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CRACK GROWTH DIAGNOSTIC OF BALL BEARING USING VIBRATION ANALYSIS

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It is known that supported ball bearings have great effects on the vibrations of the gear transmission system, above in all the presence of local faults as well as the crack growths. For this purpose, this paper focuses on shock and vibration crack growth diagnostic of ball bearing using vibration analysis. Our work is devoted first to a study the static behaviour of the ball bearing by determining the stress, strain and displacement, then its dynamic behaviour by determining the first four natural frequencies. Secondly, a dynamic analysis study of the bearing was carried with defects as a function of crack size and location. The obtained results clearly show that the natural frequencies decrease in a non-linear way with the growth of the length of the crack, on the other hand the stress increases with the presence of the singular points of the crack. Finally, this residual decrease in natural frequencies can be used as an indicator of the state of failure, as well as a parameter used for the diagnosis and screening, and to highlight the fatigue life of the bearing

Key words: ball bearing, vibration analysis, frequency, cracking, diagnosis, fatigue life.

1. Introduction

Bearings are one of the most important parts of rotating machinery. However, normal operating conditions will cause fatigue, resulting in a defect called bearing fatigue spalling. Machine monitoring is not limited to detecting the existence of defects. A thorough diagnosis to locate and quantify precisely the severity of defects must also be made. Vibration analysis is a technique that allows this kind of diagnosis because it is one of the most successful techniques used for monitoring rotating machines [1].

Recently, several research works have been carried in the area of gear transmission system and the vibration analysis of supported ball bearings problems [2-8] and multitracks [9, 10] in cracked rotating machinery. Even though some works reported a local fault in the supported bearings of the gear transmission system, but the box and shaft were considered as rigid bodies [11-12]. The effect of crack propagation on the

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vibrations of the gear system was investigated by Tian *et al*. by proposing a vibration model [13]. Other researchers presented a model of gear-shaft-bearing systems considering various excitations as a function of time [14] or by using vibration analysis taking into account the dynamic interactions in gearbox speeds [15]. Also, a model has been proposed to manage the interactions between bearings, shafts and gears [16] and to see the effect of mounting configurations on the vibration of a gearbox [17].

Standard condition maintenance techniques used to assess bearing health are widely used in many engineering industries. It is well known that the appearance of pulses in the measured acceleration signal indicates a bearing defect. A great deal of technical content related to time-domain measurement signal processing has been published in the literature and new data analysis methods are constantly being developed and improved. The mechanism of fault-related pulse generation has received considerable attention. The main role of a bearing is to provide relative positioning and freedom of rotation while transmitting a load between two elements in a machine [18]. Bearings are among the most heavily loaded components in structures and represent an initiation of failure. Commonly encountered faults are: spalling, seizure, corrosion etc. All these defects are defined as a crushing of material and cause repeated impacts of the balls on the bearing cages [19]. Rosado *et al.* investigated bearing contact fatigue and spine initiation and propagation characteristics of three bearing materials, both numerically and experimentally [20]. The detection and diagnosis of ball bearing failures, has always been a challenge during monitoring rotating machinery. Specifically, bearing diagnosis has been the subject of extensive research in the area of fault detection and diagnosis. Inigo Bediaga and others focused on revising traditional algorithms for detecting and diagnosing faulty bearings in the spindle heads of heavy milling machines. Different types of defects were deliberately caused on the test spindle head bearing. When the defects were in different stages of development, the predictive effectiveness of several detection methods was studied [21]. Bearing cages and balls were cyclically loaded under fatigue, which causes surface degradation by cracks that lead to spalling and then to the failure of the bearing. These cracks can be of superficial origin or come from the degradation of the under layer of the material. This failure can be detected on a spectral response by the determination of the natural frequencies of the bearing. They correspond to the natural frequencies when a rolling element meets a defect. They are given from the speeds at the points of contact [22]. The contact analysis with the finite element method can easily obtain the stress and strain and its cloud diagram [23]. Research involves a numerical and experimental approach to study the effect of defect size and rotational speed on vibration amplitudes and defect frequencies. The work focuses on measuring the vibration amplitudes of the test bearings experimentally in an operating speed range of *1.000* to *5.000 rpm*, with defect sizes ranging from *0.25 mm* to *2.00 mm* at the outer and inner rings [24].

2. Behavior of ball bearing

The work focuses on the development of a mathematical model for a deep groove ball bearing with a single bearing arrangement. The efforts were made by measuring the vibration amplitudes of the test bearing by FEM. The frequencies depend on the ball diameter (*d*), the bearing diameter (*D*), the number of balls (*N*), the contact angle (υ) and the relative rotational speed between the inner and outer rings $\left(f_r\right)$ and are given by the following equations [25]:
Frequency of a localized defect on the outer ring of the bearing (1)

$$f_{be} = \frac{N}{2} f_r \left[1 - \left(\frac{d}{D} \right) \cos\varphi \right]. \tag{2.1}$$

Frequency of a localized defect on the inner ring (2)

$$f_{bi} = \frac{N}{2} f_r \left[1 + \left(\frac{d}{D} \right) \cos\varphi \right]. \tag{2.2}$$

Frequency of a localized defect on the ball (3)

$$f_b = \frac{D}{d} f_r \left[1 - \left(\left(\frac{d}{D} \right) \cos\varphi \right)^2 \right]. \tag{2.3}$$

Frequency of the cage defect (4)

$$f_c = \frac{1}{2} f_r \left[1 - \left(\frac{d}{D} \right) \cos\varphi \right]. \tag{2.4}$$

The bearing is a mechanical device that allows rotation between two shafts or between a shaft and housing in good guiding conditions and with minimum energy loss. The rotation is allowed by rolling bodies (1) separated by a cage, which roll on the inner (2) and outer (3) rings [26-27].

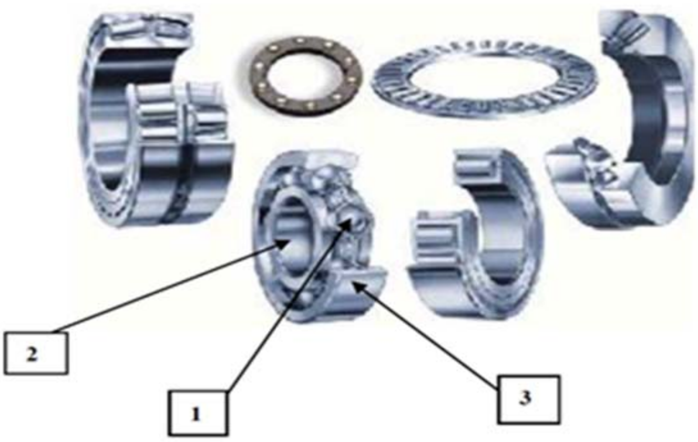


Fig.1. Constituents of a ball bearing element.

Physical defects can occur on bearing elements such as the inner ring, outer ring, and ball. These defects cause vibrations of great amplitude.

Fig.2. Inner ring with defect [28].

The geometry of the bearing is shown in Fig.3 and its dimensions are as follows:

- outer diameter, $D_o = 42\,mm$,
- bore diameter, $D_i = 20\,mm$,
- pitch diameter, $d_m = 31\ mm$,
- raceway width, $B = 12\,mm$,
- ball diameter, $D = 6.35\,mm$,
- contact angle, $\alpha = 00$,
- raceway diameter of outer ring, $d_o = 34.8\ mm$,
- raceway diameter of the inner ring, $d_i = 27.2\ mm$.

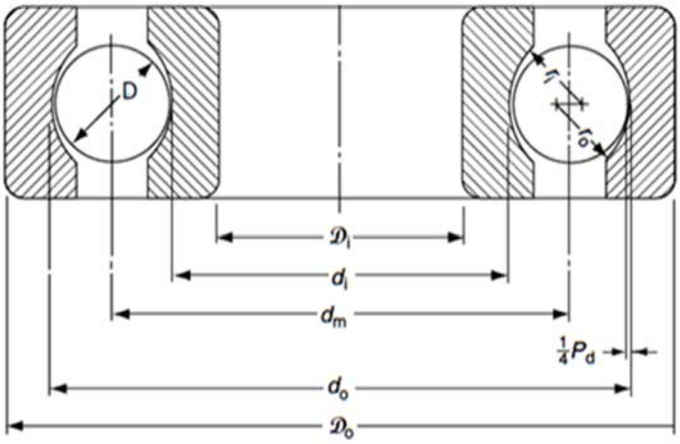


Fig.3. Dimensioning of a ball bearing.

Finite element analysis offers many possibilities for the analysis of structures. The first complete bearing model composed of several elements is modeled by the ABAQUS calculation code. Therefore, a steel bearing system is modeled as a deformable body by inserting the following material properties: density, $\rho = 7800\,kg/m^3$, the modulus of elasticity, $E = 203\,GPa$ and Poisson's ratio, $\upsilon = 0.3$.

The mesh for the three-dimensional model is presented in Fig.4.

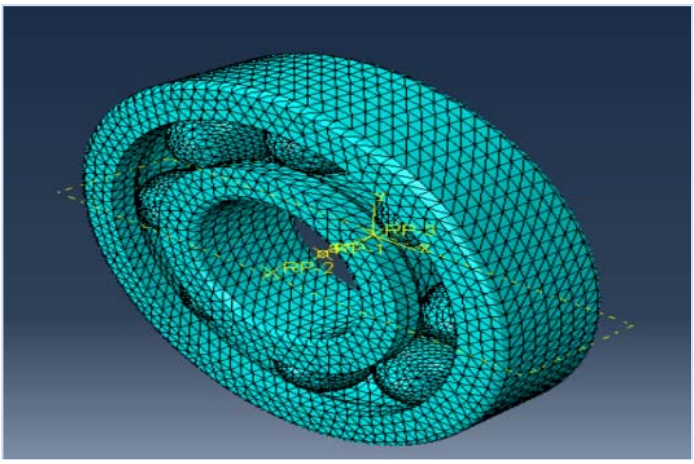
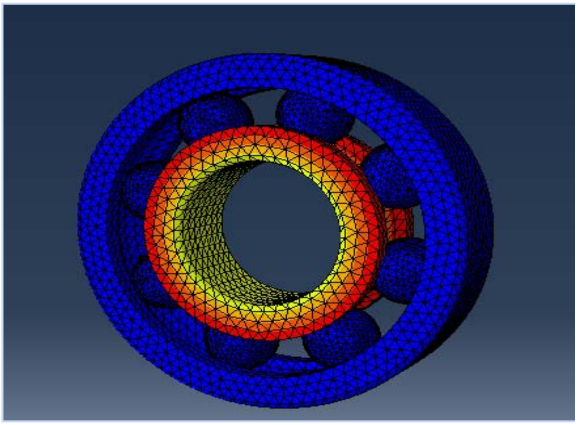


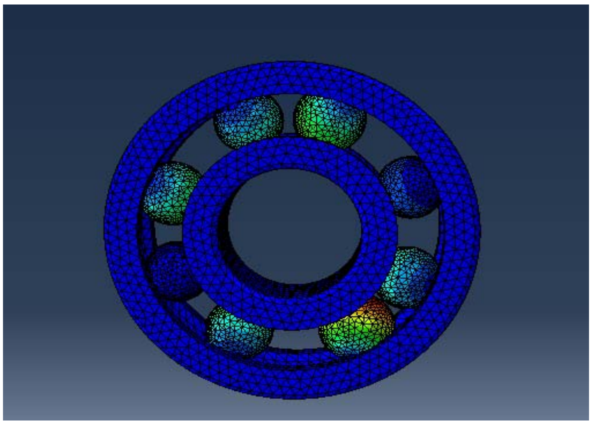
Fig.4. Design and meshing of a bearing.

The following figures show the proper modes of rolling:

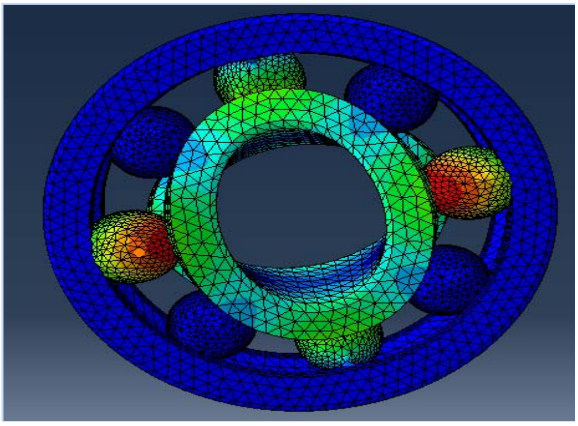
a)



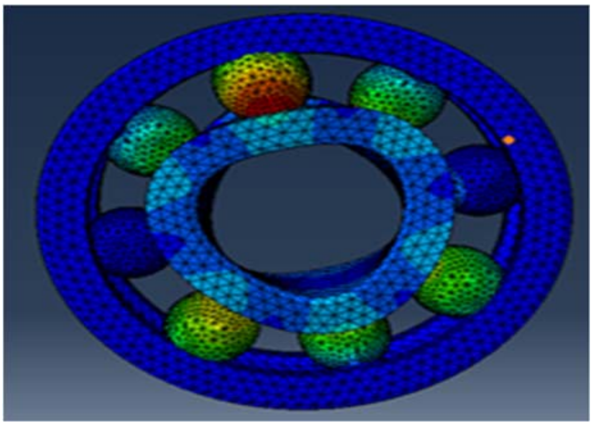
b)



c)



d)



e)

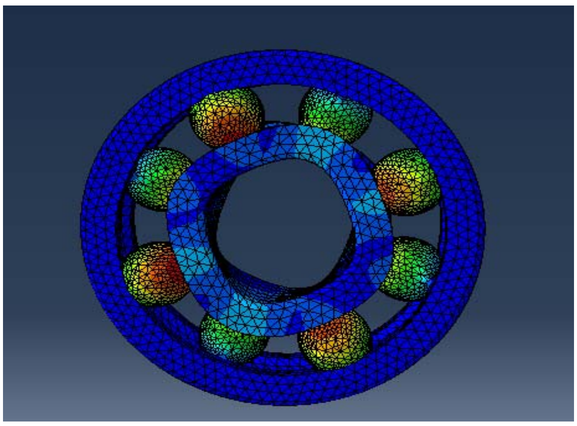


Fig.5. Proper modes of rolling: a) mode n°*1*, b) mode n°*2*, c) mode n°*3*, d) mode n°4, e) mode n°5.

The defect on the inner ring of the bearing is a crack from an initial length of $a_i = 0.2\,mm$ to a final length $a_f = 10\,mm$ as shown in Tab.1. The objective is to determine the natural frequencies of the bearings with cracks.

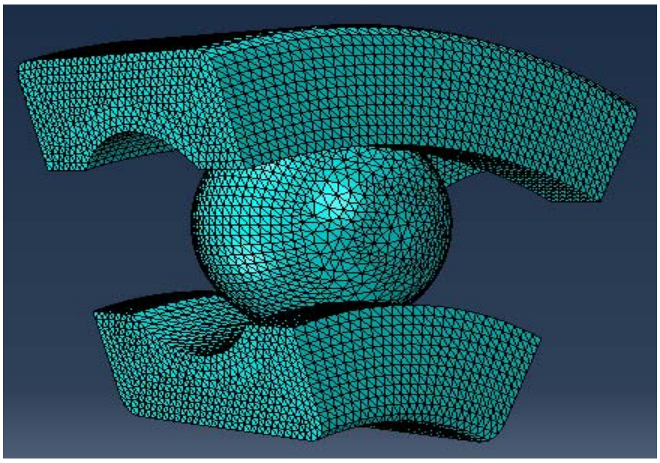


Fig.6. The mesh of the three-dimensional sub-model.

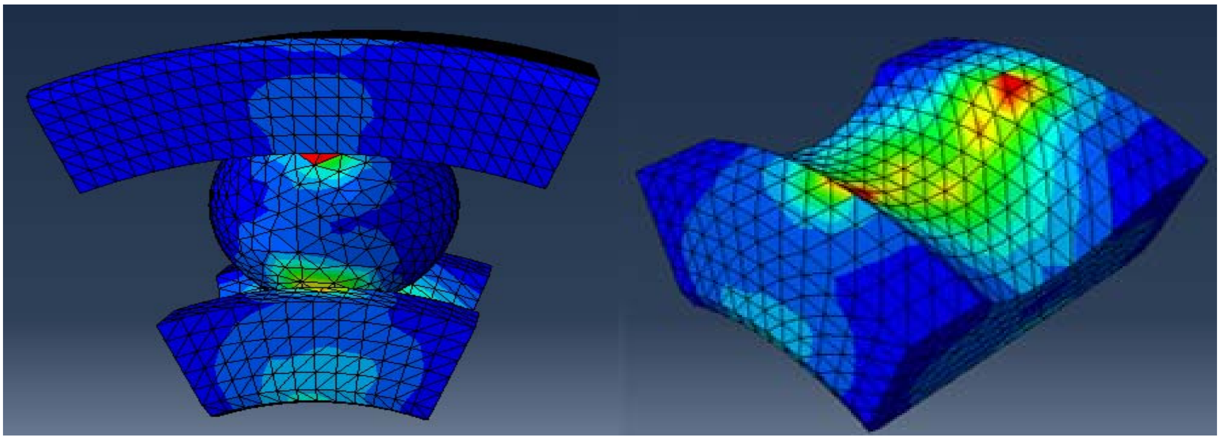


Fig.7. Stress distribution around the crack.

The second model is constructed as an optimized three-dimensional sub model for the bearing. It also assumes the same bearing material (steel), with a friction coefficient for the characterization of the contact surface between the ball and the inner and outer rings. The following figure shows the contact stress distribution between the bearing elements from the three-dimensional model.
It was noted that the stress is singular at the point of the crack with an increase $S = 105\,MPa$.

3. Fatigue of ball bearing

In our case, and according to the stress study, the crack initiation occurs on the inner cage where the singular point is located.

It should be noted that the stress is singular at the point of the crack with an increase $S = 105\,MPa$. The values obtained before and after cracking are presented in the tables and graphs below.

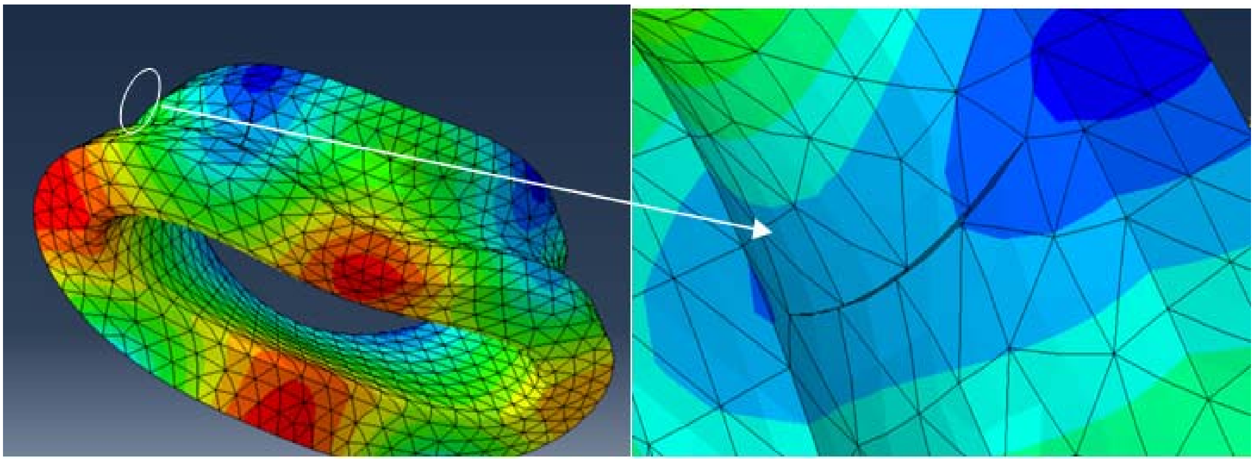


Fig.8. Crack opening with the deformation of the modes.

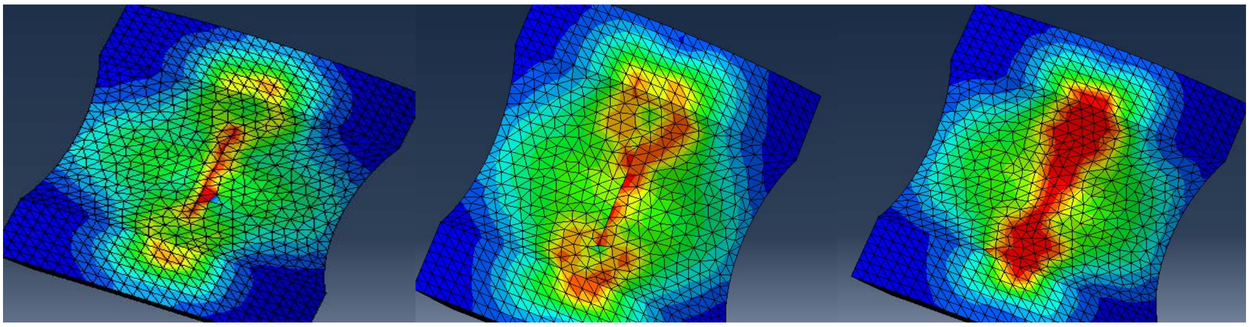


Fig.9. Stress distribution around the crack.

Table 1. Temperature and frequencies in the three areas covered by the study.

$a(mm)$	$F_1(Hz)$	$F_2(Hz)$	$F_3(Hz)$	$F_4(Hz)$	$F_5(Hz)$
0	0.86	3.35	10.03	12.38	13.02
2.5	0.38	0.32	8.40	10.11	11.50
5	0.31	3.32	7.53	10.01	11.47
7.5	0.29	3.31	7.75	9.94	11.42
10	0.29	3.31	7.44	9.90	11.28

Table 2. Five first max displacements according to the size of the crack.

$a(mm)$	$U_{\max}(mm)$	$U_{\max}(mm)$	$U_{\max}(mm)$	$U_{\max}(mm)$	$U_{\max}(mm)$
0	0.054	0.197	0.093	0.056	0.083
2.5	0.054	0.199	0.091	0.097	0.128
5	0.055	0.185	0.085	0.094	0.116
7.5	0.055	0.172	0.089	0.091	0.124
10	0.055	0.182	0.090	0.092	0.380

Figure 10 shows clearly that the values of the first five frequencies decrease in an important way with the initiation of the crack, then slightly with the propagation of the crack.

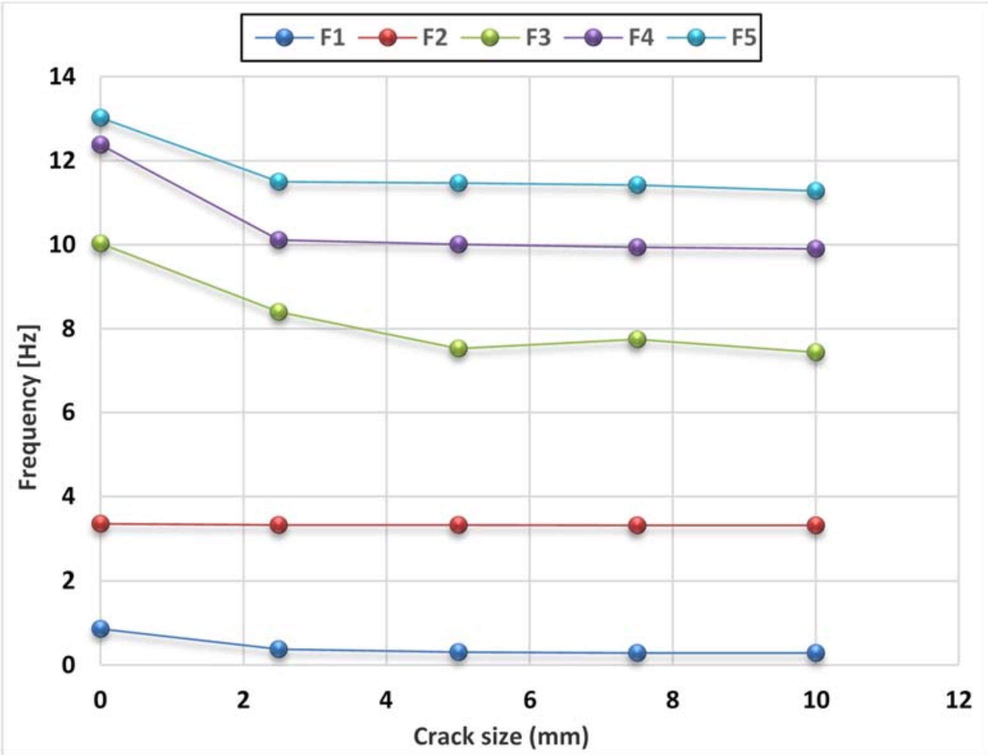


Fig.10. Five first frequencies according to the size of the crack.

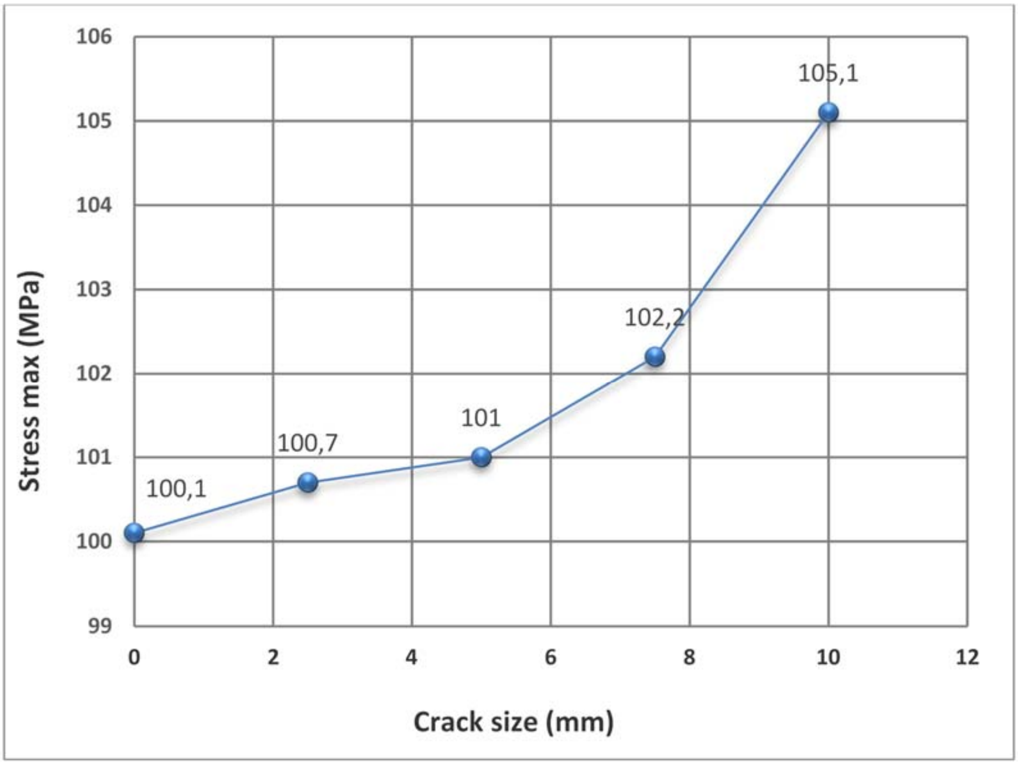


Fig.11. Stress as a function of crack size.

Table 3. Five first max displacements according to the size of the crack.

$a(mm)$	0	2.5	5	7.5	10
$\sigma(MPa)$	100.1	100.7	101	102.2	105.14
$U(mm)$	0.315	0..306	0.119	0.282	0.216

Figure 11 illustrates stress evolution as a function of crack size. According to this figure, it can be stated that the stress increases in a non-linear way with the growth of the crack length.

4. Comparison and validation of results

Bastami and Vahid [29], studied experimentally the trend of important statistical characteristics as a function of defect size. The main findings were that the RMS square of acceleration and defect size have a linear relation. At the first stages of defect growth, the peak amplitude rises sharply but then, the slope decreases considerably. When a defect grows in a bearing, the CF increases quickly and then decreases. At the onset of defect growth, kurtosis increases quickly and then decreases. Kurtosis has the highest value in the ball defect, followed by the inner ring defect and the outer ring defect. Also, Djebili *et al*. [30] studied the operation of a thrust ball bearing under normal operating conditions (axial load, shaft rotation speed and coolant flow) until its spalling. This follow-up is carried out within the framework of the predictive maintenance of thrust ball bearings by vibration analysis which consists in diagnosing a fatigue defect such as spalling. The purpose of this follow-up is to evaluate the growth rate of the spalling and to estimate the residual life of the bearing before its replacement. On the basis of a module dedicated to the fatigue of thrust bearings and through several tests, they obtained fatigue curves that follow the same trend. These trend curves made it possible to find the evolution phases of the spalling, by measuring the RMS value of the vibration according to the operating time.

5. Conclusion

In this article, we studied the failures of bearings; firstly we studied the dynamic behavior of a complete bearing, knowing that the bearing is a complex component of several elements, by determining the first five modes of natural deformation and their natural frequencies. Then a static study was made by determining the stresses and displacements. This study allowed us to locate the critical zone where the stress is maximum. Secondly, a crack was created on the inner cage at the level of the singular point. The results obtained show that the natural frequencies decrease in a non-linear way with the growth of the length of the crack; on the other hand, the stress increases with the presence of the singular points of the crack. Finally, this residual decrease in natural frequencies can be used as an indicator of the state of failure, as well as a parameter used for the diagnosis and screening. It can also highlight the fatigue life of the bearing.

Nomenclature

a – size of the crack
d – ball diameter
D – bearing diameter
E – modulus of elasticity
F_b – frequency of a localized defect on a ball
F_{be} – frequency of a localized defect on the outer ring of the bearing
F_{bi} – frequency of a localized defect on the inner ring

F_c – frequency of a cage defect

F_r –relative rotational speed between the inner and outer rings

N – number of balls

U – displacement

α – contact angle

υ – Poisson's ratio

ρ – density

σ – stress

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