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Experimental and Numerical Analysis of Helical Gear of Raw Mill in Hamma Bouzian Cement

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1. Introduction

Nowadays cement has become a rare commodity and much coveted. A fortuitous shutdown of machines results in significant losses for the company. This is why the mastery of systems and means of production has always been the major concern of manufacturers. Now, efficient maintenance of industrial systems is a major economic issue for commercial operations. The main difficulties and sources of inefficiency lie in the choice of maintenance actions to be undertaken at the appropriate time, especially when the machine plays a vital role in the production process (case of our machine). These flaws can have serious consequences. Thus, to avoid unplanned production stoppages and the considerable economic benefits that result from them, it is necessary to constantly monitor this equipment and track down all the warning signs of defect before it is too late. Guangbin Wang [1] made a targeted study on non-linear and non-stationary characteristics of different degrees with single broken faulty gear tooth, pitting and broken composite faulty gear teeth, a method of diagnosing absolute deviation of gear faults is presented. Jedrzej Maczak [2] presented a methodology for developing model-based method of gear fault diagnostics; the simulation model of the helical gearbox is discussed allowing analysis of the phenomena taking place during teeth mating in the presence of manufacturing and assembly errors. It includes observation of influence of errors on the generated signals. M. Gh. Khosroshahi [3] determined the distribution of contact stresses, bending stresses and torque transmission in the pinion for a given tangential displacement (rotation) of the pinion hole surface was investigated using FEM and the obtained stresses are then compared with Hertz theory and AGMA standard. Hanjun Jiang [4] presented, an analytical model of the mesh stiffness for cracked helical gears considering the axial mesh stiffness components and the deflection of the gear body has been proposed. Based on the analytical model, the comparison of the mesh stiffness with and without the axial mesh stiffness components or the deflection of the gear body was carried out. Jong Boon Ooi [5] did a modal analysis on three different combinations of gear train system commonly designed for portal axle. The three gear trains being analyzed are gear train without idler gear, one idler gear and two idler gears. FEM static stress analysis is also simulated on three different gear trains to study the gear teeth bending stress and contact stress behavior of the gear trains in different angular positions from 0° to 18°. The single and double pair gear teeth contact are also considered. Zhaoping Tang and al [6] analyzed meshing characters of high speed train helical gear, studied tractive performance curve of CRH380A, and took the CRH380A train's G301 traction helical gear as an example, gear pair was established by Pro/e and then imported into ANSYS workbench to explore the nonlinear contact condition, mainly studied the dynamic contact stress state of helical gear during three working conditions as start-up, continuous and high speed. Santosh S. Patil [7] made a study on the contact stresses among the helical gear pairs, under static conditions, by using a 3D finite element method. The helical gear pairs on which the analysis was carried out were 01, 51, 151, 251 helical gear sets. The Lagrange multiplier algorithm has been used between the contacting pairs to determine the stresses. The effect of friction was varied at the point of contact which made the problem nonlinear and complicated. Hui Li and al [8], in his study, Hermitian wavelet is used to diagnose the gear localized crack fault. The simulative and experimental results show that Hermitian wavelet can extract the transients from strong noise signals and can effectively diagnose the localized gear fault.

Guangfu Bin and al [9] A new method of dynamic shaft balancing without test weight requirements is proposed. they believed that the new methods developed in his work will help reduce the manufacturer's time and cost of equipment or field dynamic balancing procedures.

P.N. Saavedra and al [10], they studied the computed order tracking (COT), and a new computed procedure is proposed for solving the indeterminate results generated by the traditional method at constant speed. The effect on the accuracy of the assumptions inherent in the COT was assessed using data from various simulations. The use of these simulations allowed them to determine the effect on the overall true accuracy of the method of different user-defined factors: the signal and tachometric pulse sampling frequency, the method of amplitude interpolation, and the number of tachometric pulses per revolution.

2. Experimental method

Following abnormal noise reported by employees, and in order to anticipate degradations that may affect a mechanism, detect defects at a more or less early stage, avoid very costly production stoppages due to unforeseen failures, diagnosis was carried out using vibration analysis (spectral analysis, measurements of the overall vibratory levels in speed and acceleration and envelope analysis). The application of this approach allowed us after analysis to deduce the origin and estimate the risks of failure. Particular attention is paid to the raw mill transmission system, which represents one of the links constituting the cement production line, from the extraction of the limestone to the shipment of the cement.

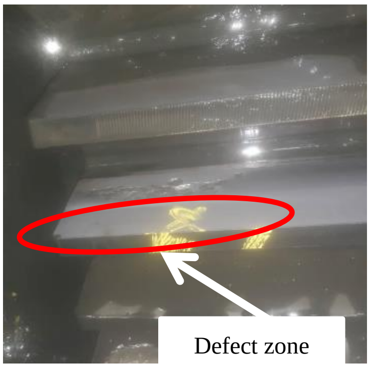

Fig. 1 Defect teeth in raw mill crown

2.1. Equipment for diagnosis

The vibration analysis equipment used for this diagnosis consists mainly of:

2.1.1. Vibration analysis software

All the information is recorded in the OMNITREND software, the tree structure of this data represents the hierarchical structure of the OMNITREND database and indicates the organization of the measuring points of the machine park, as well as the measurement tasks to be carried out execute.

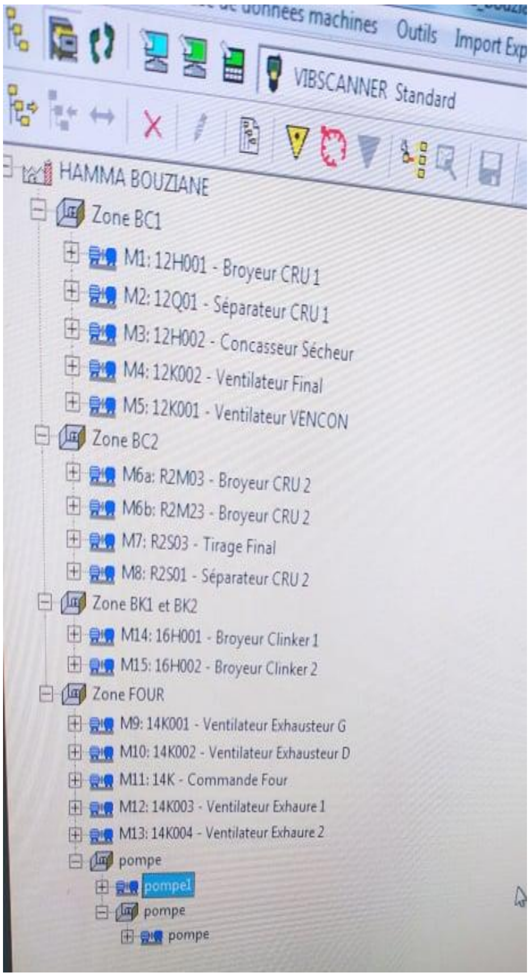

Fig. 2 The hierarchical structure of OMNITREND database

2.1.2 Measurement sensor (the accelerometer)

It is placed horizontally on the bearings and is characterized by its very large frequency range.

Fig. 3 Horizontal and vertical position of sensors

2.2.2. Experimental results

The complexity of the constitution of certain machines and the accessibility of certain places make it difficult, sometimes even impossible, to take precise measurements of the parts to be monitored. To avoid these constraints and to better understand the problem observed and identify the defective component of the system, or even the appropriate maintenance action, in addition to the bearings, other points were selected at places likely to give the maximum amount of information on the vibratory behavior of the mechanism. In total, we have configured five measuring points in each control to assess the behavior of the system. (Fig. 4).

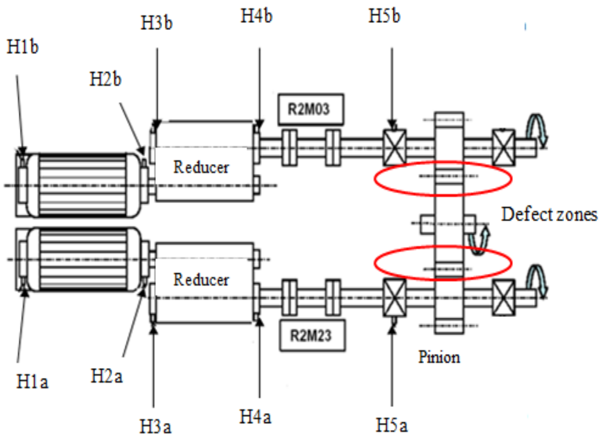

Fig. 4 Measuring points

The frequency analysis makes it possible to identify the source of the defect. The presence of such a defect leads to an increase in the amplitude of the signal at the gears frequency and its harmonics, even in the presence of other defects. The Fig. 5 clearly represent the vibration image of the measurement points in the different bearings of the two transmission systems 'a' and 'b' in the horizontal direction in the frequency range [0, 20000] Hz. The spectra show that there is an unbalance defect at the 24.61 Hz with speed amplitude of 2.95 mm/s in the bearing 1b and at the 65.63 Hz in the other bearings.

The spectrum of the bearing 1 presents a peak at the frequency of 65.63 Hz which reflects a gear defect (pinion/crown) with an amplitude of 0.60 mm/s and another peak at the frequency of 24.61 Hz which presents a unbalance defect with Amplitude 0.50 mm/s, we find the two peaks in the good band on the table of vibration levels (ISO 2372 standard).

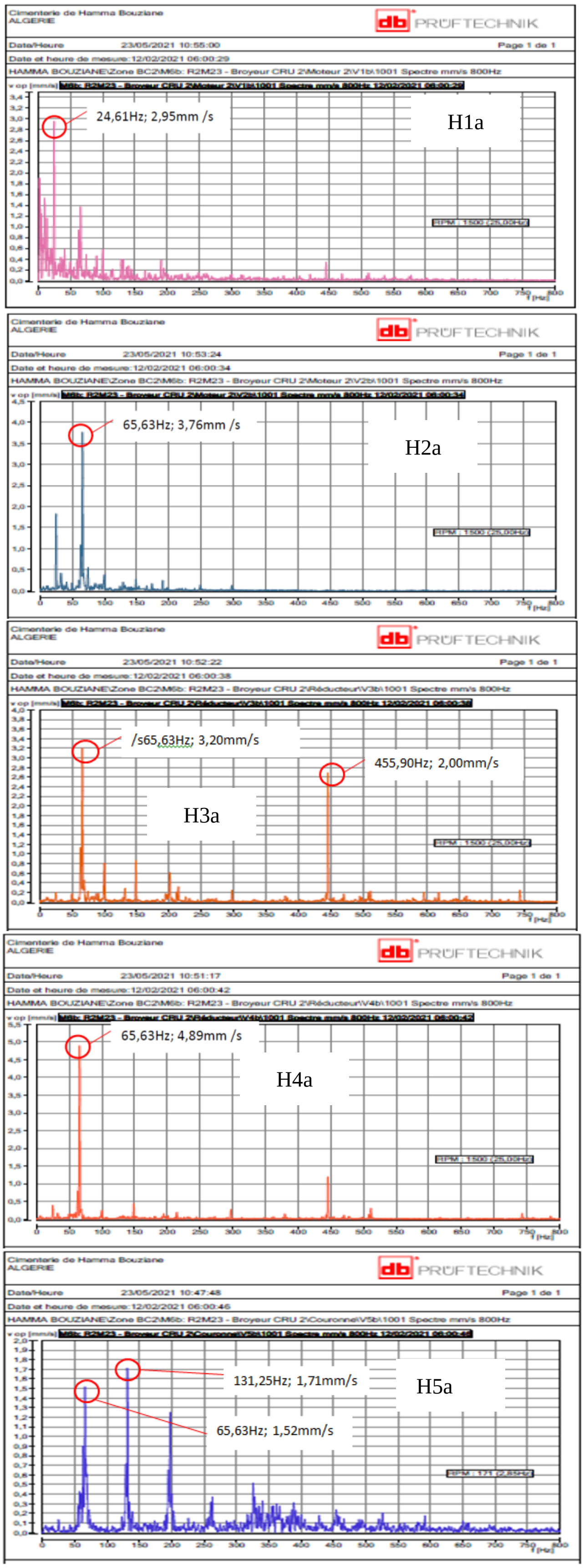

Fig. 5 Spectral analysis of the system 1

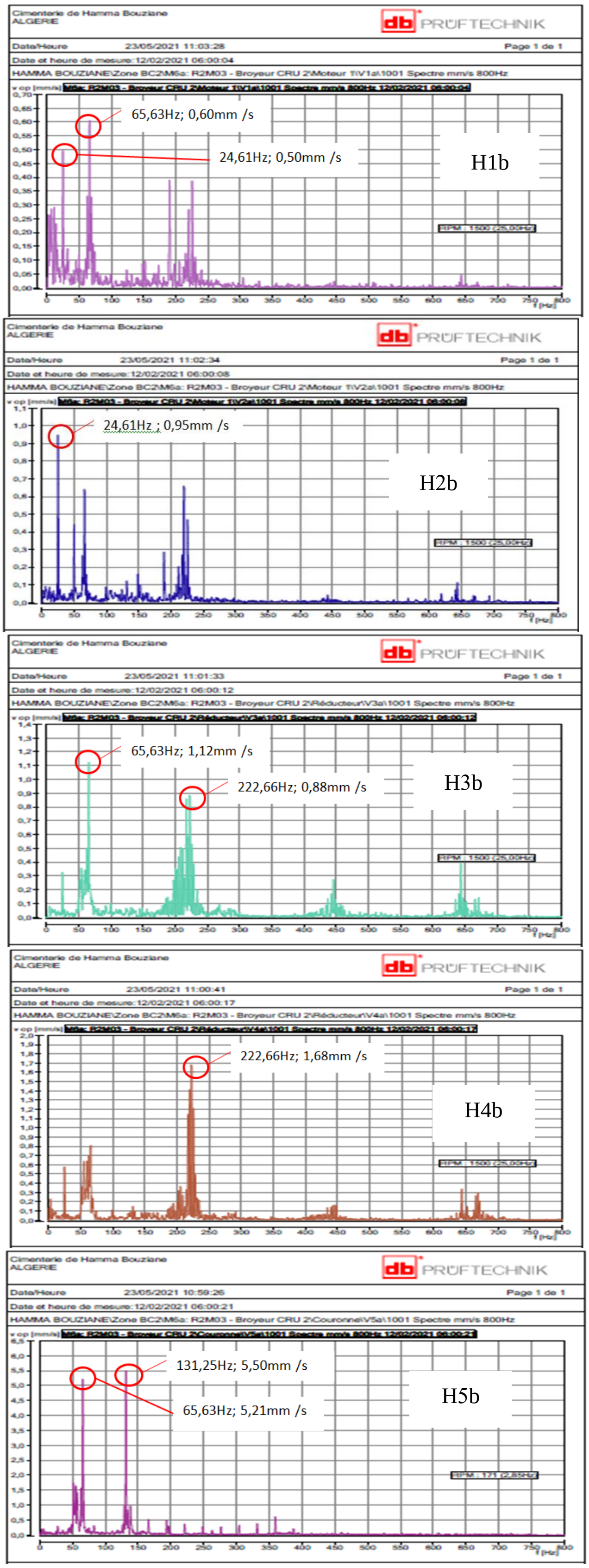

Fig. 6 Spectral analysis of the system 2

3. Numerical method

3.1. Model of gear shaft and idler gear

Fig. 7 shows the assembly model of the three-gear system of the portal axle unit. In this analysis, the model of the input gear shaft, output gear shaft, and idler gear is only considered in FEM analysis to save computing time. This ignores the interaction of the housing fitting and the bearings fitting of the portal axle.

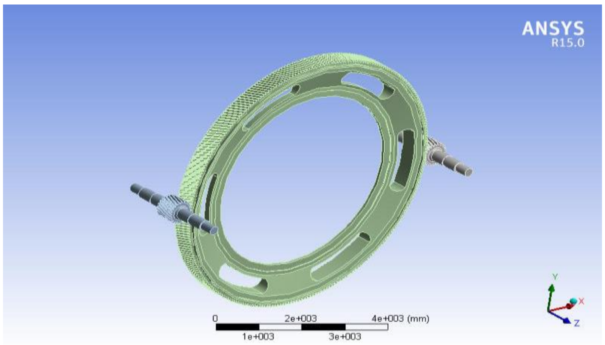

Fig. 7 Geometrical model of the system

All gears (gear shaft and idler gear) are modelled following the same gear design parameters and material properties as shown in Table 1.

Table 1

Gear shaft parameters and material properties

Parameter	Crown	Pinion
Normal module	25	25
Normal pressure angle	20 deg	20deg
Helix angle	20 deg	20 deg
Face width	485 mm	485 mm
Number of teeth	272	23
Young’s modulus	206 GPa	206 GPa
Poison’s ratio	0.3	0.3

Contact pressure maps of gear pair with different defects size are extracted and displayed in Fig. 8. Fig.8, a contact pressure maps of gear pair without defect. Figs 8, b, c are respectively the contact pressure maps with defect.

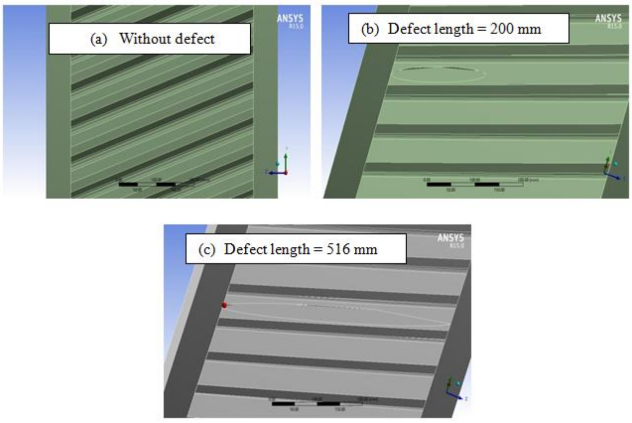

Fig. 8Gear train with different defect

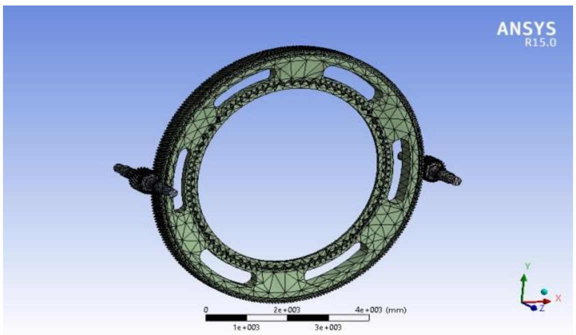

Fig. 9 Mesh model of the gear train

ANSYS Workbench, the three gear train (tow pinion and the crown) designs are imported. In harmonic analysis using FEM, each gear train model was meshed with purely tetrahedron elements. Fig. 9 shows the mesh model of the gear train with one idler gear. The model consists of 83696 nodes and 45507 elements.

3.2. Modal analysis of three different gear trains

Harmonic analysis, which means the study of the structure mode shape under excitation to its natural frequency, is important in the design stage. The modal analysis of the gear train with different combinations (no idler, one idler and two idler gears) was analyzed under free stress condition and pre-stress condition. Single pair tooth contact and double pair tooth contact between gears were also considered in the analysis.

The mass m and stiffness k are the main properties that affect the overall modal response of a system. In general, the dynamic equilibrium is given by the differential matrix system:

$$m\ddot{u} + ku = f(t), \tag{1}$$

where: u is the nodal displacement vector; $\ddot{u}$ is the nodal acceleration vector; $f(t)$ the nodal applied force vector. For a homogeneous system, the solution of the second order linear differential equation is:

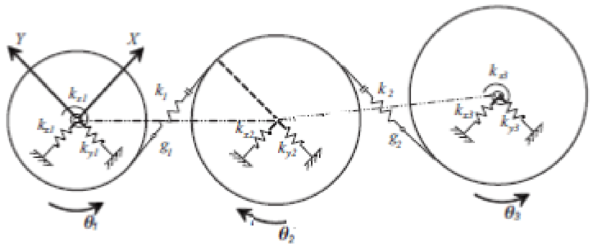

Fig. 10 Translational-rotational model of idler gearsets

$$\begin{cases} m_1\ddot{x}_1 + k_{x1}x_1 + F_1 = 0 \\ m_1\ddot{y}_1 + k_{y1}y_1 + F_1 = 0 \ , \\ I_1\ddot{\theta}_1 + k_{s1}\theta_1 + r_1F_1 = T_1 \end{cases} \tag{2}$$

$$\begin{cases} m_2\ddot{x}_2 + k_{x2}x_2 + F_1 - F_2 = 0 \\ m_2\ddot{y}_2 + k_{y2}y_2 + F_2 = 0 \quad , \\ I_2\ddot{\theta}_2 + r_2F_1 - r_2F_2 = 0 \end{cases} \tag{3}$$

$$\begin{cases} m_3\ddot{x}_3 + k_{x3}x_3 - F_2 = 0 \\ m_3\ddot{y}_3 + k_{y3}y_3 + F_2 = 0 \, , \\ I_3\ddot{\theta}_3 + k_{s3}\theta_3 - r_3F_2 = 0 \end{cases} \tag{4}$$

The matrix $|K|$ is not singular if the boundary conditions are considered. Thus the non-zero solutions exist if and only if:

$$\det\left\| K \right| + \left| M \right| \omega^2 \right| = 0. \tag{5}$$

3.3. Boundary condition settings of gear train

ANSYS Workbench program was used to apply the motion constraints and contact conditions. There are two different conditions being analyzed in conducting modal analysis of the gear trains. The first six natural frequencies and their mode shapes were analyzed on the gear train without pre-stress state involving constraints only and pre-stress state which is inclusive of loads and constraints.

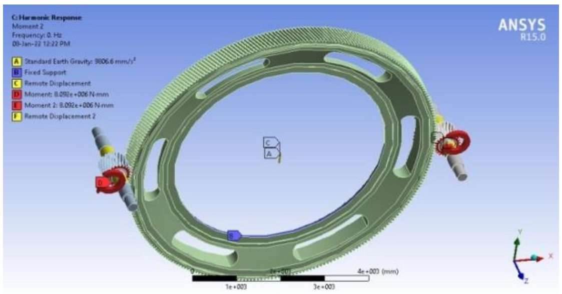

Fig. 11 Boundary conditions

3.4. Numerical results

The gear vibration velocity and displacement in the third study cases indicated above are shown in Fig. 12, we have found that the right pinion supports large vibrations in our system.

From the graph shown in Fig. 12 (without defect case 1) it is reveal that there is vibration of 1.7316 mm/s peak in 60 Hz. Hence, the crown is running in satisfactory condition in horizontal direction.

In the second and the third cases we noticed that the influence of defect on our system is very important, where the amplitude of vibrations reached maximum values 3.3936 mm/s in second case and 5.6369 mm/s in the third case in the 60 Hz. From the above three figures, it can be seen that there is an average vibration in the horizontal direction which indicates an average vibration level. It is observed that the system operates in severe condition and in order to minimize this problem, it is necessary to rectify the gear defect and to verify the dynamic analysis.

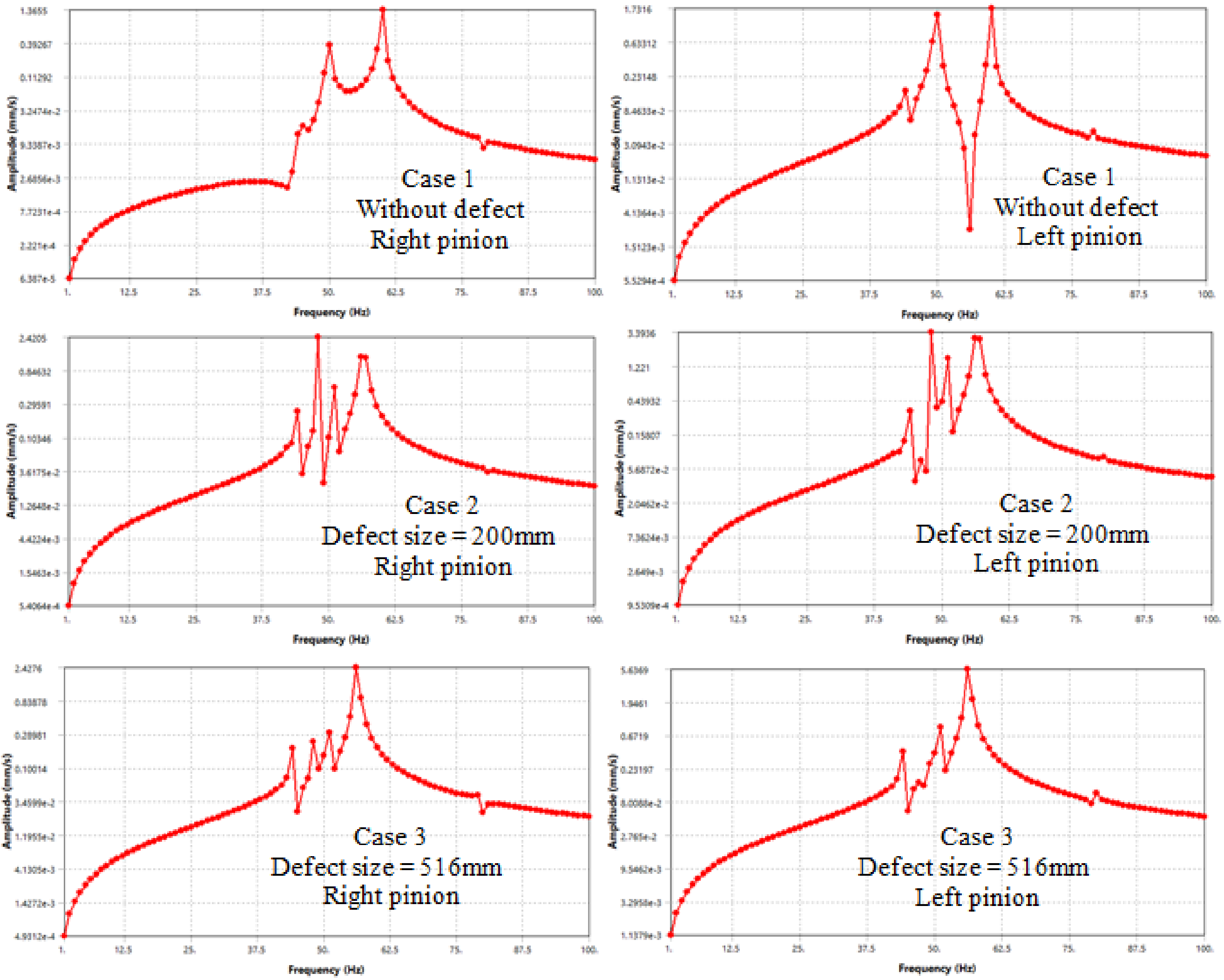

Fig. 12 Evolution of vibration velocity

From the graph shown in Fig. 13 (without defect case) it is reveal that there is vibration of 4.5933e^{-3} mm peak in 60 Hz. Hence, the crown is running in satisfactory condition in horizontal direction. In the second and the third cases we noticed that the influence of defect on our system is very important, where the amplitude of vibrations reached maximum values 1.1252 e^{-2} mm in second case and 1.602 e^{-2} mm in the third case in the 60 Hz.

From the above three figures, it can be seen that there is an average vibration in the horizontal direction which indicates an average vibration level. It is observed that the system operates in severe condition and in order to minimize this problem, it is necessary to rectify the gear defect and to verify the dynamic analysis.

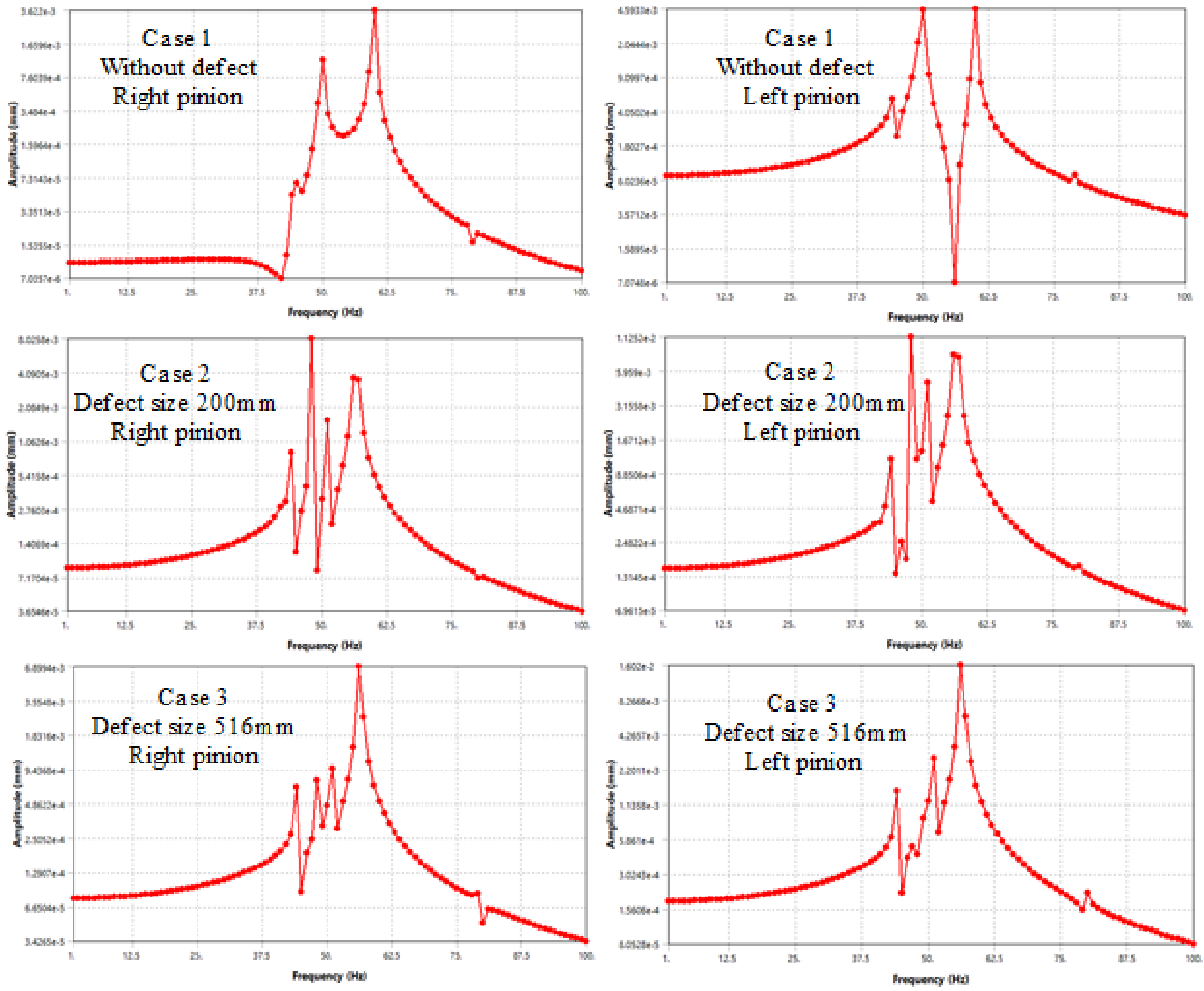

Fig. 13 Evolution of vibration displacement

To validate our calculation, we made a comparison between the numerical and experimental method (Table 2), we found that the calculation error is acceptable so we can say that the numerical results are correct.

Table 2

The comparison between numerical and experimental methods

	Vibration velocity	Frequency
Experimental results	1.52 mm/s	65.63 Hz
Numerical results	1.3655 mm/s	60 Hz
Error	0.101	0.082

4. Conclusion

The vibrations therefore contain all the information concerning the condition of the mechanical parts of the machine. The difficulty lies in the analysis of the vibratory signals and in the identification of the components relating to the elements to be monitored. This work is aimed in particular at the diagnostic process, the detection of anomalies such as gear and bearing faults, which is why a vibration analysis (spectral analysis) was carried out on a BC2 raw mill.

We found a very large gear defect which required us to call for an effective intervention while still recovering the raw mill in good working order. The numerical study by the software ANSYS shows that there is a defect of meshing at the level of the crown of the cement raw mill; one note that there is a concordance with the experimental results.

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EXPERIMENTAL AND NUMERICAL ANALYSIS OF HELICAL GEAR OF RAW MILL IN HAMMA BOUZIAN CEMENT

S u m m a r y

The matting of gears can be considered one of the most complicated faults to diagnose because its vibratory signature is not really known. In addition, the integration of cracks in digital models is not a simple task. On the other hand, the diagnosis of gears can be done in the time domain through statistical method or in the frequency domain through spectral analysis. In this study we made a vibratory analysis on the whole transmission system from raw mill to the Hamma bouzien-constantine cement plant, and then we compared these results with a numerical model. The numerical results found are very close to the experimental one. The experimental study was carried out by the OMNITREND processing software and the numerical simulation by ansys workbench.

Keywords: raw mill, helical gear, experimental method, numerical method, defect gear.

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