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To cite this article: Mohamed Djemana *et al* 2022 *IOP Conf. Ser.: Mater. Sci. Eng.* **1244** 012022

Detection of unbalance and looseness faults in a ventilation turbine using vibration signature analysis

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Abstract. Heavy industry, which generally uses turbomachines, often uses both technologies to perform high-performance vibration monitoring of its production tool. Vibration analysis is one of the means used to monitor the health of rotating machinery in operation. This is part of a policy of forecast maintenance of the industrial production tool. This work is part of the monitoring and diagnosis of rotating machines by vibration analysis taking as an example the X205 circulation fan. Numerical simulation was done to test the capabilities and limitations of a dynamic simulation. The modelling of vibratory phenomena is developed using SolidWorks software to perform a dynamic simulation of the 3D model. The numerical simulations were performed to find the effect of different types of defects, such as defect of unbalance fin fan and bolt joint looseness, on the output of system. The numerical results were confirmed either by experimentally using employing results available in the open literature that existent at factory of cement plant (SCHS) and a good agreement was observed. The proposed and derived model has demonstrated the viability of dynamic simulation approach to rotating machines by vibration analysis and serves as a significant alternative approach to the direct experimentation on the same systems in terms of cost and time.

1. Introduction

Multiple fields of industry employ rotating machines. When these machines' working conditions are continuously monitored, the threat of catastrophic failures can be avoided. One method of condition monitoring is known as vibration analysis, and it is highly accurate and cost-effective [1]. The practice of vibration analysis is based on a solid theory foundation and has been in use long enough to have matured as a measurement tool [2]. It should be noted that rotating machinery typically generates a complex high frequency signal that is the result of many superimposed components, each of which can have different amplitude, non-linearity, and non-stationarity characteristics [3]. This complexity requires more robust and effective ways to separate individual vibration signals from the whole.

Fault diagnosis has recently become the focus of researchers. Gao et al. [4,5] performed a systematic review of four fault-diagnosis approaches namely, knowledge-based, signal-based, model-based and hybrid/active. The key to a method's success often relies on three steps: fault symptom determination, sensitive feature extraction, and condition pattern classification. Tayarani-Bathaie et al. [6] proposed a dynamic neural network that would perform the diagnosis of gas turbine faults. Their artificial neural network and empirical mode decomposition were used to identify automatic bearing faults apparent within vibration signals.

Qin et al. [7] proposed a concurrent fault diagnosis method based on Bayesian discriminating and time series analysis with dimensionless parameters. They applied their method to a centrifugal multi-level impeller blower with six states, including single and simultaneous defects in gear and bearing. Asr et al. [8] performed a combined bearing and gear fault diagnosis of the gearbox system using empirical mode decomposition (EMD), statistical features, and a non-Naive Bayesian classifier. Pan et al. [9] performed a compound fault detection of the bearing outer race and the cracked tooth of gear using the proposed symplectic geometry mode decomposition method.

The development of computer tools and numerical techniques has meant that numerical simulation now occupies a very important place and has established itself as a means that is both effective and economically inexpensive in comparison with experimental testing [1], numerical simulation is one of the tools for simulating real phenomena. It is a process whereby a computer program is run on a computer to simulate a complex physical phenomenon. It is to be noted that, most of the existing works have considered the rotating machine faults within the context of experimental set up but none have taken into consideration the effect of bolt joint looseness of fixed bearing and examine that numerically. Therefore, the results presented here are of unique significance and novelty.

In this work, this method is used in order to build a model that will be used to predict and detect efficiently the anomalies of a ventilation turbine and that will offer us the advantage of optimizing the potential of the latter. To this end, a dynamic simulation under the same conditions as the real model has been carried out, where the vibratory signal is the source of information retained to inform us about the state of this machine and the experimental validation of the numerical model. Firstly, we are interested in reproducing the unbalance fault which is the main cause of the faults affecting this equipment, and which is characterized by a mass imbalance around the axis of rotation which produces centrifugal forces generating vibrations at the level of the bearings likely to accelerate their degradation. Initially, any part has a greater or lesser degree of un-balance, the main causes of which may be machining or assembly faults or asymmetric heating of the rotor during operation [10]. Secondly, we are interested in reproducing the tightening defect, which can be the result of an initial unbalance [11]. In the present study, the vibratory information characterizing these faults is acquired for the un-balance by two methods, the experimental method and the numerical simulation, and for the loosening by a numerical simulation.

2. Vibration analysis

Vibrations are common phenomenon in rotating machinery, which could carry important information about condition of the rotating machinery. All specific failures in rotating machinery have their own characteristics of vibrations. Measurements and analysis of the vibratory behavior of the system and vibration changes leads to the detection of problems and faults [12,13]. Almost all machine vibrations are due to one or more reasons like unbalance, eccentricity, misalignment, looseness, belt-drive problems, gear defects, bearing defects, electrical faults, resonance.

Unbalance fault has been classified as one of the most common causes of vibration in rotating machines [14]. Machinery vibrations due to unbalance fault is usually characterized by a dominant peak at the fundamental rotational frequency (1x RPM), which usually changes in proportion to the square of the rotational speed and in the radial direction. The total elimination of unbalance fault in rotating machines is almost impracticable, due to the difficulties associated with achieving perfection in the manufacture of components as well as their installation. Based on this premise, a significant number of research have been centered on estimating unbalance and its correction.

Mechanical looseness can either occur in the form of internal looseness between the machine and its base mounting, internal looseness of the machine components or structural looseness. Internal looseness occurs due to lack of proper fit between the components of a machine (bearing-to-bearing housing, shaft-to-bearing, coupling-to-shaft, etc.), which consequently leads to the excitation of numerous harmonics of the machine running speed. These harmonics are usually generated due to combined effects of exciting forces from the rotor and the non-linear response from the loose components [15]. Looseness between machine and base mounting is commonly the resultant of a crack in the base structure or bearing pedestal, which may generate high 2x RPM component and some harmonics. Structural looseness on the other hand occurs when there is a weakness in the foundation or looseness in the supporting structure.

The presence of mechanical looseness in a machine train is often characterized by chaotic response, which have been observed to sometimes excite multiples of $\frac{1}{2}$x or $\frac{1}{3}$x RPM [16]. Other studies on mechanical looseness in rotating machines such as those related to bearing caps or supports have also been seen to have large numbers of harmonics and sub-harmonics, which were dependent on the analysis direction and point [17]. The vibration signatures produced by a rotating machine with mechanical looseness have also indicated that the vibration amplitude at the higher order frequency region will usually be greater than half of the vibration amplitude caused by the rotational speed, which will persist after machine balancing operations [18].

3. Methodology

In this work, the experimental and simulation analysis were carried out to predict and detect unbalance and looseness faults in an industrial fan. The modelling and simulation analysis were performed by SolidWorks software. On the experimental part, the vibration measurement on the rotating machinery were performed using accelerometer and tachometer. Validation of simulation and experimental results was carried out for the detection of unbalance fault.

3.1. Experimental procedure using hardware

For this study, we use the most common conditional maintenance approach for the diagnosis of rotating machines, vibration analysis. This technique provides a global view of the state of equipment and to detect possible malfunctions and monitor their evolution [19].

This application concerns the X205 industrial fan, which consists of an asynchronous motor (980 rpm), a coupling and an impeller with blades mounted on a shaft with two bearings, fitted with cylindrical roller bearings. The installation is shown in Figure 1.

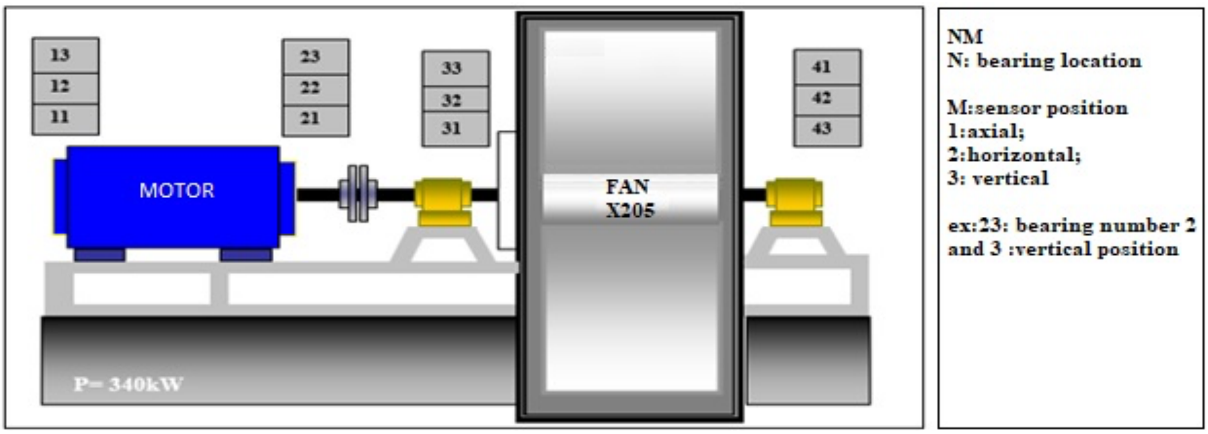

Fig. 1 Block diagram of the fan chain.

The acquisition of the vibration signals was carried out using a system consisting of a FALCON WLS wireless triaxial transducer associated with a FALCON analyzer collector. The software used for the vibration analysis was OneProd XPR300. Twelve measurement points were set up in all three directions in order to evaluate the behavior of the fan.

3.2. Numerical simulation

The numerical simulation was carried out using the SolidWorks Motion add-on, which is an integrated module in SolidWorks that allows the dynamic animation or simulation of motorized or externally forced assemblies by considering predefined contact conditions between the assembled components, which are assumed to be rigid.

The use of SolidWorks motion for the analysis of mechanisms consists of three steps: model generation, analysis (or simulation) and visualization of the results, as illustrated in Figure 2 [20].

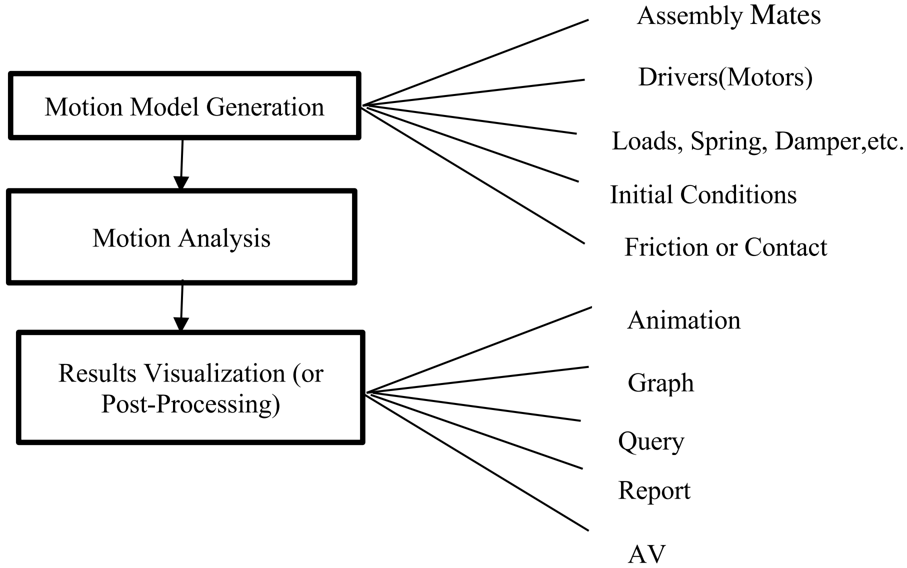

Fig. 2 General process of using SolidWorks Motion

The modelling concerns a part of the fan chain consisting of two roller bearings, a shaft, and an impeller. Figures 3 and 4 shows the modelling of the different parts of the fan chain.

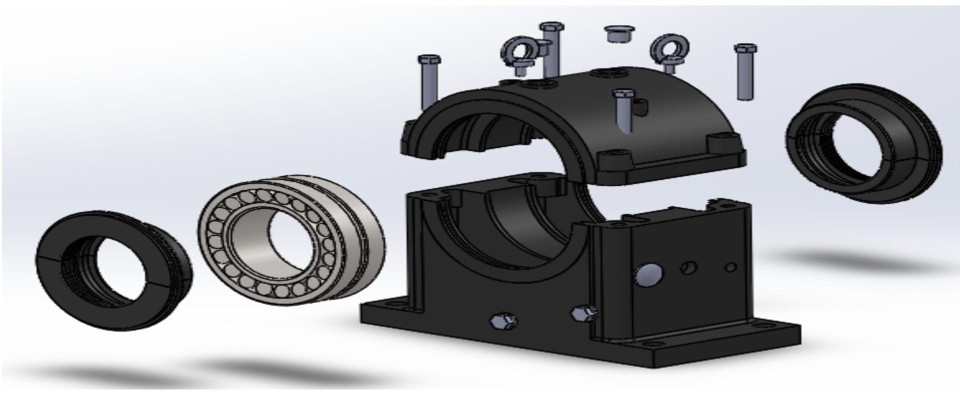

Fig. 3 Exploded view of the fixed bearing

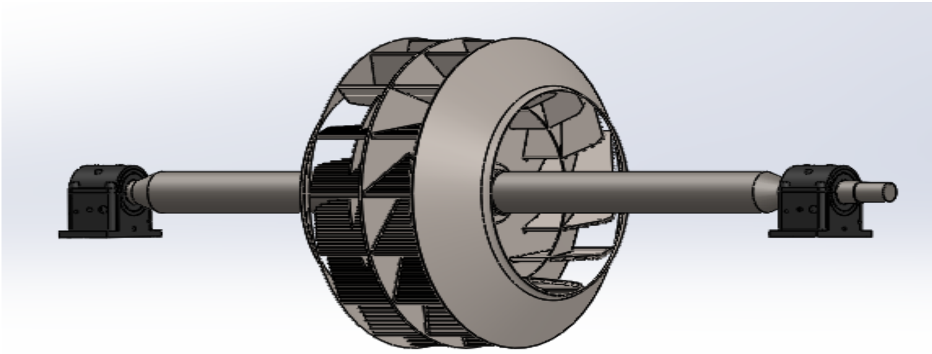

Fig. 4 Assembly of components

In the case of the fan, the unbalance phenomenon presented in the centrifugal impeller due to dust or wet flour and is superimposed non homogeneously on the impeller blades. The unbalance fault modelled by modelling dust bars on the impeller blades as shown in Figure 5.

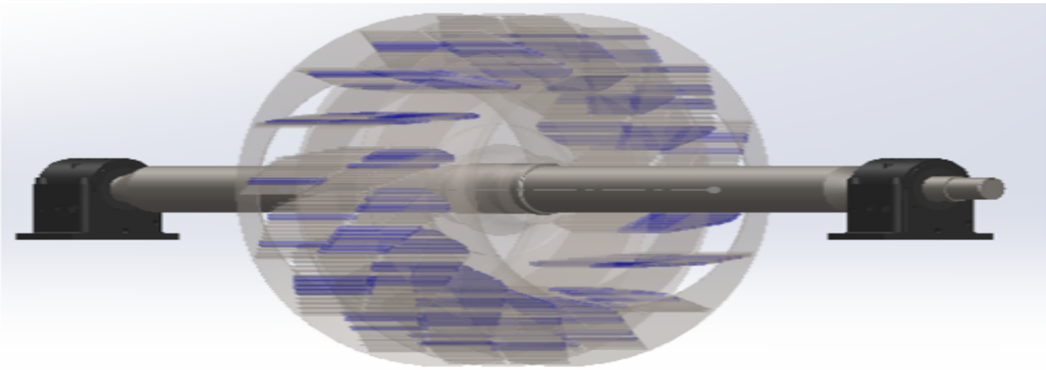

Fig. 5 Dust distribution on the turbine

Time signals of linear acceleration were generated from a dynamic simulation of the motion performed with the assembled model, a circular motor was applied to affect the motion to the turbine with a progressively increasing rotational speed reaching a constant angular velocity of 102.63 rad/s, for an estimated total motion simulation time of 6s. Figure 6 shows the velocity curve applied to the motor.

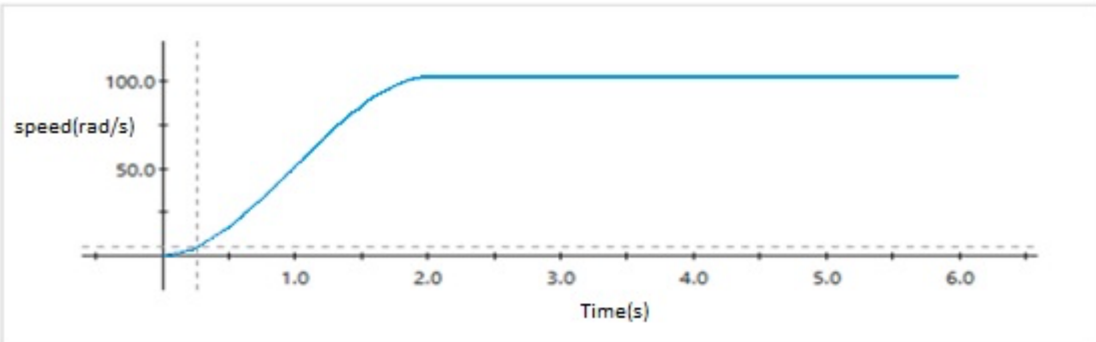

Fig. 6 Progressive speed curve

We have introduced the predefined contact conditions in SolidWorks motion for the lubricated contact of the bearing components Outer ring, Inner ring and rollers as shown in Figure 7.

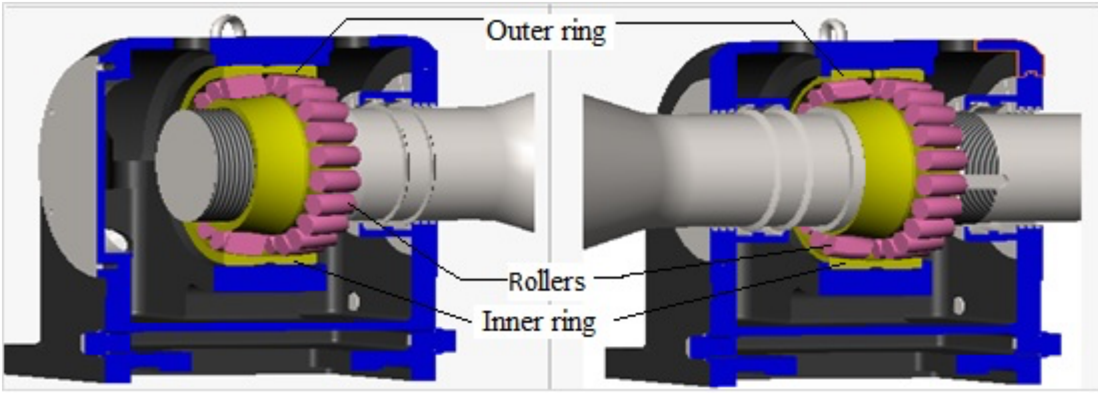

Fig. 7 Cross-section of contacting components

Table 1 Properties of contacts conditions

Elastic properties (Impact)	
Stiffness	100000.00 N/mm
Exponent	1.5
Max. damping	49.91566312 N/(mm/s)
Penetration	0.10 mm
friction	
v_k	10.16 mm/s
μ_k	0.05
Static friction	
v_s	0.10 mm/s
μ_s	0.08

Once the acquisition is complete, the signals are imported into MATLAB, so that their Fourier transform can be performed and compared with the experimental signals.

In order to model the clamping failure, we assumed that there is a clearance between the two bearing parts that occurs while the fan is in motion. We modelled this clearance by changing the stresses on the perfect bearing; to maintain a clearance between 0 mm and 0.5 mm when the fan is rotating, which in real life conditions embodies that the screws are loose (Figure 8).

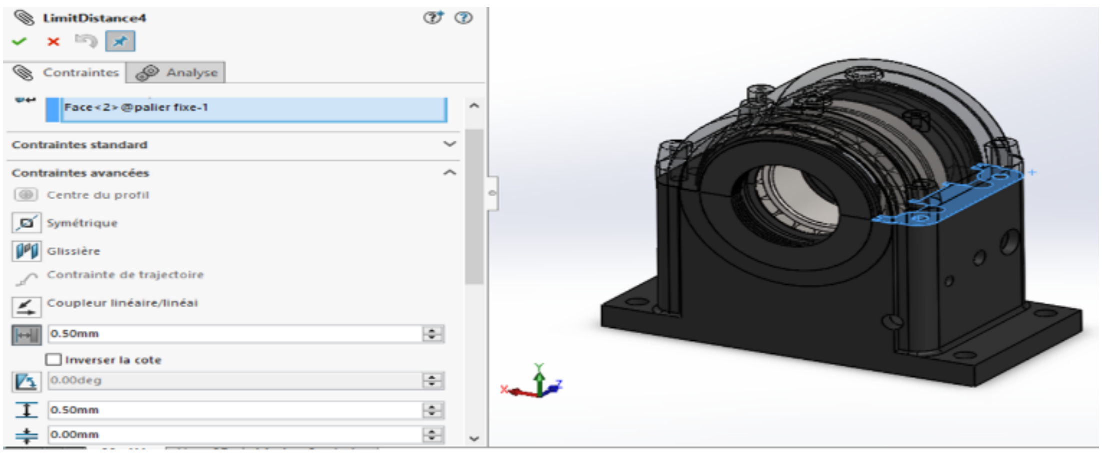

Fig. 8 The modelling of the clamping fault at the fixed bearing.

4. Results and discussion

4.1. Unbalance of fin fan

Figure 9 shows the spectrum recorded in point 32 (shown in fig.1) which represents the horizontal direction of bearing 3 showing a typical unbalance fault manifested by a high vibration amplitude at the rotation frequency (16.62 Hz).

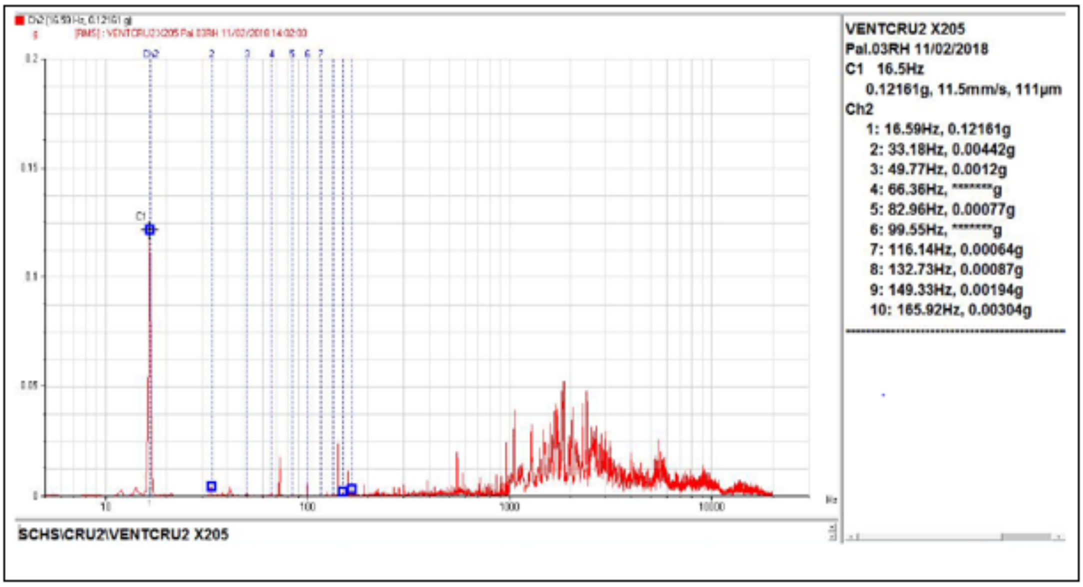

Fig. 9 The experimentally measured spectrum on bearing.

The results of the dynamic simulation are shown in figure 10 and 11 display the comparison between the experimental and simulated spectra.

From the results shown in Figure 11, it can be seen that the FFT of the experimental model and the simulated FFT, are correctly superimposed, the peaks at the rotational frequency (16.33 Hz) which characterize the unbalance defect coincide with a decrease in the vibratory amplitude visible in the simulation spectrum and the absence of some peaks. The differences observed can be explained by the external conditions that may affect the operation of the equipment (wind, temperature, etc.) and which are not considered in this study.

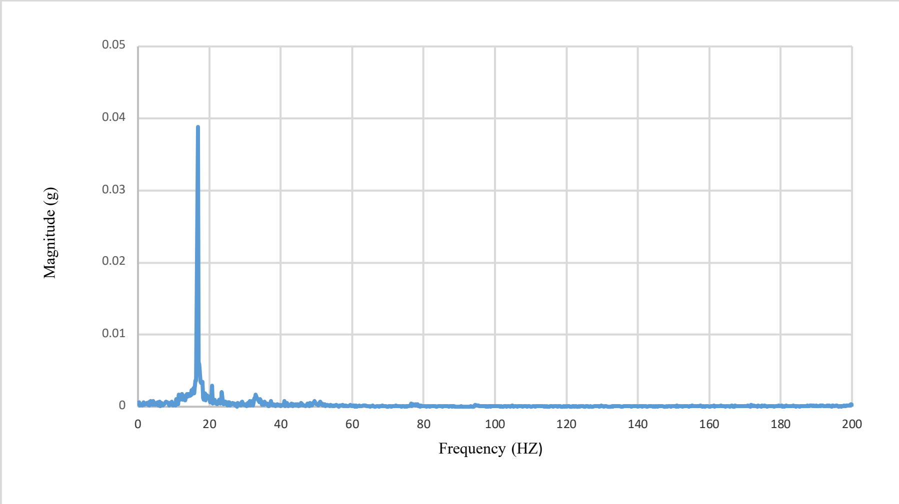

Fig. 10 Simulated spectrum of the unbalance fault.

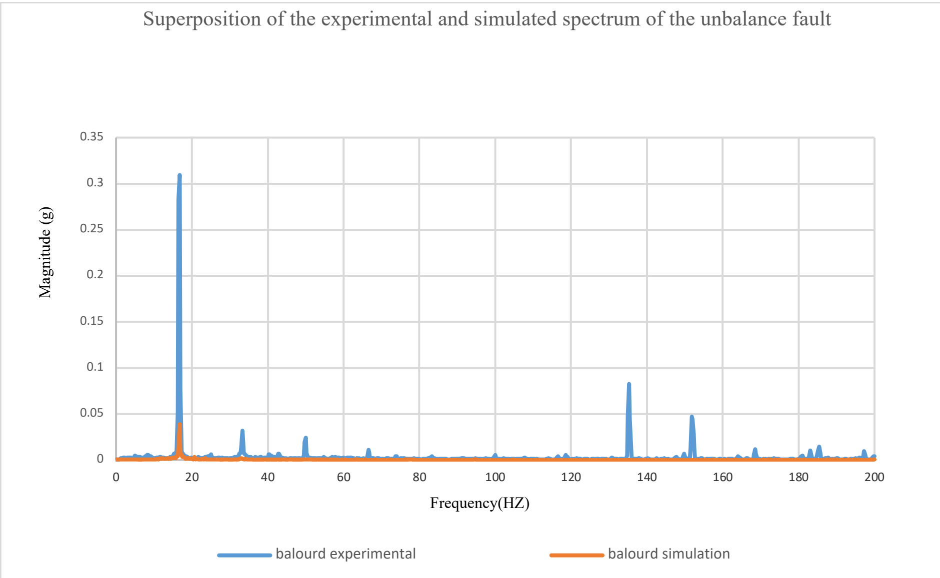

Fig. 11 Comparison of experimental and simulated FFTs.

one of the more commonly used indices include the correlation coefficient deviation metric (CCDM). When a fan chain is considered healthy, baseline, or reference, vibration signatures are taken. These are compared to later signatures to assess system health.

The correlation coefficient is the basis for the second index, CCDM, and is calculated by

$$CCDM = 1 - C_C$$

Where CC is the correlation coefficient calculated using vibration signatures for the fan chain under healthy and damaged conditions, as defined before, in the same frequency range. It’s calculated using the following equation:

$$C_C = \frac{\sum_{k=\omega_i}^{\omega_F}\left[Z_{E,H}(k)-\bar{Z}_{E,H}\right]\left[Z_{E,D}(k)-\bar{Z}_{E,D}\right]}{\sqrt{\sum_{k=\omega_i}^{\omega_F}\left[Z_{E,H}(k)-\bar{Z}_{E,H}\right]^2}\sqrt{\sum_{k=\omega_i}^{\omega_F}\left[Z_{E,D}(k)-\bar{Z}_{E,D}\right]^2}}$$

Comparing the results by measuring the degree of intercorrelation between the two signals corresponding to the experimental and numerical models gave us a similarity rate of 85%. This level of correlation between the two models confirms the good modelling of the fan chain and the assumptions of the dynamic simulation model and is a good start for further test studies.

4.2. Bolt joint Looseness of fixed bearing

The typical spectrum of a mis-tightening measured on a machine in which there is a backlash contains a large number of peaks at frequencies multiple of the rotational frequency (up to 10th order). It is also sometimes possible to find peaks at ½ harmonic (1/2 x rotational frequency) and its multiples at a level that is much lower than the harmonic of the fundamental frequency. [21]

A clamping fault in fixed bearing represented in figures 12 and 13. An absence of some peaks which can be explained by external conditions such as environment (temperature, humidity, noise..etc.) that are not taken into account in this study.

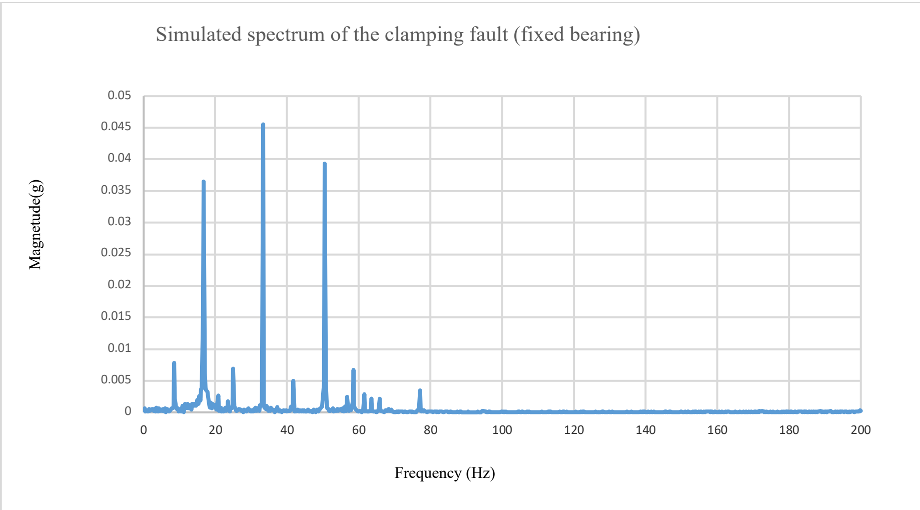

Fig. 12 Simulated loosening spectrum (Fixed bearing).

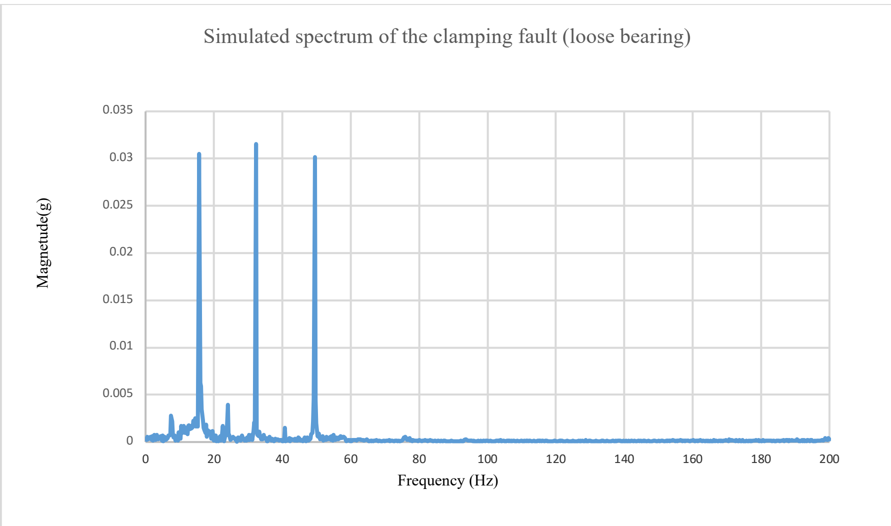

Fig. 13 Simulated loosening spectrum (Floating bearing).

5. Conclusion

The activity of validating models and simulations has become an essential activity, to ensure the credibility that can be given to the results provided by the simulation. We have created a dynamic simulation of the 3D model of the X205 circulation fan. So that it can be used as a decision support tool in the future, helping us to predict possible faults that may occur in the future.

The numerical simulation approach of the unbalance fault has given quite satisfactory results specific to the fan studied. Although this approach is far from covering all the physical phenomena that can exist in rotating machines, it remains quite promising and opens up an interesting perspective in terms of learning about the faults that can occur, saving time and money and making it easier to adapt to the various rotating machines that exist in industry. This approach, therefore, deserves to be explored for other faults and in other machines, and may one day replace learning by the classic methods (test bench and feedback).

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