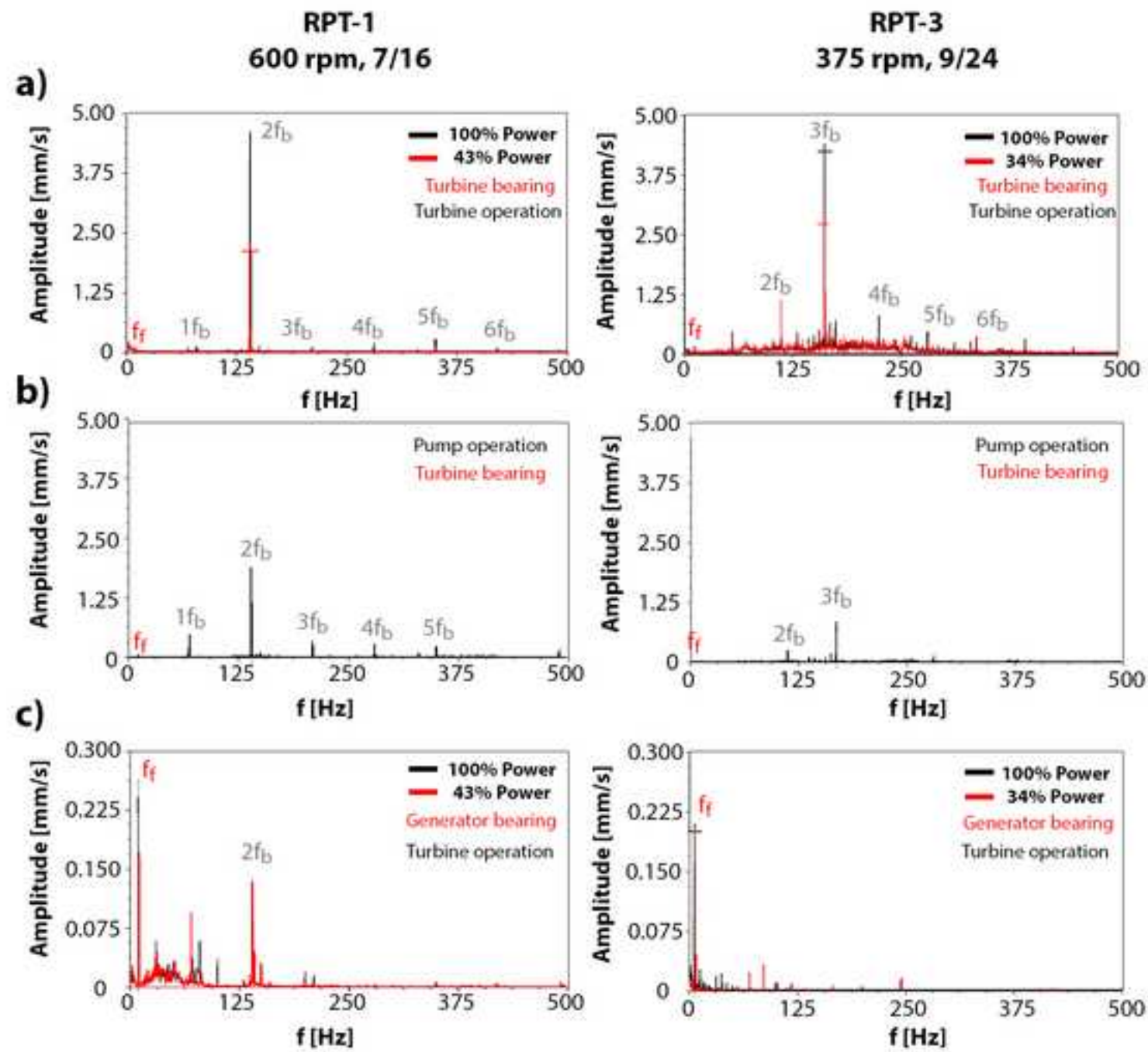


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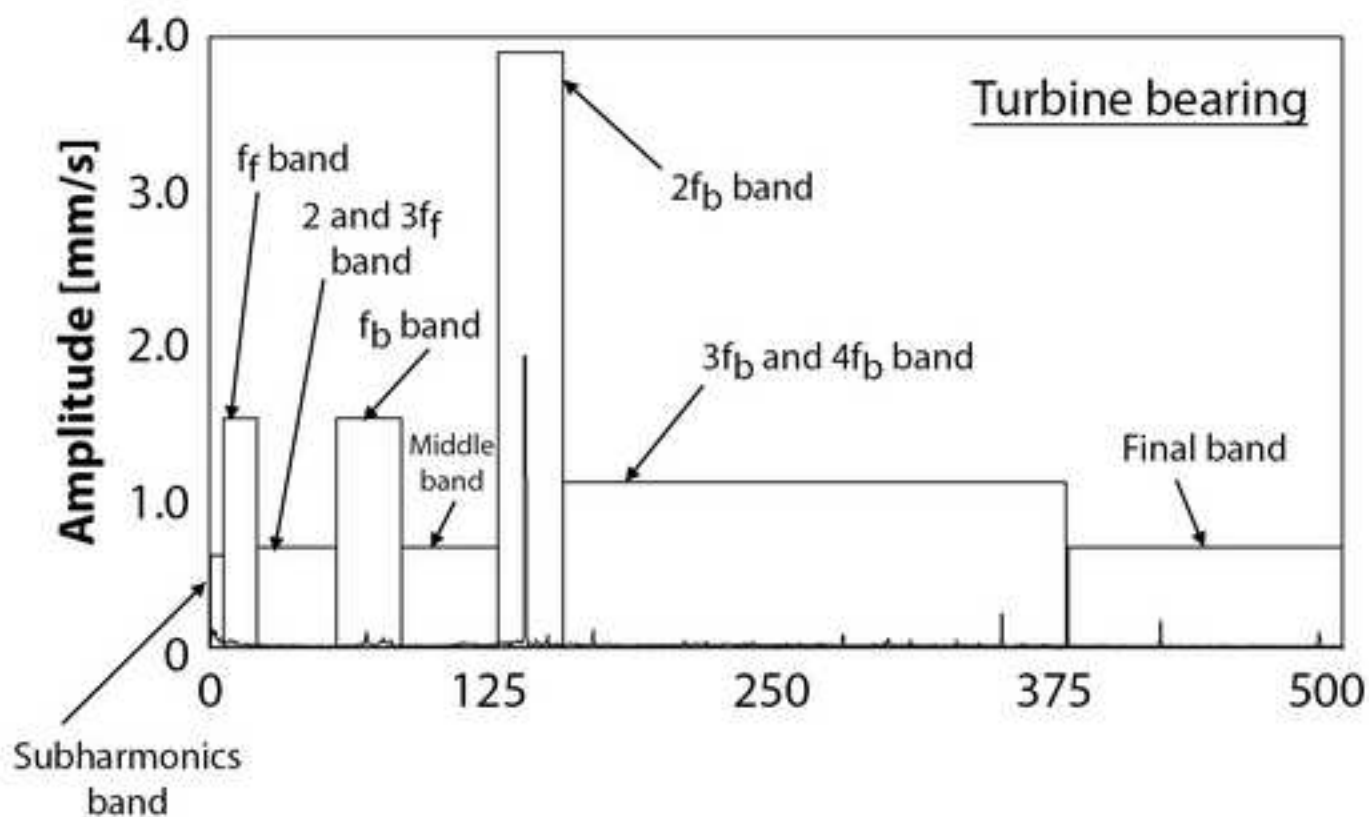
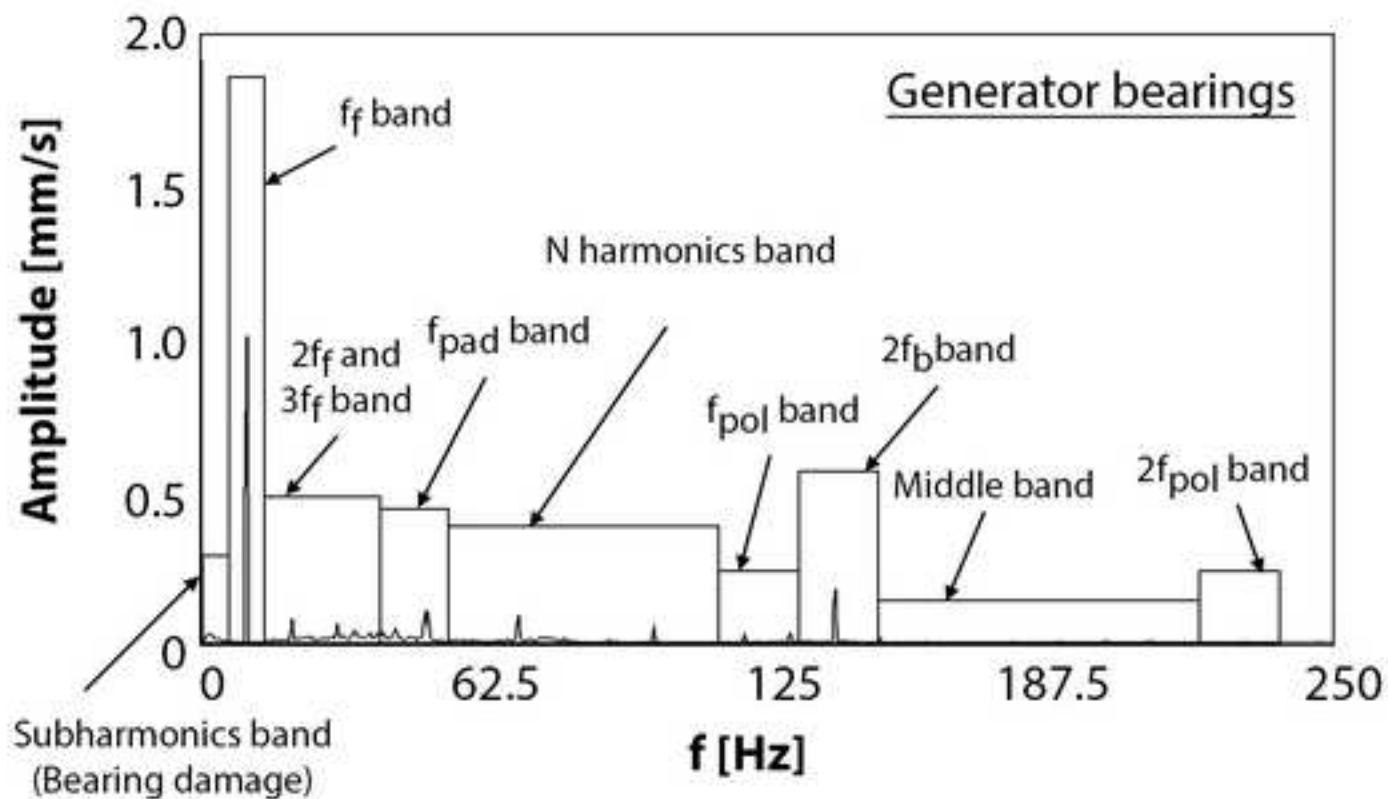


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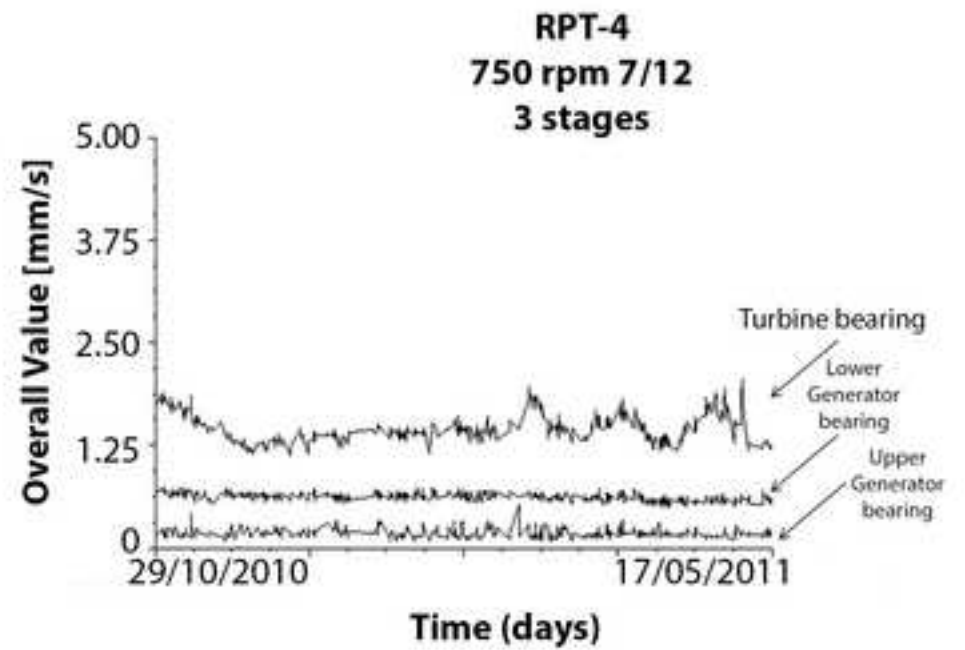
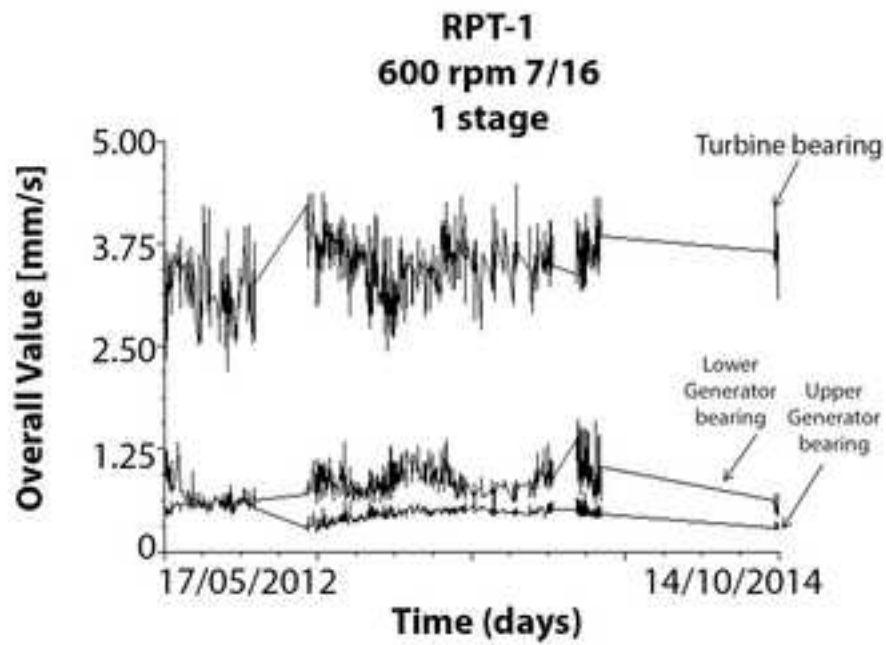


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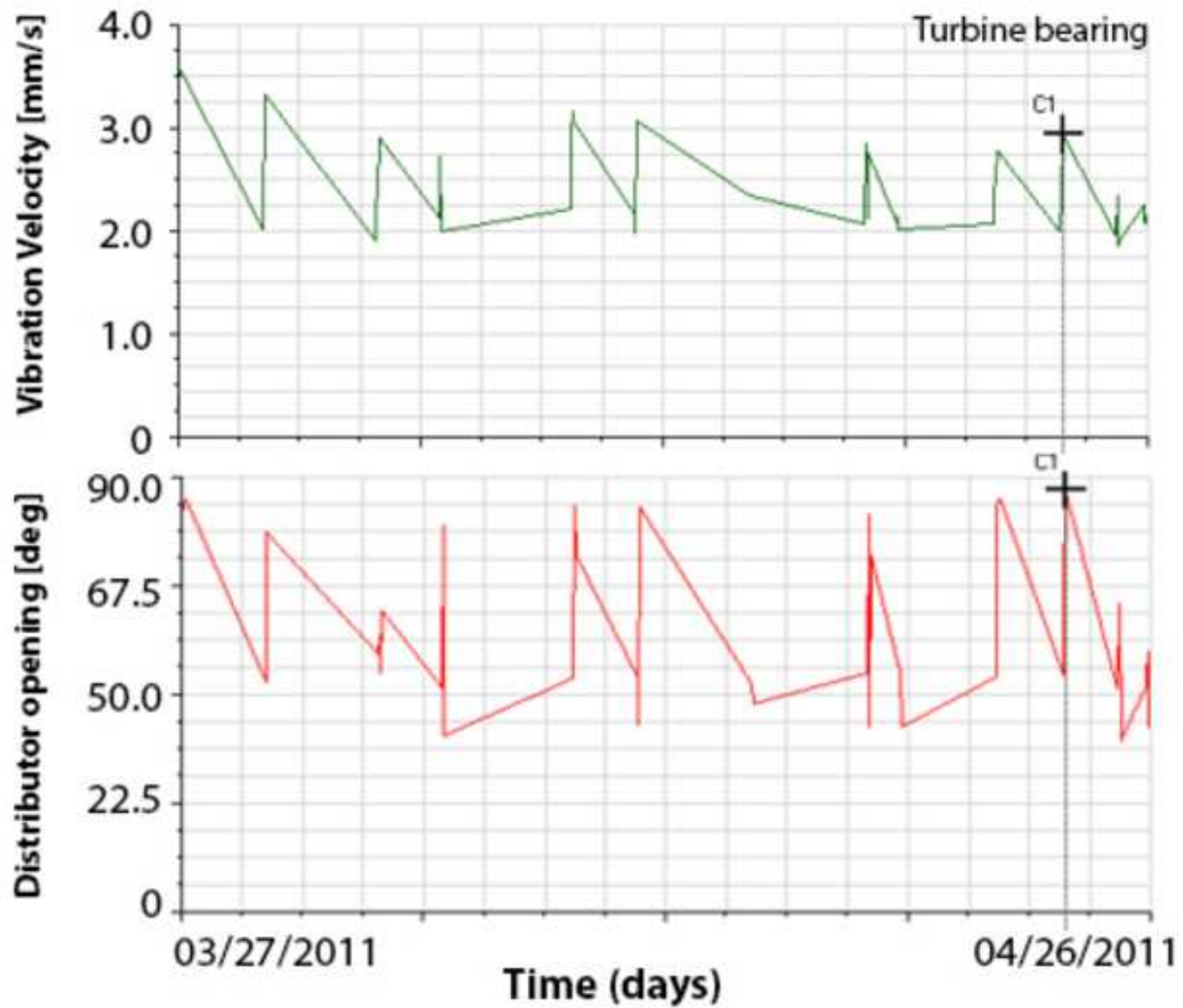


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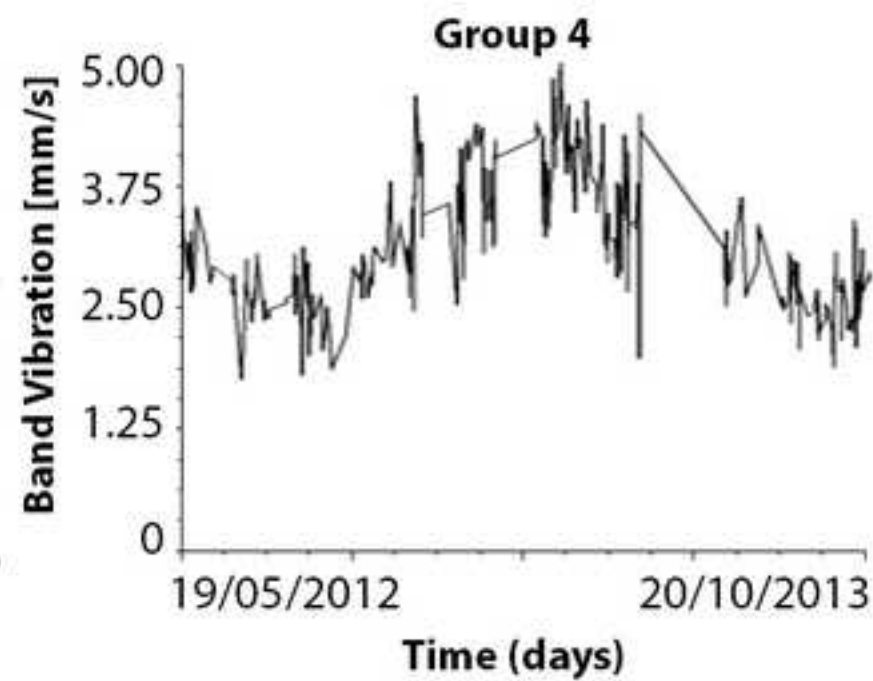
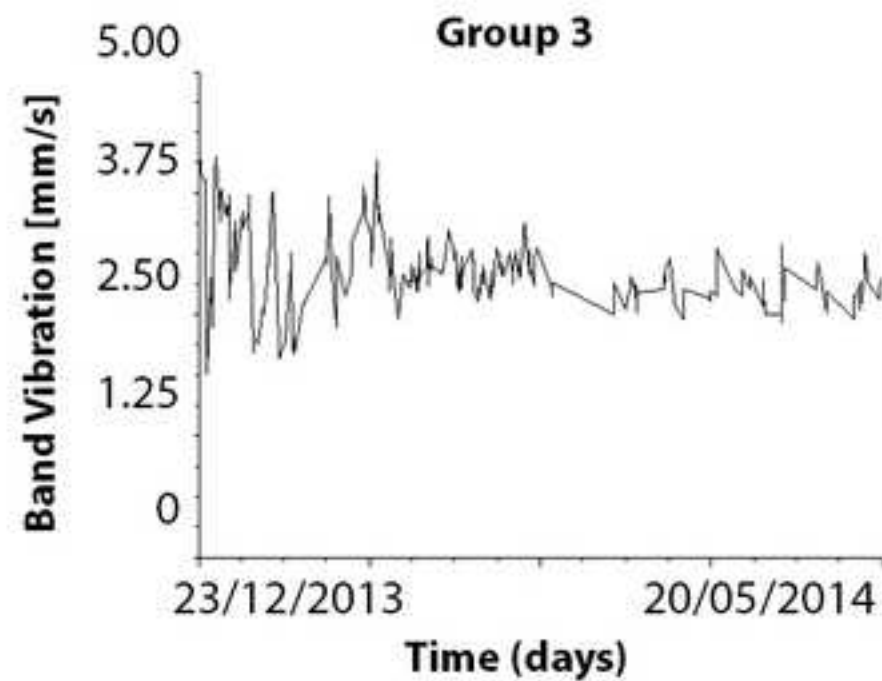
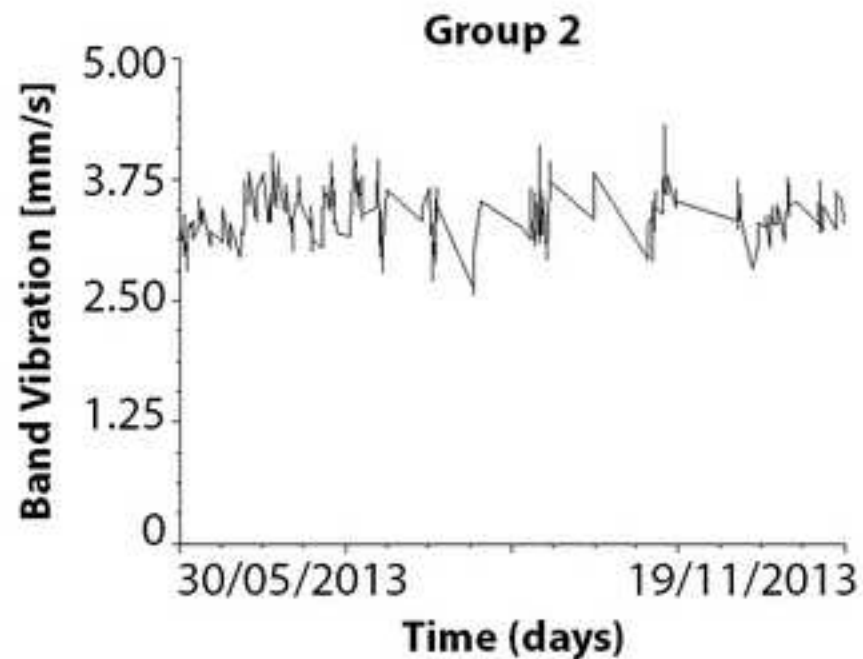
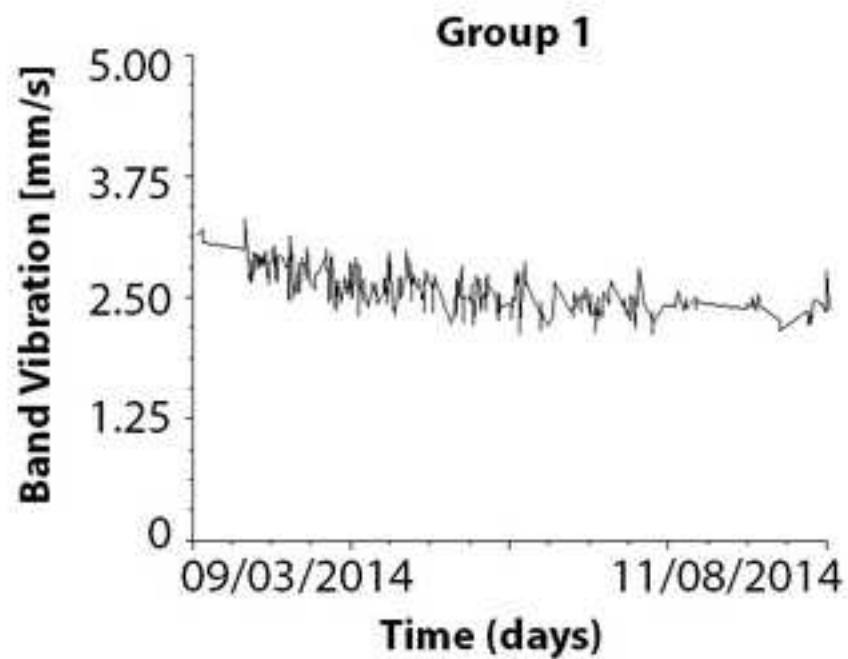




Figure 9  
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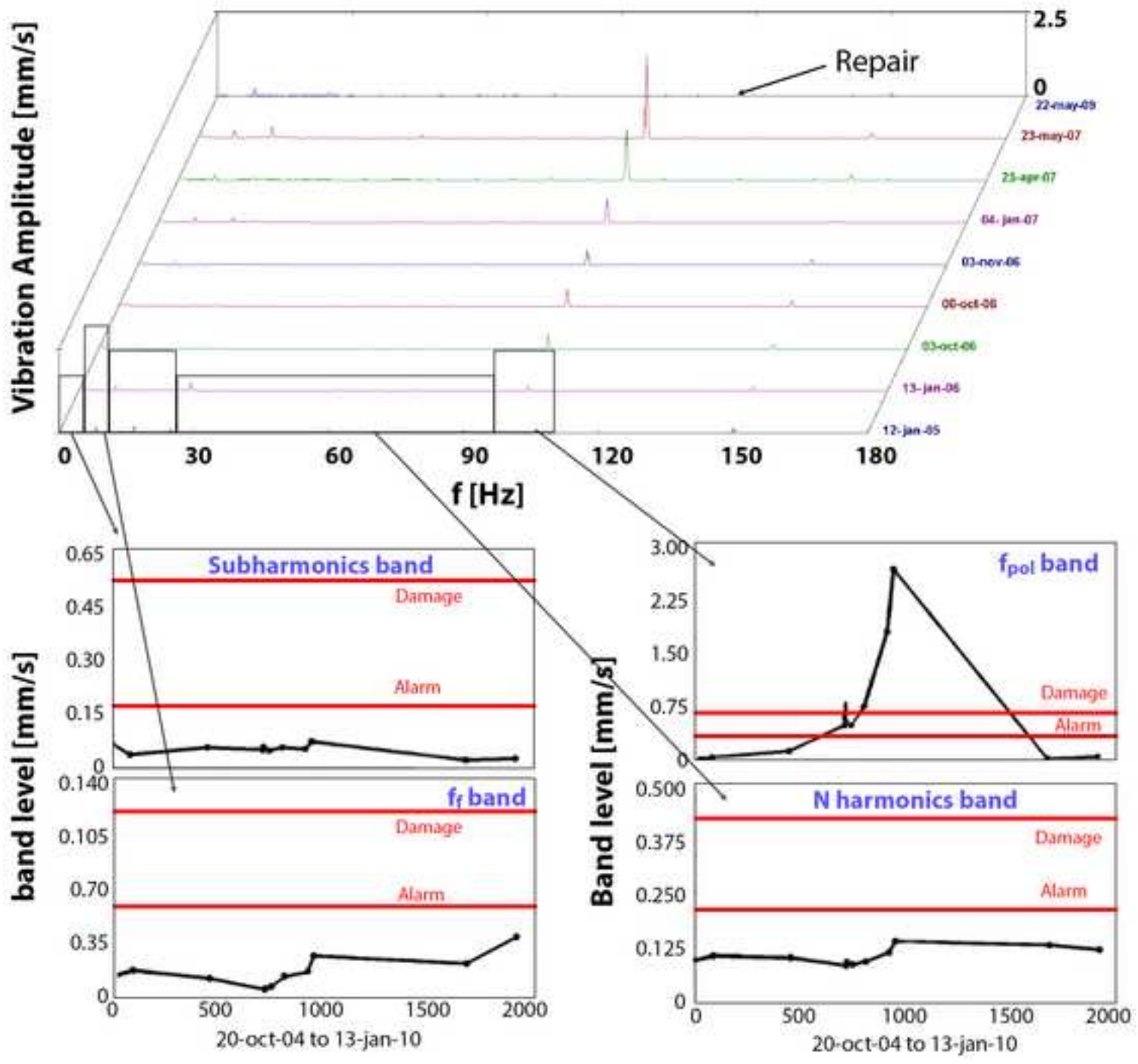


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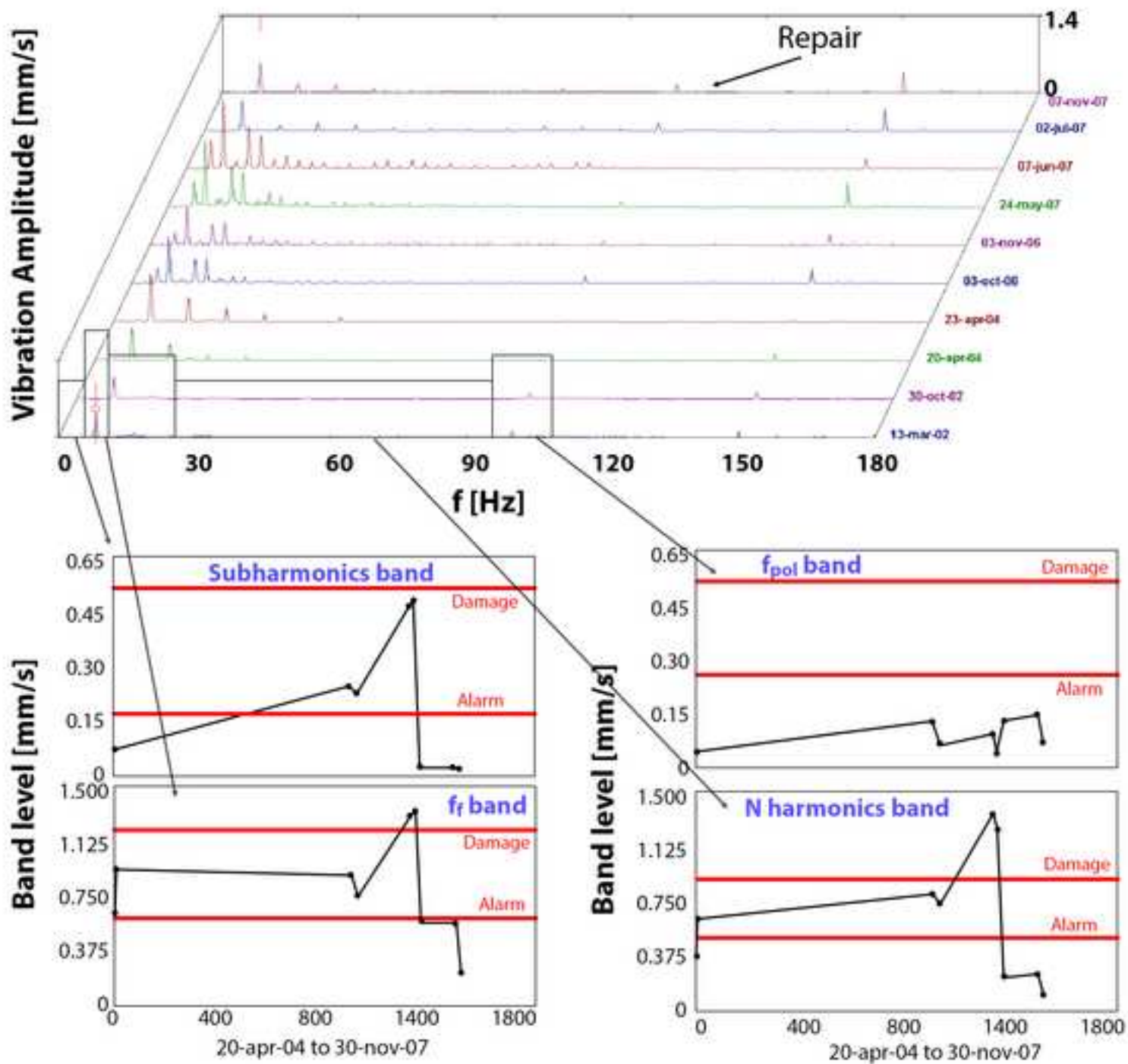


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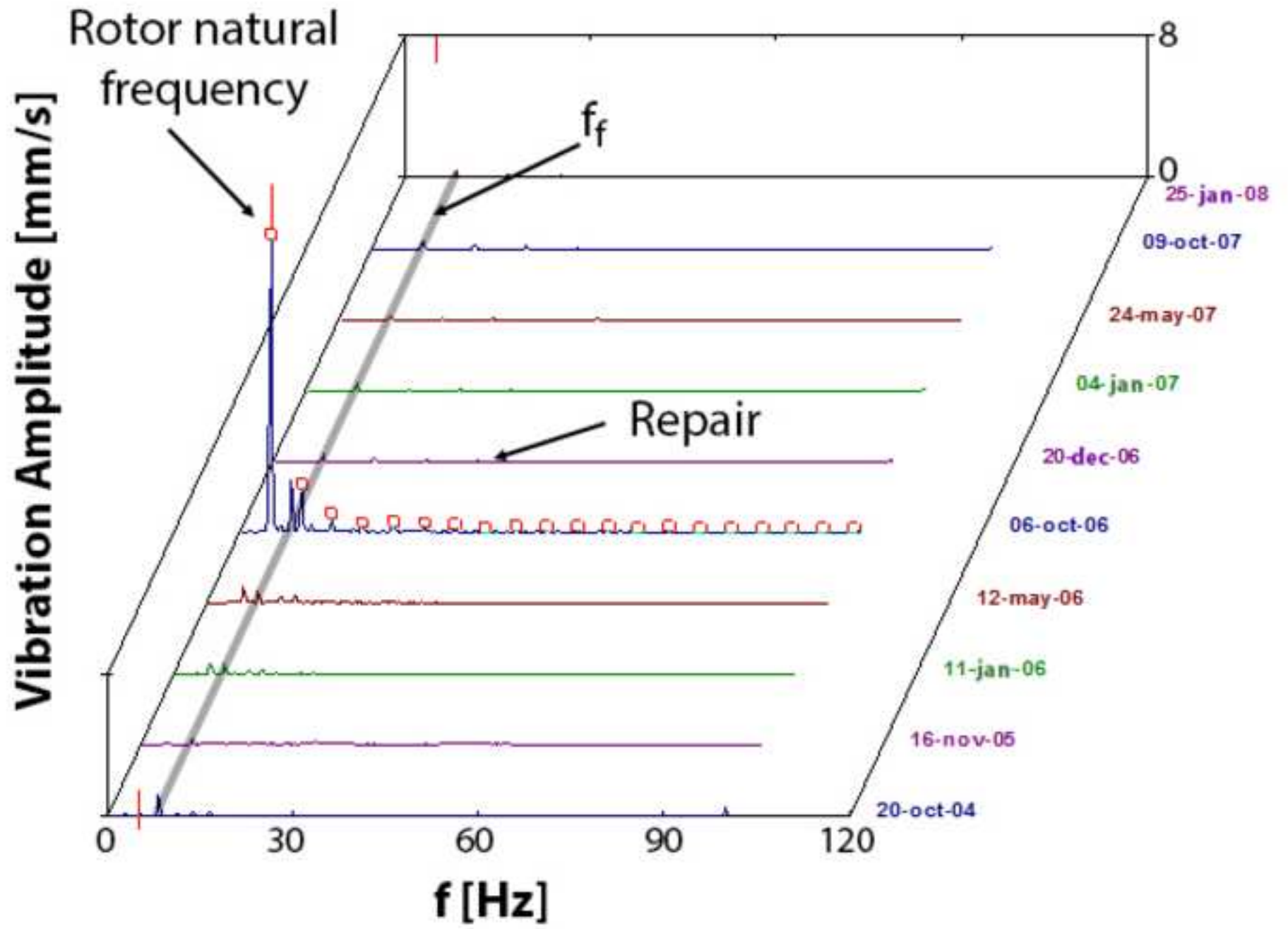




Figure 12

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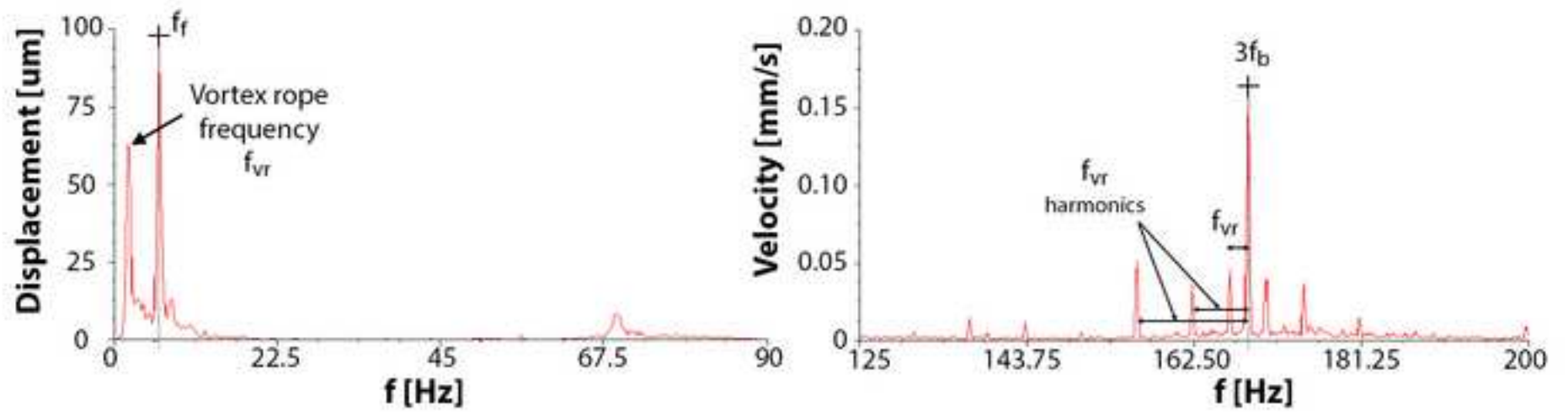


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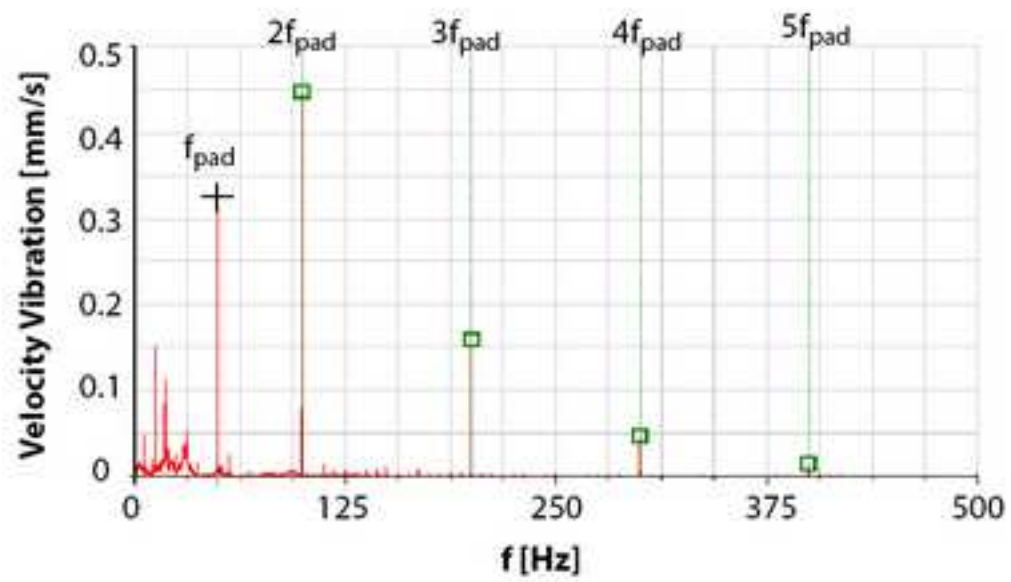


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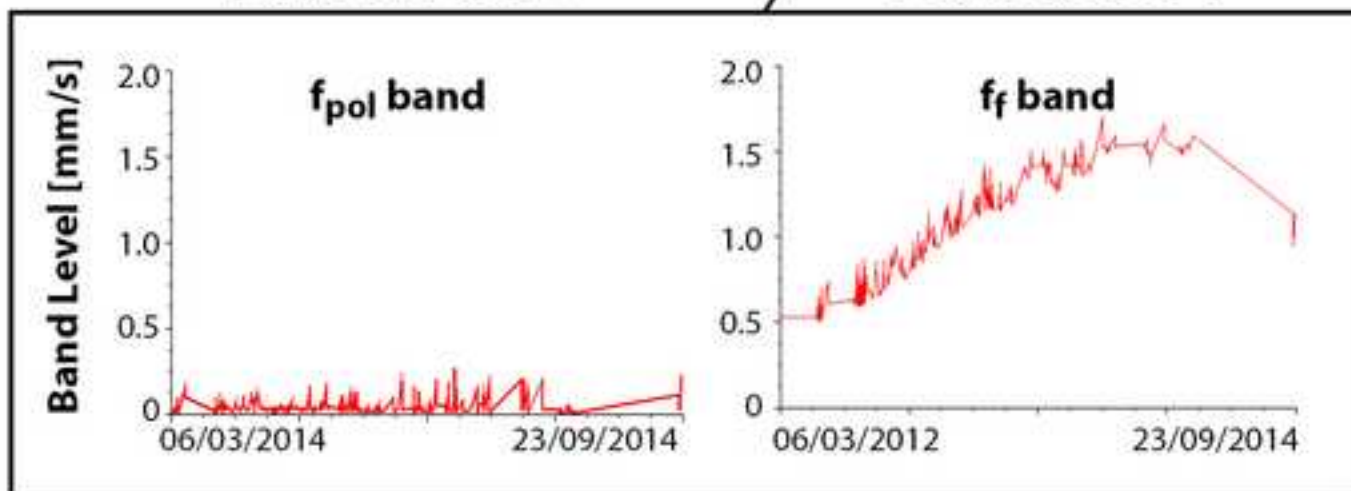
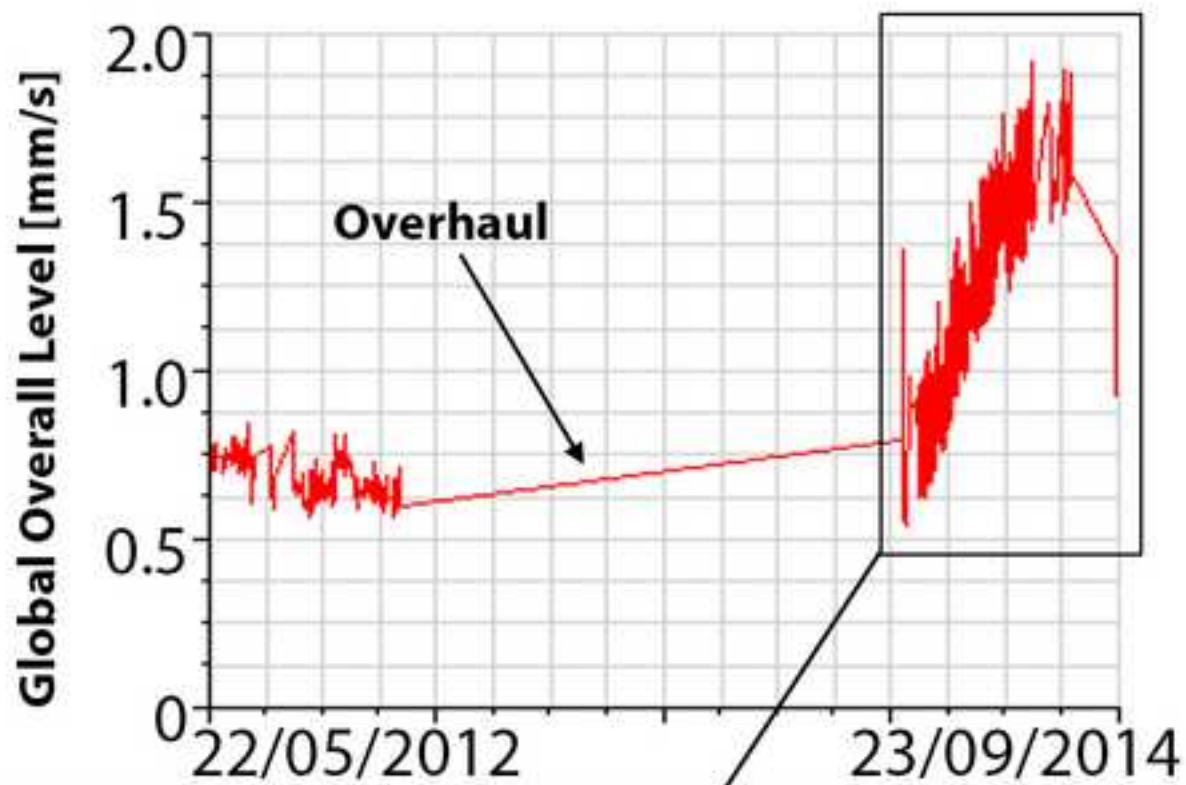


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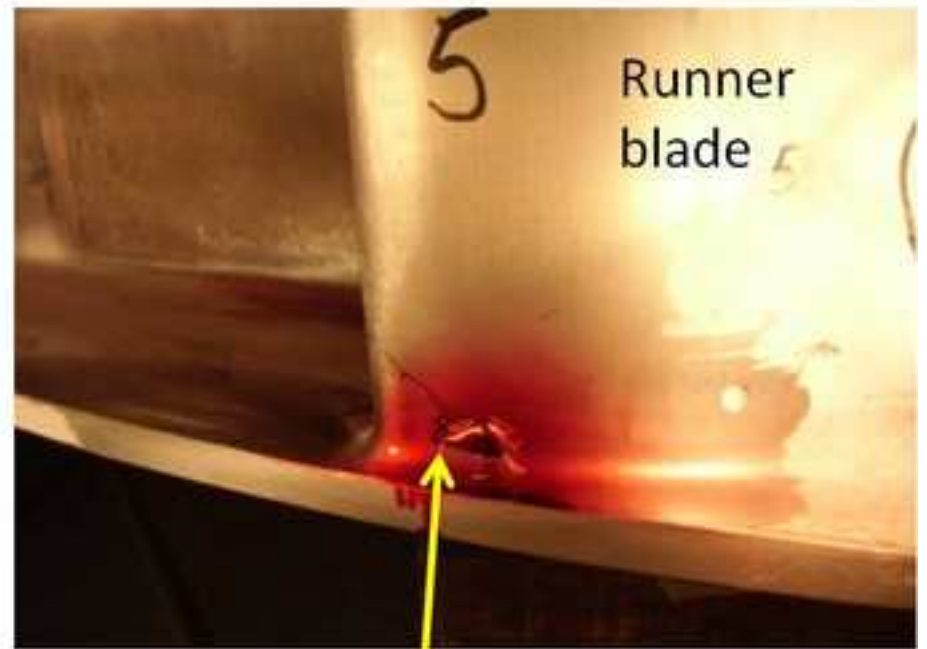
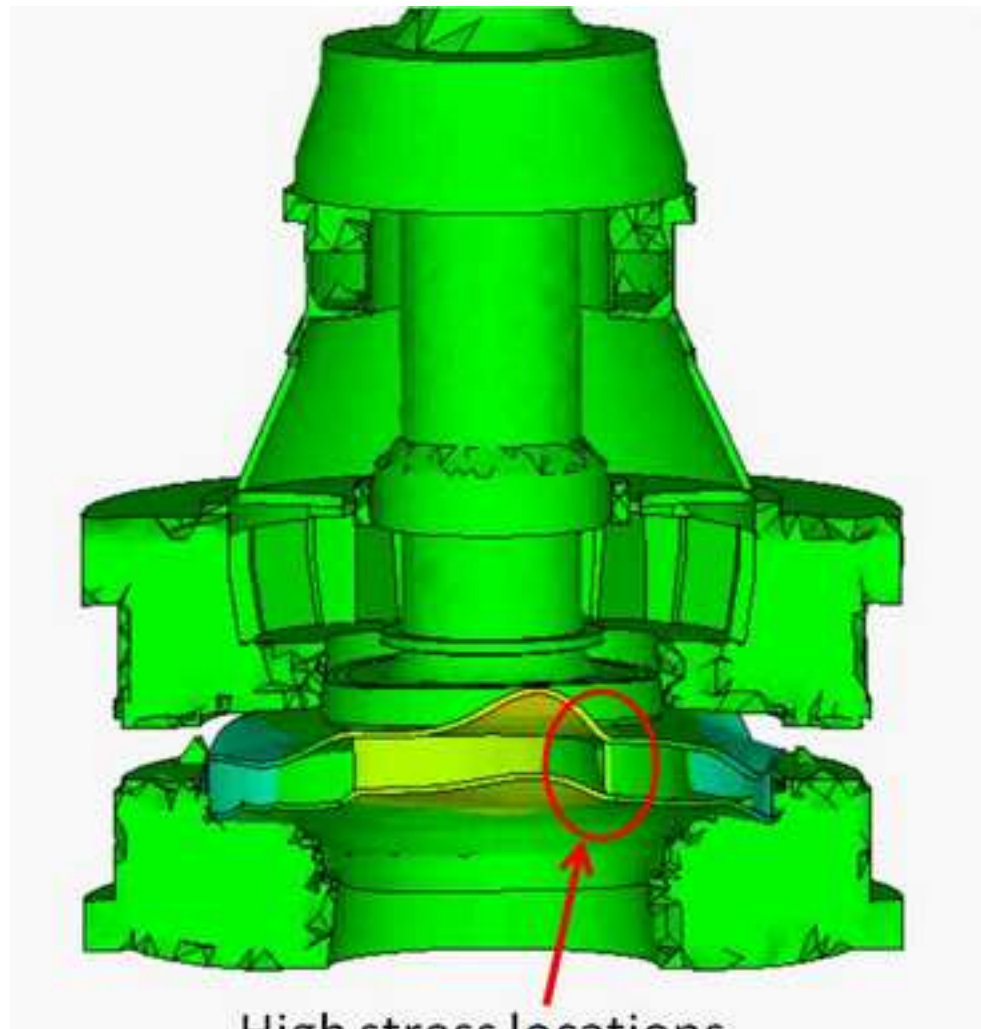


Figure 16  
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Runner damage

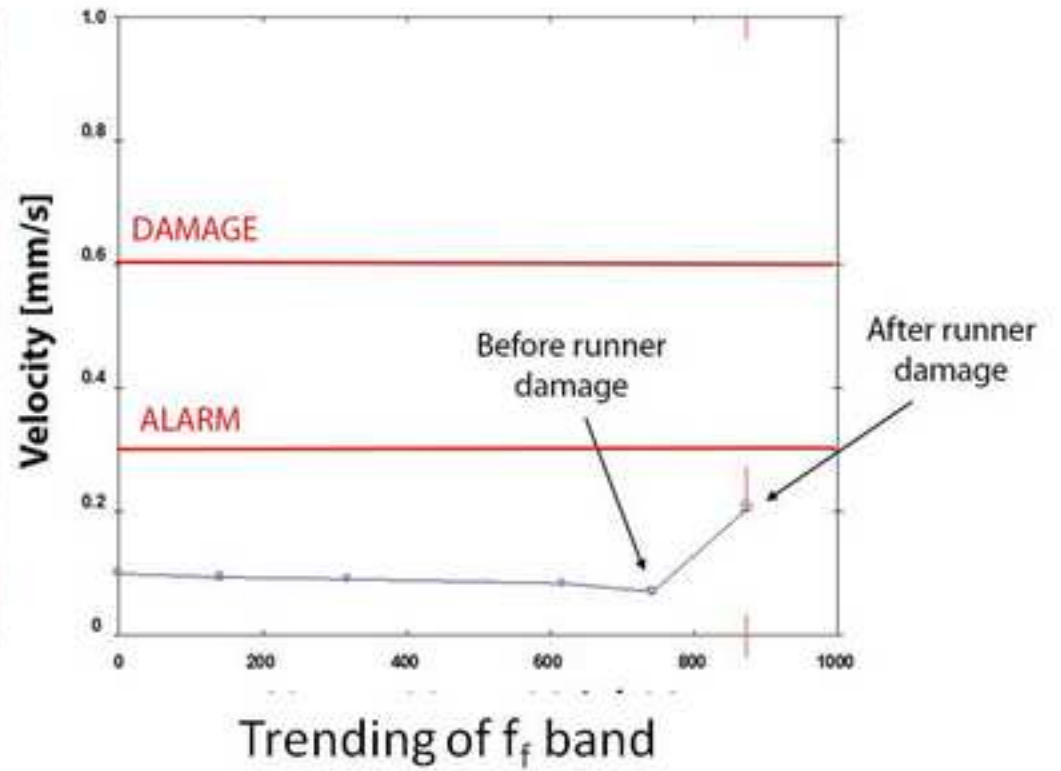


Table 1

	Power (MW)	Head (m)	$f_f$ (Hz)	Stage s	$Z_b/Z_v$	Main $k$	(m,n)	Main frequency	Corresponding frequency (Hz)
RPT-1	108	400	10	1	16/7	+2	(1,2)	$2 f_b$	140
RPT-2	90	350	8.33	1	20/9	+2	(1,2)	$2 f_b$	150
RPT-3	71	250	6.25	1	24/9	-3	(1,3)	$3 f_b$	168.75
RPT-4	82	850	1	3	12/7	-2	(1,2)	$2 f_b$	175

Table 2

Band name	Subharm	$f_f$	$2f_f - 3f_f$	$f_{pad}$	N harm.	$f_{pol}$	$nf_b$	$2f_{pol}$
Frequency band	$1Hz - 0.9 f_f$	$f_f \pm 0.1 f_f$	$1.1f_f - 3.1 f_f$	$f_{pad} \pm 1.1 f_f$	$3,1 f_f - N f_f$	$f_{pol} \pm 1.1 f_f$	$nf_b \pm 1.1 f_f$	$2f_{pol} \pm 1.1 f_f$
Main frequency	-	$f_f$	$2f_f, 3f_f$	$f_{pad}$		$f_{pol}$	$nf_b$	$2f_{pol}$

Table 3

Band name	Subharmonic	$f_f$	$2f_f - 3f_f$	$f_b$	$nf_b$	$f_b$ -harm
Frequency range	$1Hz - 0,9 f_f$	$f_f \pm 0,1 f_f$	$1,1 f_f - 3,1 f_f$	$f_b \pm 1,1 f_f$	$nf_b \pm 1,1 f_f$	$nf_b + 1,1 f_f - (n+2)f_b + 1,1 f_f$
Main frequency	$f_{vr}$	$f_f$	$2f_f, 3f_f$	$f_b$	$nf_b$	$(n+1)f_b, (n+2)f_b$



Table 4

Damage	Consequence	Symptom frequency	Spectral band	Comments
Generator stator eccentricity, shorted stator laminations	Uneven stationary air gap between rotor and stator	$2 \cdot f_{\text{line}}$	$f_{\text{pol}}$	Stator problems generate high vibrations at two times the line frequency (50Hz in Europe).  Compare vibrations with machine running idle and with excitation
Plain bearing wear	Fraction subharmonics of rotating speed	$\frac{1}{2}f_r, \frac{1}{3}f_r$	Sub-harmonic and 2-3 $f_r$	Different symptoms depending on the level of damage and type of bearing
	Large number of rotating speed harmonics	$f_r, 2f_r, 3f_r, 4f_r, 5f_r, \dots$	N harmonics mainly	Latter stages of bearing wear
Draft tube cavitation surge	Low frequency pressure pulsation	0.2 to 0.3 $f_r$	Sub-harmonic	Occurs only at part load
Low bearing stiffness	Change in rotor natural frequencies	2,5 – 3.5 $f_r$	2-3 $f_r$	Large vibration at around the rotating frequency and first harmonic
Tilting pad bearing damage	Vibration at the pad passing frequency	$f_r \cdot Z_{\text{pad}}$	$f_{\text{pad}}$	Vertical measurement in thrust bearing
Broken runner	Mechanical unbalance  Change in pressure pulses	$f_r$ mainly  $N \cdot f_b$	$f_r$ mainly	Small changes in unbalance  Harmonics of $f_b$ and side bands of $f_r$ around $f_b$
Distributor blockage	Flow distortion	$f_b$	$f_b$	Increase in amplitude of $f_b$ . Harmonics of $f_b$
Runner blockage	Hydraulic unbalance	$f_r$	$f_r$	Amplitude depends much on discharge. Compare $f_r$ amplitudes at low and maximum power
Cavitation	High frequency vibration	High frequency broad band vibration	Highest frequency bands	Demodulation of high frequency vibration measured in turbine bearing is convenient to identify erosive cavitation
Off design operation (low or high heads)	Turbulence	Broad band	$f_b$ , middle bands	Maximum amplitude at part load  Almost disappears at full load