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Crack identification through natural frequencies and fatigue crack propagation modeling of a shaft

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Abstract: Identifying cracks in shafts requires specific instrumentation. Research developed by Bently and Muszynska (1986) presents an identification by using vibration, showing the possibility of crack propagation monitoring using the orbit technique and associating with the natural modes. Other works such as Lebold et al. (2003) and Lissenden et al. (2007) have correlated experimental and computational results in order to track crack progression. This work presents a prediction method for the natural frequencies monitoring during crack propagation and it can be made with already available instruments in predictive maintenance routine aided by computational simulation tools. To carry out this research, theoretical and experimental methods were explored to determine the natural frequencies of a machined shaft made with SAE 1045 carbon steel which included computer simulation and monitoring of vibration and sound. A comparison of the results was performed to validate the applied methodology and characterization..

Keywords: Crack propagation, shaft, natural frequencies, impact test, vibration analysis.

INTRODUCTION

The development of mechanical defects and the occurrence of fractures in rotating machine shafts are obstacles to asset reliability, especially when it results in unexpected breakdown. The possibility of generating industrial losses has become focus of attention due to the consequences of a failure in the production process. Among them, the phenomenon of fatigue in metals stands out in mechanical systems that produce cyclic loads, this phenomenon can cause the nucleation of cracks in shafts of important rotating machines, something alarming to predictive maintenance teams because they are not aware of efficient day use methods to detect such discontinuities.

Research carried out by Bently and Muszynska (1986) shows the identification of cracks on a shaft through vibration monitoring and the possibility of tracking cracks propagation using orbit measurement, associating them with the natural modes. Works such as the methods of Saez and Navarro (2002), Lebold et al. (2003) and Lissenden et al. (2007) correlate experimental and computational results, using modal tests to identify the natural modes and computational methods modelling the progression of the crack.

This work presents vibrational analysis impact test to track shaft natural frequencies that associated with computational modelling of natural modes progression during crack propagation aims to give to the analyst the ability to identify and track the crack with day use instruments as a reliable way to monitor these potential failures and prevent associated losses by assertive prognostics.

The fatigue phenomenon

The first publications and observations regarding the phenomenon of fatigue date from the last century, Juvinall an Mascheck (2006) relates that cyclic loads were typically treated as static loads, however with high safety factors. Historically, brittle and sudden ruptures in ductile material shafts from railroad cars have been witnessed at first after a short period in service. Norton (2011) notes that decades later, German engineer August Wöhler carried out one of the first and notable investigations into the fatigue phenomenon. Wöhler became a major reference in the fracture mechanics literature due to the development of the graph named S-N curve that characterizes the performance of materials relating their stress to the number of loading cycles on a logarithmic scale.

Mechanical components such as rotating shafts are normally prone to such failure, particularly drive shafts. The failures in axles related to the presence of a crack, in general, are catastrophic and can cause great damage to the equipment, productive losses, in addition to putting at risk the integrity of professionals in the surroundings.

Additionally, Callister (2000) presents the phenomenon of fatigue as the single biggest cause of metal failure, being estimated at approximately 90% of all metallurgical failures. It's then verified the importance of a great academic commitment regarding to fatigue, as well as of the maintenance engineering in understanding how to execute a follow-up that is able to predict how many cycles this component subject to dynamic requests will support before a sudden failure, thus avoiding unforeseen events, accidents, emergency correctives and, consequently, economic losses.

The crack problem and its evolution mechanism

The surface of the fatigue failure, in general, is perpendicular to the direction of the applied tensile stress, such stress can be axial (tension-compression), flexural (bending) or torsional nature as Callister shows (2000). According to Budynas and Nisbett (2008), the loads that occur in machines produce tensions that are called variable, repeated, alternating or fluctuating tensions because they oscillate in time.

The cracking mechanism of a material subjected to fatigue is essentially divided into 3 stages:

- Crack nucleation;
- Crack propagation and;
- Final fracture (ASM INTERNATIONAL, 2012).

The first two stages are predominant during the failure process, being that the first stage can have a short duration, the second stage involves the longest part of the life of the part and the third and final stage is instantaneous (NORTON, 2011). Therefore, a fatigue failure usually represents at least two different zones, one induced by gradually and progressive crack propagation and other with a brittle rupture pattern.

Computational modal analysis and impact test

Computational analysis of a dynamic systems involves mathematical modeling, obtaining the governing equations, solving the equations and interpreting the results. After three-dimensional modeling of the shaft, its natural frequencies can be obtained using the finite element method (FEM). It uses the principles of dynamics and derives the equations that describe the vibration of the system.

According to Rao (2008), "modal analysis or modal testing deals with the determination of natural frequencies, damping factors and modal shapes through vibration tests". According to the author, knowing such modal forms allows us to have a complete dynamic description of the machine or structure because the natural frequency is an intrinsic property of the body in free vibration, mainly determined by its geometry, mass and composition and can be identified by impacting the object. of analysis exciting their natural ways.

The impact generates a harmonic forced response or frequency response function (FRF) obtained through the signal analyzer, such response can be represented by the ratio between response and excitation, that is, a cause and effect relationship that describes the behavior of the system with an input and an output. Bueno (2010) describes the test as a functional bump test or impact test that falls into the category of experimental analysis used in the work in question because every impact excites the natural frequency of the element that was subjected to excitation. It's noted that the pulse duration and frequency response shape are dependent on the mass and stiffness of the structure.

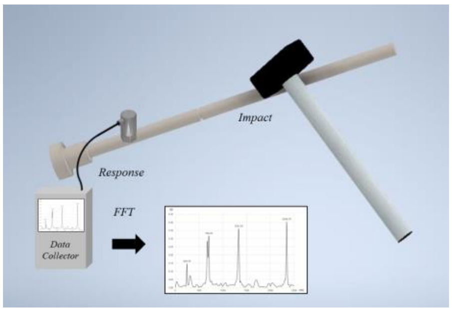

Figure 1 – Impact test schematic model (adapted from Schwarz and Richardson, 1999).

Bump test is a quick and economical means of finding modal parameters from the response to an impact on an element, as the impact in question does not necessarily have a controlled force and the measurement is made at a single response location at a time. According to Bueno (2010), the number of impacts means the number of measures that the analyzer is configured. Additionally, Taylor (2003) warns that impact tests may not be effective on all machines or components, especially on large equipment, due to the magnitude of the required excitation energy.

IMPACT TEST RESULTS

According to Rao (2008), knowing such modal forms allows us to have a complete dynamic description of the machine or structure because the natural frequency is an intrinsic property of the body in free vibration, mainly determined by its geometry, mass and composition and can be identified by impacting the object of analysis exciting the natural modes.

To perform the impact tests, the following items have been used:

- A machined shaft 20 mm diameter, 550 mm length with a central notch;
- A common ball peen 500g hammer;
- A 100mV/g piezoelectric accelerometer;
- Portable vibration analyzer Dynamix™ 2500.

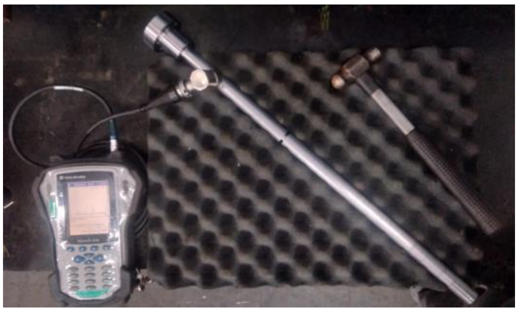

Figure 2 – Impact test conditions.

This test is a quick and cost-effective way of finding modal parameters. In this case a controlled force is not needed, so a regular hammer can be used, and the measurement of the response to the impact is made at a single location at a time. According to Bueno (2010), the number of impacts means the number of measures that the analyzer is configured.

Data collection was performed with the shaft supported on a damping base (Figure 3), to avoid (as possible) the dissipation of vibration energy.

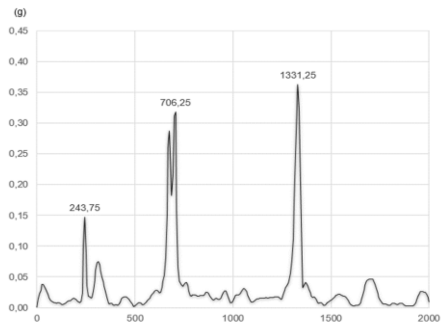

Figure 3 – Impact test results: Shaft main natural frequencies.

The main natural frequencies were observed at 243.25, 706.25 and 1331.25 Hz on the FFT (Fast Fourier Transformer).

COMPUTATIONAL MODELLING

Theoretical analysis of a vibration system involves mathematical modelling, derivation of the governing equations, solution of derived equations and interpreting the results. The model should include enough boundary conditions, equivalent as the reality in order to obtain a fine discretization and reliable results.

The shaft was modeled and the investigation of the natural modes of the shaft have been carried out using ANSYS® Workbench™ FEM (Finite Element Method) software.

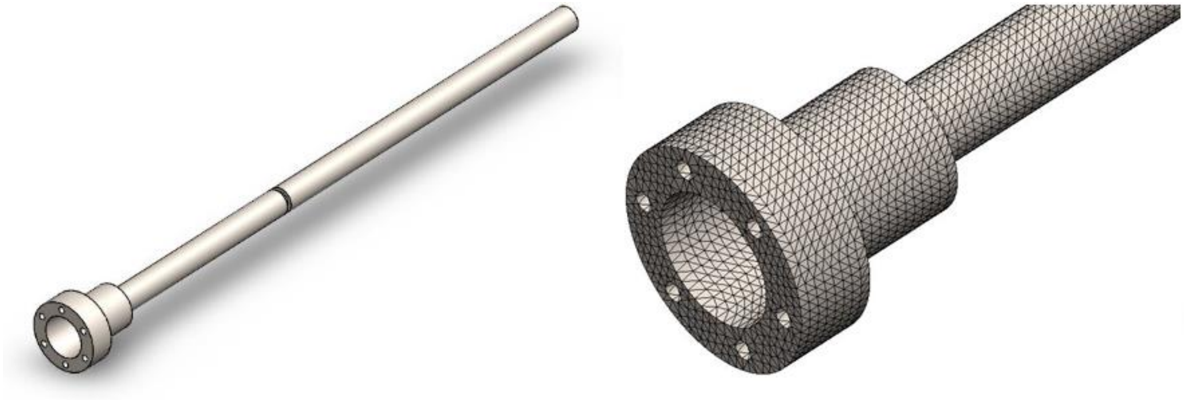

Figure 4 – 3D model with the notched detail on the left. Predominant tetrahedral mesh detail on the right.

Material properties and boundary conditions were applied as using SAE 1045 Carbon Steel. Considering a free body boundary condition was the aproach in order to obtain equivalent frequencies as the impact test as we can see as follow.

Table 1 – Material Properties applied to the shaft model

Shaft material properties - SAE 1045 Carbon Steel	Value	Unit
Young Modulus (E)	190	GPa
Length (L)	0.55	m
Diameter (d)	0.02	m
Mass (m)	1.557	kg
Moment of Inertia (I)	$7.85 \text{ z } 10^{9}$	m^4
Weight per length (mm)	27.77	N/m

The first simulation was obtained with the model of the intact shaft, without cracks. Figure 5 shows in the left the first natural mode as 243.86 Hz, in the middle is the second natural mode as 724.06 Hz and in the right is the third natural mode as 1406.46 Hz.

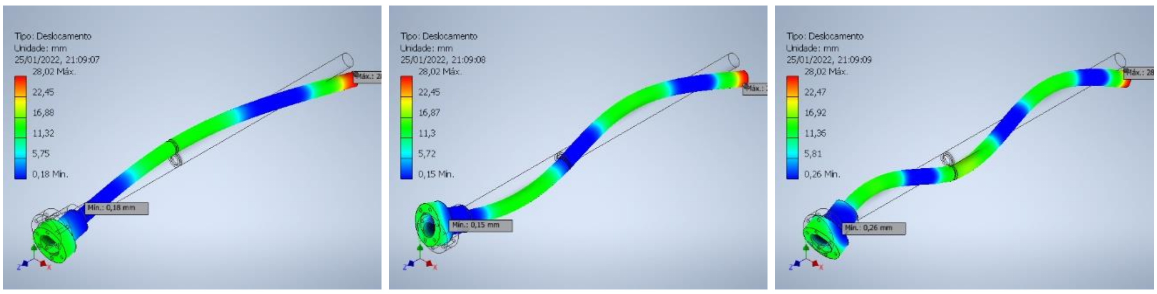

Figure 5 – Modal analysis: The main three natural modes of the shaft with no failure induced.

As the natural frequencies depend on the stiffnesses, masses and damping of the system, the simulation results strongly depend on the applied boundary conditions. In this work, we tried to apply boundary conditions equivalent to those of the experimental impact test, but even so, we noticed a small deviation between experimental and computational results.

The convergence of the values of the natural frequencies referring to the first three excitation modes obtained by the experimental method was verified. First natural frequency shows minimal difference of 0.25%, the second shows 2.46% and third mode 5.35% difference. We impute this difference to the complexity of replicating these conditions in the simulations.

The separation in the central notch section was then applied to the simulation model. It had been modeled on the notch diameter of 14 mm (d) without contact between the faces. Ten successive simulations were carried out, starting at a depth of 1 millimeter (a), advancing one millimeter each simulation until the final depth of 10 mm. In this way, it was possible to follow the evolution of the main natural frequencies of the axis. The approach is derived from Lissenden et. al (2007) considering the faces wouldn't contact or friction during the process.

The figures below show the initial and final stage in the simulation.

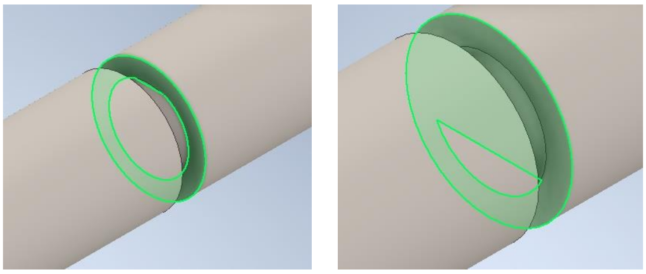

Figure 6 – Application the failure in the three-dimensional model. On the left the initial crack 1 mm deep and on the right the final fracture 10 mm deep.

Failure has been represented up to approximately 80% of the section area, considering that the final stage of the crack, immediately before its abrupt failure, would no longer be a fatigue progression. The natural modes obtained were analyzed and the natural frequencies progression during the propagation can be found below.

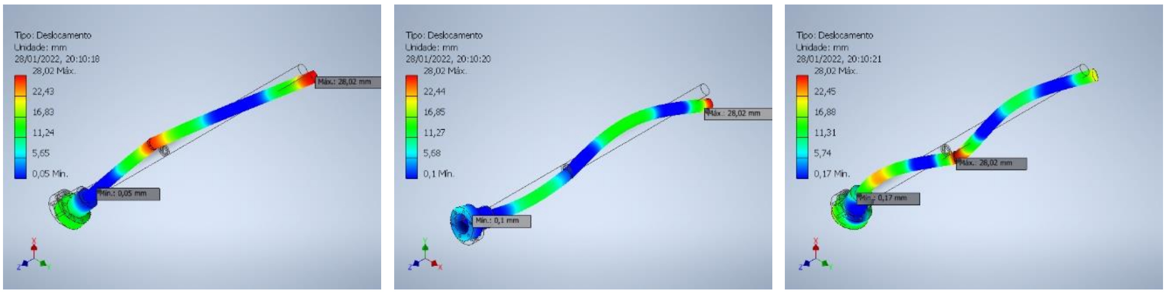

Figure 7 – Modal analysis the main three natural modes with 10 mm depth in the central notch. On the left the first natural mode in 137,92 Hz, on center the second natural mode in 705,61 Hz and on the right the third natural mode in 1193,51 Hz.

Table 2 – Computational results for natural frequencies (Hz) progression versus notch depth (mm)

Natural mode	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00
1st mode	243.86	242.02	240.45	237.88	231.55	224.44	215.07	202.57	185.52	164.60	137.92
2nd mode	724.06	724.06	723.61	723.17	721.87	720.38	718.56	716.46	715.23	713.00	705.61
3rd mode	1406.92	1400.66	1395.05	1384.31	1369.15	1349.90	1326.25	1298.15	1264.22	1229.56	1193.51

It was observed that in the first mode the natural frequency decays in 43.44% (from 243.86 Hz to 137.92 Hz) during crack propagation. In the second mode the decay was 2.55% and in the third mode it was 15.17%. These three main frequencies also had relevant amplitudes in the applied methods.

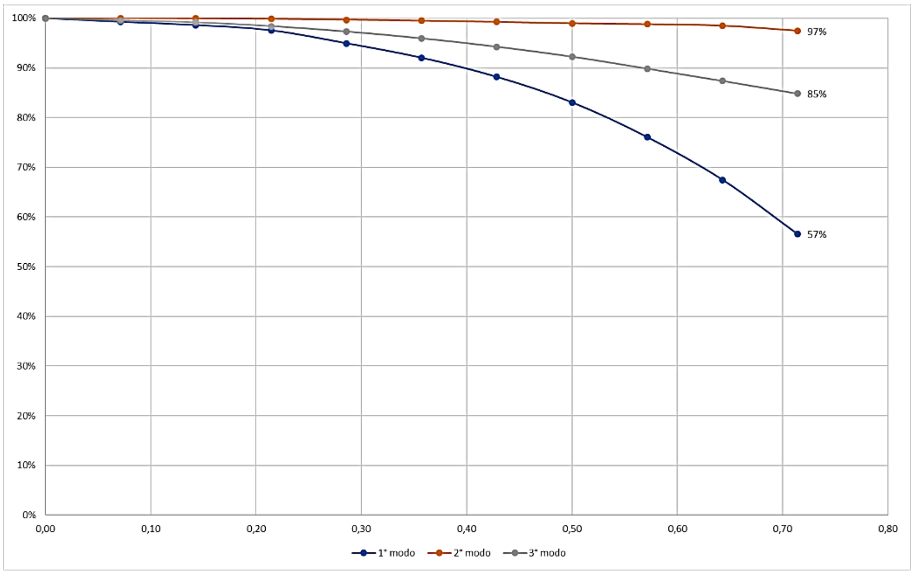

Figure 8 – Three main natural frequencies decay during notch separation progression, The first natural frequency is represented in blue 137,92 Hz, the second natural frequency 705,61 Hz is represented in gray and on the third natural frequency 1193,51 Hz is represented in red.

Figure 8 shows the decay of the first main natural frequency as the crack notch depth increases (a/d) related to the evolution of separation. On the vertical axis, the natural frequency obtained for each of the simulations is represented proportionally as the value of the depth was increased, and on the horizontal axis, the depth is represented in relation to the diameter of the section.

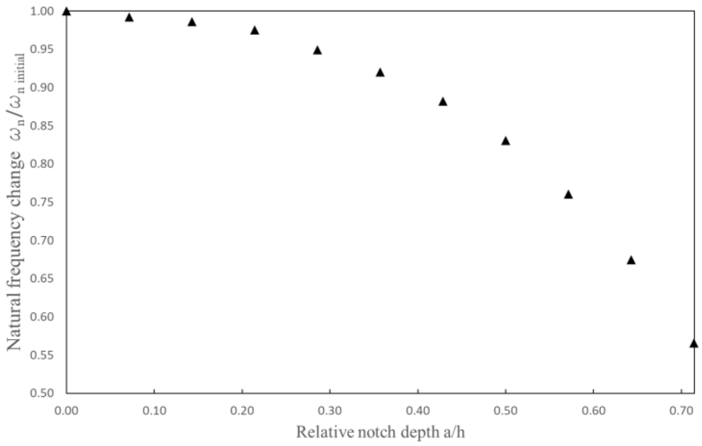

Figure 9 – Application of crack propagation in the three-dimensional model. On the left the initial condition and on the right the final condition.

Figure 9 shows more in detail just the decay of the first main natural frequency as the notch depth increases (a/d) related to the evolution of separation. On the vertical axis, the natural frequency obtained for each of the simulations is represented proportionally as the value of the depth was increased, and on the horizontal axis, the depth is represented in relation to the diameter of the section. It was observed that in the first mode the natural frequency decays 43.44%, from 243.86 Hz to 137.92 Hz.

CONCLUSIONS

This research focused on the theoretical and experimental study of vibration analysis techniques aimed at characterizing the natural frequencies of the axis. The boundary conditions were presented as an influencing factor, that is, to reach the convergence of the results it was necessary to maintain a similar condition between the methods considering a free axis, avoiding the dissipation of vibration energy.

From the gathering of the data referring to the theoretical and experimental approaches collected along the research, a remarkable convergence of the results found in the computational and experimental methods, validating the applied methodologies, reaching the conclusion that it is possible to characterize the propagation of the cracks through vibration monitoring with day use tools.

The first natural frequency was the one that had the greatest decay in relation to its original form, being the preferred frequency to be followed for the purposes of prediction and monitoring of crack progression. It was observed that in the first mode the natural frequency decays 43.44%, from 243.86 Hz to 137.92 Hz. It's an interesting behavior that may allow inspectors to avoid complete ruptures of shafts using vibration analysis monitoring regarding this related frequency progression if it's known in the first hand by performing the impact test taking the initial shaft condition as a baseline.

ACKNOWLEDGMENTS

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