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CONTACT STRESS ANALYSIS OF MATING INVOLUTE HELICAL GEAR TEETH OF RAW MILL
CASE STUDY: THE CEMENT PLANT OF HAMMA BOUZIANE. ALGERIA

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Abstract: *Noise and vibrations are the main reasons for failure of transmission system, minimizing the noise and vibration in power system is a constant development, gear faults are the main sources that create vibrations in mechanical systems.*
According to our experience, to apply at the cement plant of Hamma Bouziane in Industrial Zone of Constantine and using very sophisticated measuring means (OMNITREND processing software) and the development of the vibration sensors, an anomaly was detected in the part (pinion / crown).
The objective of this research is to determine the influence of the defect size on the mechanical behavior of helical gear under dynamic load and the different mode shape of vibration of pinion-crown. The distribution of contact stresses, bending stresses was investigated using FEM and the obtained stresses are then compared with AGMA standard. The modeling of gears was performed using SOLID WORKS software and imported into the finite element code ANSYS.
Key words: *Pinion, Crown, Raw mill, Dynamic load, Ansys, Omni trend.*

1. INTRODUCTION

Helical Gears are used in the engineering industry as a means of transmitting power from one shaft to another. They are obtained by cutting on special machine tools either by reproduction or by generation. The precision of execution depends essentially on the value and the precision of the ratio of the gears introduced in the kinematic chain which links the rotation of the milling cutter and the rotation of the work piece spindle.

In a gear, the most important excitation sources are produced mainly by the variation of the stiffness of the mesh, itself due to the variation of the load and the contact length over time. However, it is now established that the excitations produced in this way alone cannot explain the vibratory behavior of gears and that geometric deviations represent a significant source of internal vibrations. In fact, the contacts between the teeth and the elastic couplings between the structural parts lead to a greater or lesser relocation of the contact zones. This is all amplified by the reduction in mass which involves the integration of a mechanical assembly with complex geometries and flexible parts (light wheels, thin sails, etc.) in the design of the gearboxes. GuangbinWang [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. J˛edrzej M˛aczak [2] presented a study based on a diagnostic model of gear faults in a helical gearbox with the presence of manufacturing and assembly faults. It determines the influence of errors on the generated signals. M. Gh. KHOSROSHAHI [3] made a comparative study between FEM and the theory of Hertz and the AGMA standard of the distribution of bending stresses and contract pressure at the level of the teeth. Hanjun Jiang [4] presented an analytical study on the contact stiffness of helical teeth with and without crack. JongBoon Ooi [5] carried out a modal analysis on a gear train with and without an idler gear, he

took into account the type of contact either double or pair. the results obtained by the FEM method were compared by the analytical equations of AGMASantosh S. Patil [6] carried out a study on the contact pressure of helical gears by the 3D finite element method, taking into account the variation of friction in the contact zone.
In this research, a FEM package, ANSYS, capable of analyzing contacts, was used to determine the stress distribution on a transmission system of a raw mill. To do this, three helical gears (a crown and two attack pinions) mesh with and without fault. The contact and bending stresses are calculated by the finite element method (FEM) and are then compared to the stress results obtained from the AGMA standard.

2. VIBRATION ANALYSIS

Following an abnormal noise reported by employees, a diagnosis was made using vibration analysis (spectral analysis, measurements of overall vibration levels in speed and acceleration and envelope analysis) (Figure 1).

Fig. 1. Matting defect on the toothing of the crown.

The Accelerometer is placed horizontally on the bearings and is characterized by its' frequency large range.

Fig. 2. The sensor control point.

The measurement highlighted the evolution of spectral analysis in speed at crown gear of the raw mill, the spectrum shows that there is a fault at the frequency 65.63 Hz with amplitude 20.29 mm/s which exceeds the danger threshold according to the vibration level table (ISO 10816 standard).

This defect presents a general wear of the teeth which reflects a matting of the profile of the teeth; a periodic shock is obtained at the gear frequency.

Table 1

Admissible vibration levels (ISO 10816)

VIBRATION SEVERITY PER ISO 10816						
Machine			Class I small machines	Class II medium machines	Class III large rigid foundation	Class IV large soft foundation
	in/s	mm/s				
Vibration Velocity Vrms	0.01	0.28				
	0.02	0.45				
	0.03	0.71		good		
	0.04	1.12				
	0.07	1.80				
	0.11	2.80		satisfactory		
	0.18	4.50				
	0.28	7.10		unsatisfactory		
	0.44	11.2				
	0.70	18.0				
	0.71	28.0		unacceptable		
	1.10	45.0				

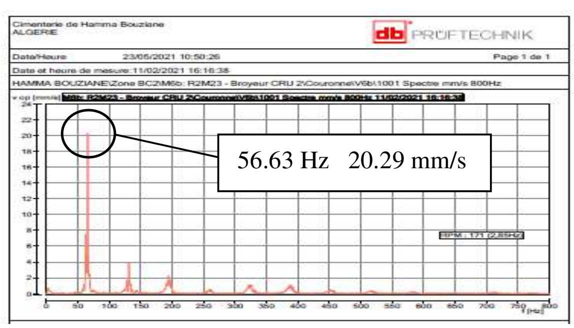

Fig. 3. Spectral analysis of the crown.

The transmission system of a double contact raw mill consists of two electric motors, two reducers, two pinions, a single crown wheel and bearings constituting a rigid chain.

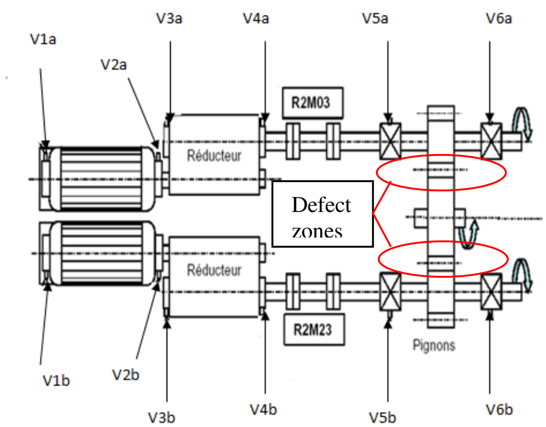

Fig. 4. Synoptic diagram of the measuring points.

3. GEAR TOOTH BENDING STRESS USING LEWIS EQUATION

Wilfred Lewis, put in equation to determine the bending stresses at the root of the tooth. In this equation the gear tooth is considered as a simple cantilever beam as shown in Figure 5.

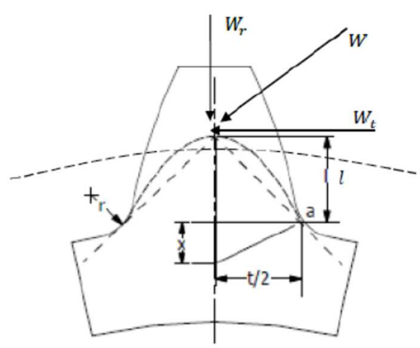

Fig. 5. Loads and length dimensions used in determining tooth bending stress.

The Lewis equation is stated as below:

$$\sigma_t = \frac{W_t P_d}{b_w Y} \tag{1}$$

Where:
P_d is the diametric pitch, b_w is the face width, and he Lewis form factor, Y is :

$$Y = \frac{2xP_d}{3} \tag{2}$$

And x dimension can be determined from:

$$l = \frac{t^2}{4x} \tag{3}$$

4. GEOMETRICAL MODEL

When using ANSYS workbench to analyze helical gear contact stress, if the single tooth model is used, results like helical gear meshing basic status, contact deformation and stress cannot be denoted precisely. But if full tooth model is adopted, the elements quantity of finite element method will become oversize, therefore, the calculation time and efficiency of finite element method will be influenced. The material attributes of small and large gears are defined as structural steel in ANSYS Workbench, the elastic module of crown and the tow pinions is 2 $\times$ 10^{11} and . Poisson ratio is 0.3. The tooth surface that has the possibility to contact should be grid refined at the same time. The model owns 152490 nodes and 88497 elements totally shown in the Figure 6.

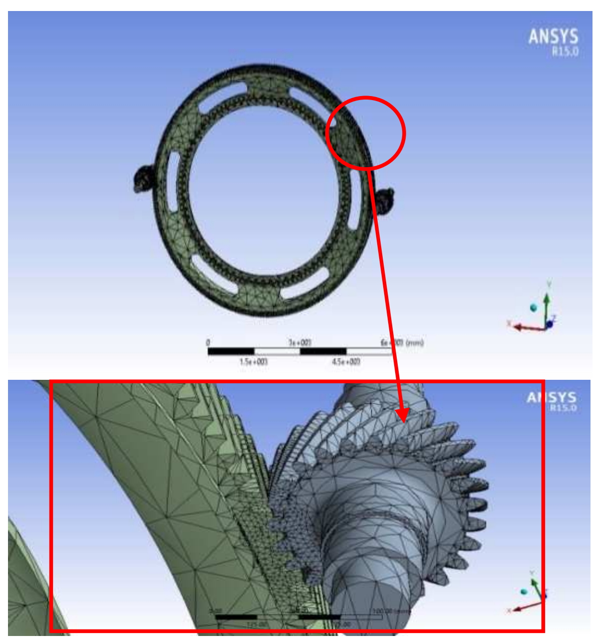

Fig. 6. Numerical model of gears.

All gears (pinions and crown) are modeled following the same gear design parameters and material properties as shown in Table 1.

Table 1

Gears 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

The mechanical boundary conditions are shown in Figure 7. The analysis was carried out considering a variable torque which is illustrated in the figure 8.

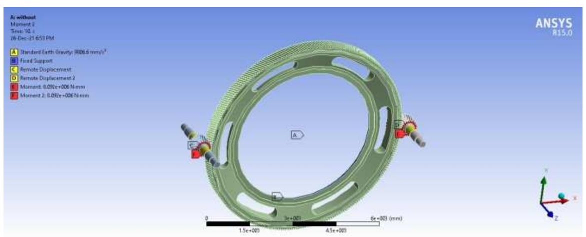

Fig. 7. Mechanical boundary conditions.

Transient dynamic analysis is a time domain analysis can determine the dynamic response process of the structure under the varying effect of time load. The load changed with time is the input data, and the output data is the displacement which changes with time or other quantities such as strain and stress. Known from classical force theory, the common dynamic equation is as below.

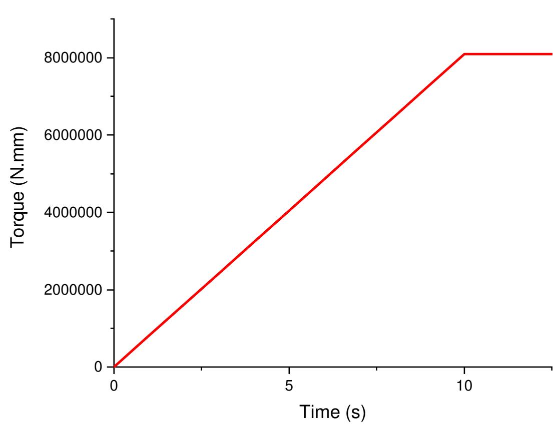

Fig. 8. Variable torque applied.

5. RESULTS AND DISCUSSION

Contact pressure maps of gear pair with different defects size are extracted and displayed in Figure. 9.

Figure (a), contact pressure maps of gear pair without defect. Figure (b) and (c) are respectively the contact pressure maps with defect.

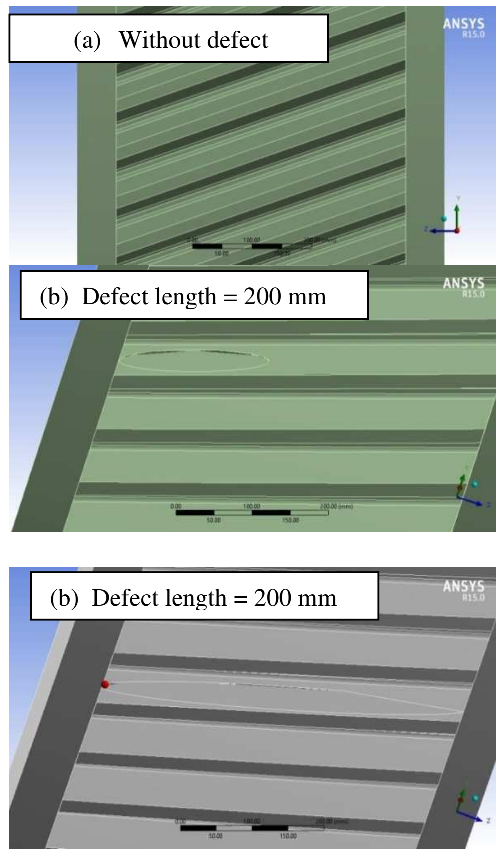

Fig. 9. Different defects size

5.1 Results comparison between the theoretical Lewis bending

The gear tooth bending stress results calculated from the 3D FE model of the gear train without defect were compared to gear tooth bending stress results calculated using the Lewis equation. Figure 10 shows the comparison between the FEM simulation results and theoretical calculation results for gear tooth bending stress.

The two methods show that there is linearity between the numeric calculation by finite element and the theoretical calculation by the Lewis equation. However, the FEM stress results are slightly lower than the one calculated from the Lewis formula. This is because FEM takes into account the radial load component of the resultant force exerted from the torque load, which causes lower stress results.

The percentage difference between the theoretical and FEM stress results is of average 7%, which is still acceptable.

Therefore, this validates the calculated FE stress results and also the FE model of the gear system.

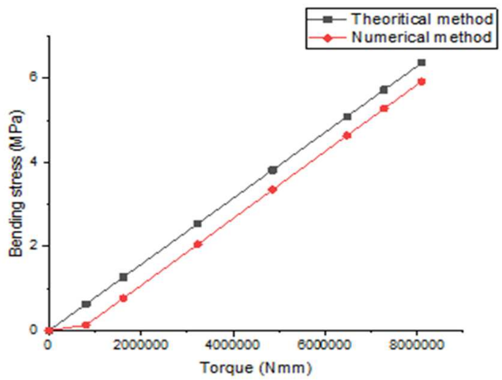

Fig.10. Lewis theoretical stress results and the FEM simulation stress results.

The figure above presents the variation of bending stresses in the teeth face with different defects sizes. The results found by the fem method show that the bending stresses increase linearly with the applied torque.

However, the FEM stress results are very high in the case where there are no defects, this shows that the backlash created by the defect between the teeth reduced the bending stresses (poor contact between the teeth).

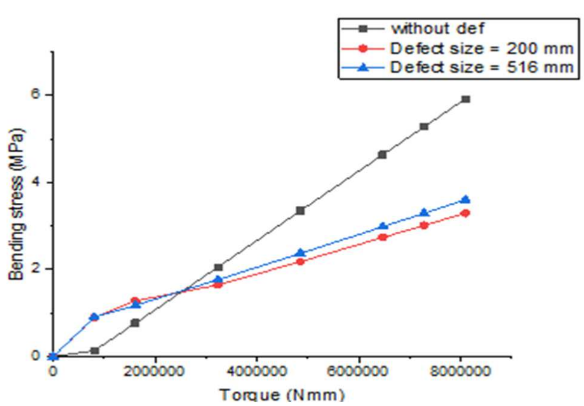

Fig. 11. Evolution of the bending stress in the different cases studied

5.2 Dynamic analysis of gear trains

The figures indicate that both the location and dimension of the maximum contact pressure have certain variations at different defect sizes, As we have noticed, the greater defect, the greater the pressure.

In order to compare the different contact pressure maps of gear pair in the third cases, the maximum contact pressures of helical gear pair in different defects size are extracted and shown in the next figure. It has been observed that the penetration increases linearly as a function of time in the surface of the defect teeth.

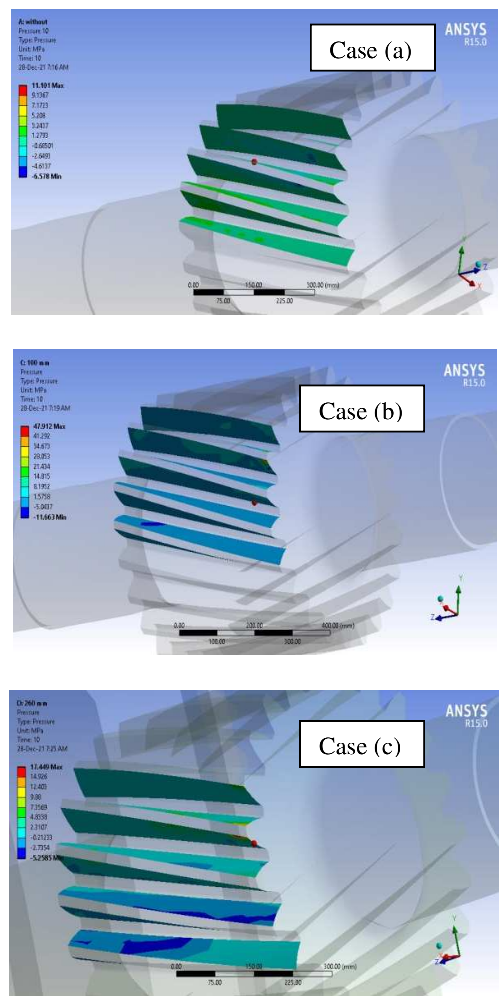

Fig. 12. Contact pressure.

To determine the evolution of the contact pressure along the failed tooth, we are choose the path that shown in the next figure.

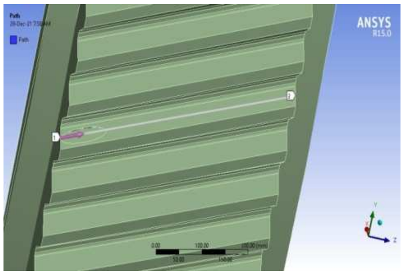

Fig. 13. The path used along the failed tooth.

In Figure 14, it has been observed that there is a high concentration of stresses at the first and last three instants which can cause the early creation of defects and consequently a dulling phenomenon, as shown in Figure 1. In the remaining instants it was observed that the stresses are reduced.

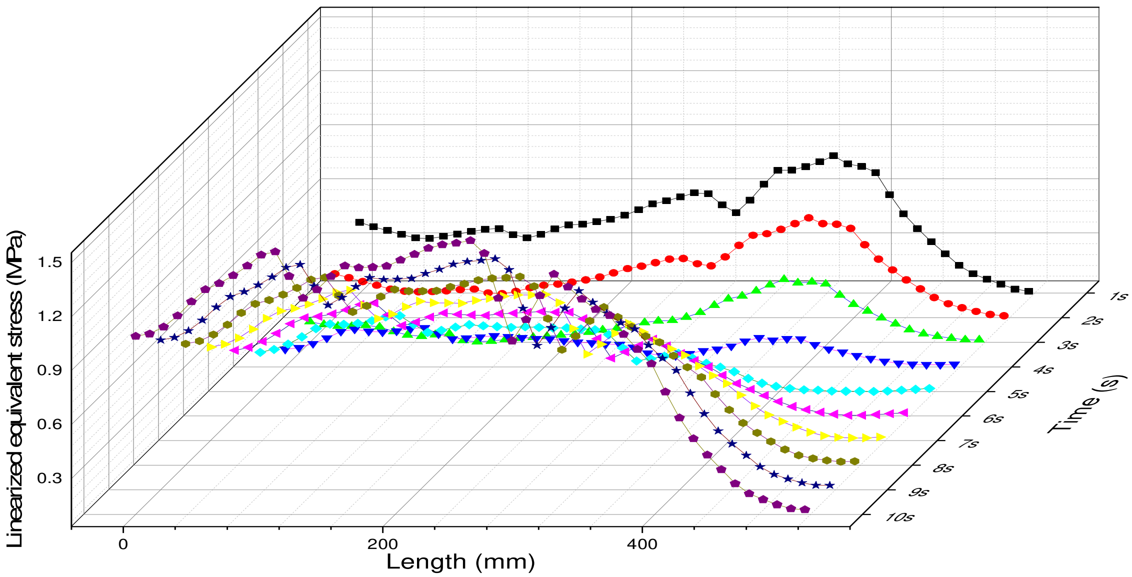

Fig. 14. Evolution of stress along the line of contact (case ‘a’).

In the opinion of many authors, this maximum shear stress is responsible for the surface failure of the elements in contact. The explanation is that a defect originates at the point of maximum shear stress below the surface and progresses towards the surface and then, due to the pressure of the lubricant, the chip breaks off.

3D finite element simulations of teeth in contact showed that the maximum value of the von-Mises stress is slightly below the contact area (instead of the maximum shear stress, i.e. 21 mm below the surface of the teeth), as shown in figure 15.

5.3 Modal analysis of gear trains in free stress state

Figures 16 show the first six mode shapes of the gear train under free stress state and subjected to single pair tooth contact. The three first mode shape of the gear train corresponded to bending vibrations in tow points of crown. In the fourth and fifth mode shape the crown was exposed to bending vibration, whereas the last mode shape show bending vibration in six points. But the pinion was subjected to torsional vibration in the first three modes, in other modes the pinion was exposed to bending and torsional mode shape.

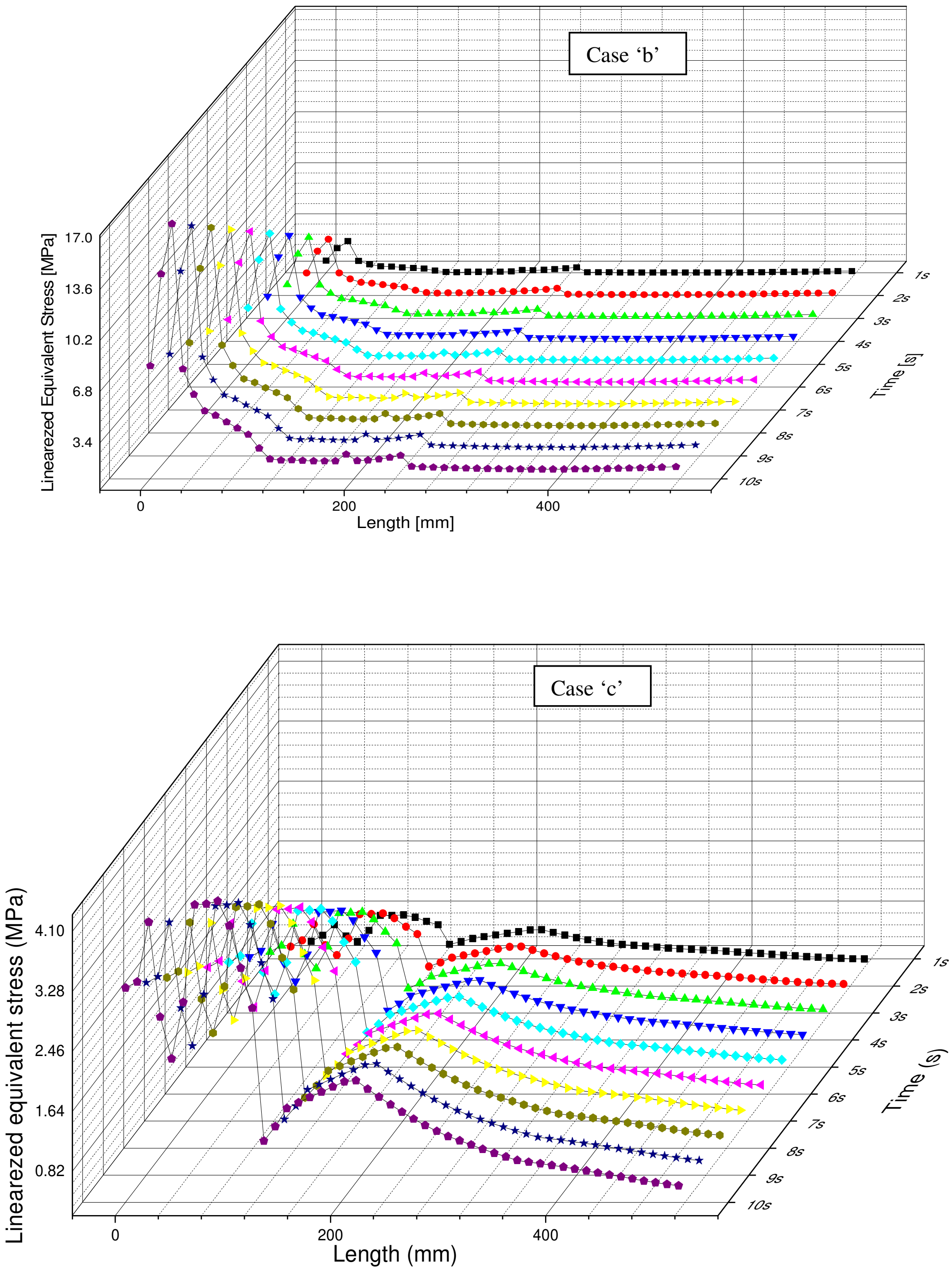

Fig. 15. Evolution of stress along the line of contact (cases 'b' and 'c')

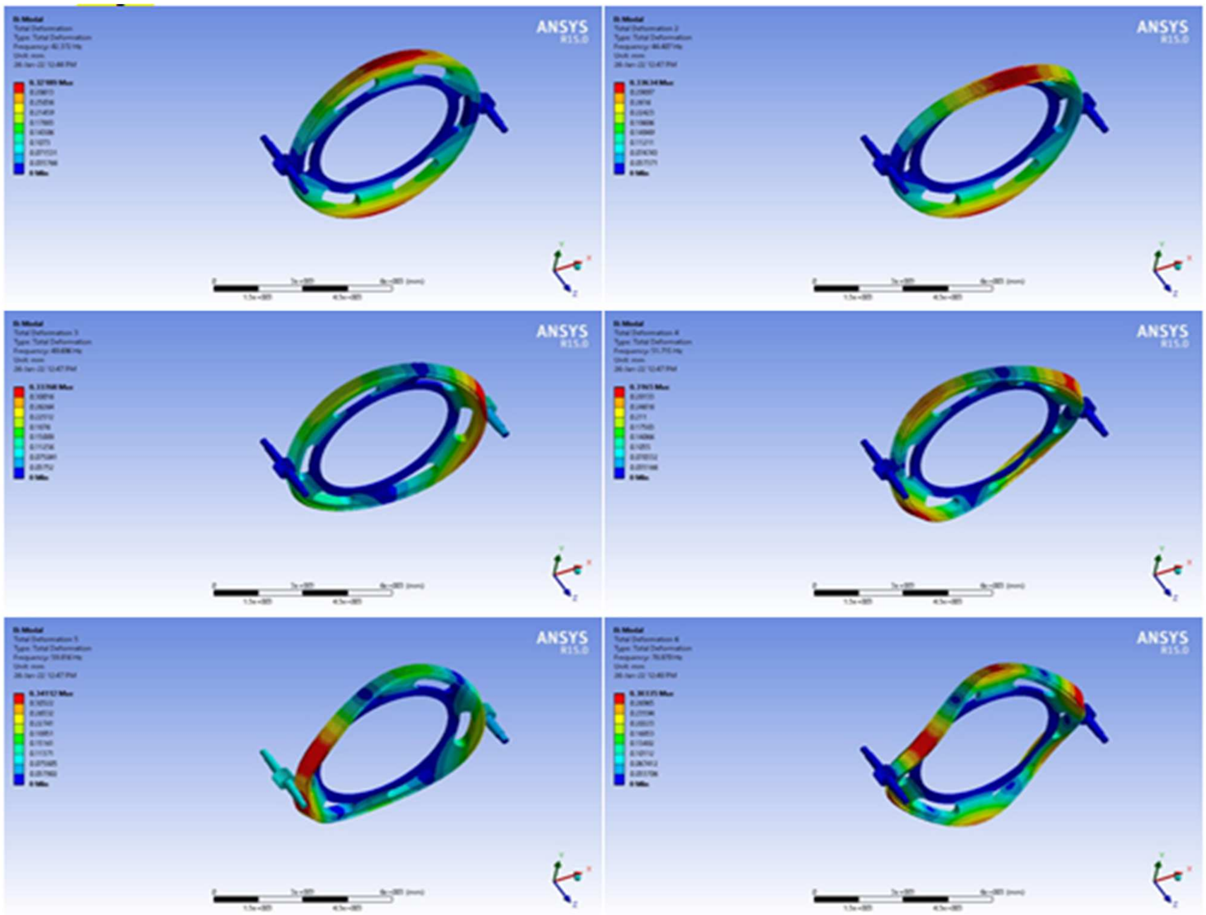

Fig. 16. First six mode shapes of the gear train

Figure 17 shows the first six natural frequencies of the three different gear trains subjected to single pair tooth contact obtained from FE simulations. In comparison, the third to fifth mode shape of the three cases have farther natural frequencies and the other modes have identical frequencies, but they are excited by different types of mode shapes such as bending and torsional vibrations.

The system exhibits quite similar natural frequency characteristics as shown on the graph; however, the fashion shapes are not the same. The natural frequencies of the gear system become higher from the fifth mode shape and so on.

6. CONCLUSION

The influence of gear faults on the vibratory behavior of the system is very important, in this context researchers are trying to find solutions for the importance of these faults. FEM analysis has been use for performing meshing simulation.

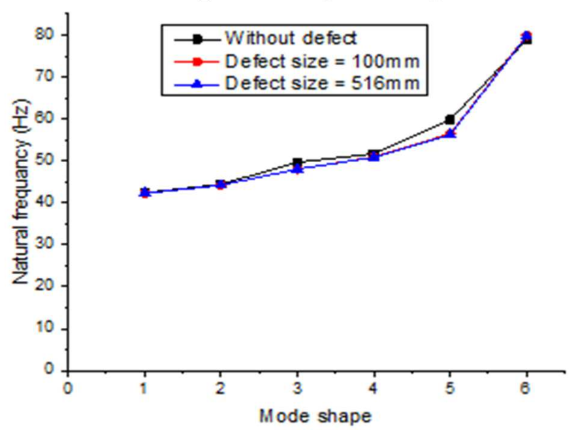

Fig. 17. First six natural frequencies of the three different cases.

The face width and helix angle are an important geometrical parameter during the design of gear.

Our study focused on the study of a gearing defect at the level of the crown of a raw mill of the hamma bouzien cement factory in constantine-algeria. According to the dynamic study of the system, it was found that the presence of the defect increases the stresses along the tooth; it varies between 1.5 to 17 Mpa in case c and from 1.5 to 4.1 Mpa in case b. The concentration of maximum contact pressure changes with gear displacement.

Using finite element method can obtain working status information about each part of gears. It provides some certain references and foundations to structure design and modification, strength analysis and life design.

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ANALIZA STRESULUI DE CONTACT AL ÎMPERECHERII DINȚILOR ANGRENAJULUI ELICOIDAL INVOLUTE DE MOARĂ BRUTĂ STUDIU DE CAZ: FABRICA DE CIMENT DIN HAMMA BOUZIANE. ALGERIA

Rezumat: Zgomotul și vibrațiile sunt principalele motive pentru defectarea sistemului de transmisie, minimizarea zgomotului și vibrațiilor în system de putere este o dezvoltare constantă, defectele

angrenajelor sunt principalele surse care creează vibrații în sistemele mecanice. Conform experienței noastre, pentru a aplica la fabrica de ciment din Hamma Bouziane din Zona Industrială a Constantinului și folosind mijloace de măsurare foarte sofisticate (software de procesare OMNITREND) și dezvoltarea senzorilor de vibrații, a fost detectată o anomalie în piesă (pinion / coroană).
Obiectivul acestei cercetări este de a determina influența dimensiunii defectului asupra comportamentului mecanic al angrenajului elicoidal sub sarcină dinamică și forma diferită a modului de vibrație a pinionului-coroană. Distribuția tensiunilor de contact, a tensiunilor de îndoire a fost investigată cu ajutorul FEM, iar solicitările obținute sunt apoi comparate cu standardul AGMA. Modelarea angrenajelor a fost efectuată cu ajutorul software-ului SOLID WORKS și importată în codul elementului finit ANSYS.

Cuvinte cheie: Pinion, Coroană, Moara brută, Sarcină dinamică, Ansys, Omnitrend.

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