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The Effect of a Polymer-Based Tuned Mass Damper on the Vibration Characteristics of an Anti-Vibration Boring Bar

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Abstract

In recent decades, chatter has been a complex phenomenon facing machining industries. Although several toolholder manufacturers attempting to enhance the potential of the chatter-free boring process, numerous obstacles and downsides are challenging their efficiency. This study aimed to discover the sources of the unpredictable behavior of anti-vibration toolholders with a focus on SECO Tools' Steadyline boring bars. We used experimental modal analysis (EMA) and analytical modeling to detect the sources of nonlinearities and efficiency limitations. Understanding the system's dynamic behavior can help to improve the existing products and provide new technological solutions.

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Keywords: Chatter; Boring operation; Anti-vibration toolholder; Modal Analysis

1. Introduction

Boring, also known as internal turning is a material removal technique. It uses a single-point cutting tool to finish or enlarge the cylindrical or conical internal surfaces of rotating pieces (see Fig. 1). Before surface quality check, it is necessary to control machine-tool vibration, affected by cutting parameters, tool geometry, and bar rigidity. Chatter is a critical problem for the tool-workpiece interface in machining operations because it can reduce tool efficiency and destroy the workpiece's surface (see Fig. 2). In 1904, Taylor [1] identified the chatter problem in machining industry and Arnold's study [2] was the first on chatter theory in 1940s. However, chatter is still today an important issue in machine-tool field. Quintana and Ciurana [3] and Siddhpura and Paurobally [4] are some recent reviews on the subject. A high length (L) to diameter (D) ratio is the primary cause of high amplitude vibrations in boring operation [5, 6]. Thus, as the machine-made hole increases in size, the toolholder becomes less stable making long hole machining more challenging.

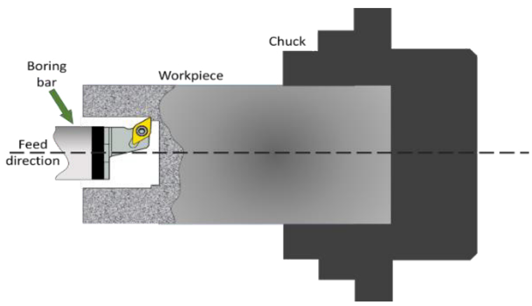

Fig. 1. Mechanics of boring processes.

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Fig. 2. A poor surface finish as an example for chatter.

Nomenclature

EMA	Experimental Modal Analysis
TMD	Tuned Mass Damper
ID	Impact Damper
FD	Friction Dampers
SLD	Stability Lobe Diagram
DAQ	Data Acquisition System
DS	Deflection Shape
EOM	Equation of Motion
m	mass (Kg)
c	damping constant (Ns/m)
k	stiffness constant (N/m)
F	force (N)
x	displacement (m)
sgn	sign function

Active, semi-active, and passive damping are three main strategies for improving stability of boring bars [7]. The entire chain of acquisition, decision-making, and driving systems in active damping solutions alter the dynamic behavior of dampers simultaneously, making it more complicated and costly [7]. On the other hand, passive dampers are effective for a specific frequency range while being low-cost and simple to incorporate.

Passive dampers require installing a spring-mass system in the toolholder that needs to be precisely tuned at the natural frequency of the bar. The three major passive damping techniques are tuned mass dampers (TMD), impact dampers (ID) [8, 9], and friction dampers (FD) [10], which are mainly classified by the damping source [11]. Saindane et al. [12] tried to improve the damping of boring bar by reducing the static stiffness loss. They used a passive damper for it. They compared a passive damper made of nylon and polyurethane with a classic boring. The results indicated a decrease in deflection and an improvement in surface quality. Wadhwankar et al. [13] developed a boring tool constructed of carbon fiber laminated with multiple fiber orientations.This design minimizes the level of noise and frequency of cutting operations.

Prasannavenkadesan et al. [14] developed a boring tool using cartridge brass that was passively fixed. Boring bar with brass damper proved to be more efficient than the identical tool without damper. In addition, Hahn [15] presented the original idea of using a passive Lanchester damper to reduce chatter in machining.

TMD, which is the focus of this study, is a well-known vibration control principle commonly used in many industries, such as manufacturing cars, skyscrapers, aerospace, and machining. A spring-mass-damper mechanism is connected to the structure in this method to dampen the structure's target mode. TMDs are tuned to have the same natural frequency as the critical mode of the structure and break the original mode into two modes [16, 17]. The TMDs' main advantages are their simplicity and consistency. To reduce the equivalent mass of the target mode, the TMD should be located in places where the original mode has large modal displacements [18]. In boring applications, the cutting zone is usually the region of large displacement of toolholders with considerably limited space. To achieve a good combination of weight and space, high-density materials such as tungsten and carbide have been used for the mass of TMDs in boring bars [19]. A variety of more complicated passive damping systems, such as dampers with multiple degrees of freedom [20] and multiple TMDs [21], have been implemented to damp a single mode.

One of the best anti-vibration solutions for boring bars is the Steadyline from SECO tools tooling system (SECO) that has been analyzed in this study. The main idea of this production line is to use a heavy mass placed inside the boring bar, supported by standard polymer parts (commercial O-rings), and between the mass and the inner surface of the toolholder. The elastic elements act as springs in the TMD (see Fig. 3), and as an energy dissipation element.

This study demonstrated the advantages and disadvantages of using Steadyline boring bar, and the major limitations for optimizing the current product in machining processes. The tuning technique of an anti-vibration boring bar has been discussed in the first section of our study. Then, the frequency response of the Steadyline boring bar has been analyzed in the subsequent section. Deflection shape at different frequencies has also been investigated, using a scanning vibrometer. The next section is focused on amplitude dependency investigations and a general nonlinear model has been proposed to account for this nonlinear effect in rubber-like material behavior. In section 2.4, experimental results are presented for repeatability of toolholders from one boring bar to another.

Tuning process

The real and imaginary parts of dynamic response to an impact excitation and stability limit prediction of the bar without TMD are shown in Fig. 4. TMDs are always tuned to the optimal parameters, which dampens the critical mode of the original structure [22]. One should understand the chatter dynamics and apply concepts and methodologies coming from this particular field, which is still in research. The classical

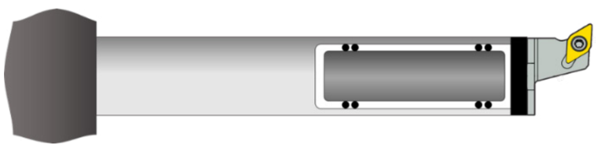

Fig. 3. Steadyline boring bar with internal TMD representation

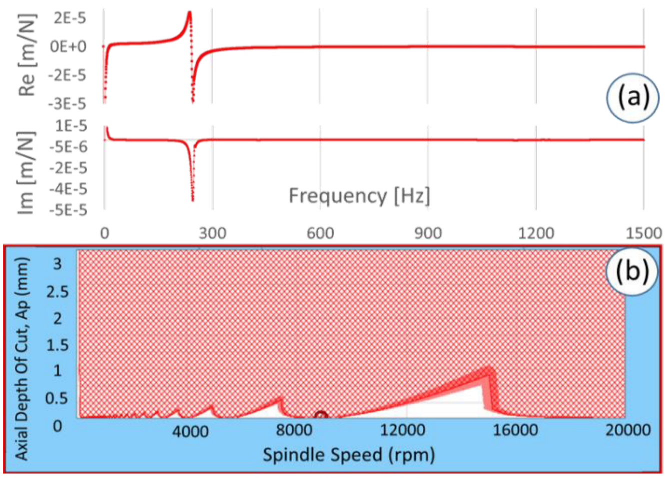

Fig. 4. (a) Real & imaginary part of dynamic response; (b) predicted SLD

way to relate the overall machine dynamics and the stability of the cutting operation in tuning is to look at the imaginary part of the FRF. Hammer test can determine the dynamic behavior of the toolholder when adjusting the Steadyline boring bar. Sometimes the effectiveness of the tuning process is ultimately verified by machining tests, which is a very time-consuming approach. The purpose of this approach should be to reduce the first mode's imaginary part as much as possible. However, SECO field experience showed non-typical behavior and sometimes the best results were reported for boring bars tuned to minimize the real part of the FRF instead of the imaginary part. This is in contradiction with the well-established linear modeling of chatter in continuous cutting processes.

To achieve the best TMD configuration in Steadyline, the material, dimensions, and position of the elastic elements (O-rings) need to be carefully calibrated. METALMAX^{TM} software proposed by Manufacturing Laboratories Inc. is used to process the impact test signals and predict stability lobes diagram (SLD). The evolution of the dynamic response of the bar with TMD is shown in Fig 5. The blue curve is a sample from the tuning process, while the green curve represents the best TMD obtained configuration compared to the other conceivable setups. The absolute values of real and imaginary parts of the best configuration of Steadyline (see Fig. 5) is approximately 10 times less than the bar without TMD (see Fig. 4), showing TMD's significant effect on dynamic response of the bar. SLDs are then computed with METALMAX^{TM}. Generally, they show a considerable change in the stable zone for machining with Steadyline boring bar (see Fig. 6). This linear approach does not take into account the fact that, process damping is influencing the SLDs, but one should consider that this is mostly a comparative method and the tendencies obtained by this approach are usually respected in real world.

The tuning setup for the bars is identical, and each bar is evaluated by hammer test to ensure that it is aligned with the exciting reference. As a result, in terms of production and calibration, all toolholders are equivalent. However, it has been documented by SECO that some of them show different behaviors in service condition. The damper seems to be disabled in some circumstances. Hence, we carried out a

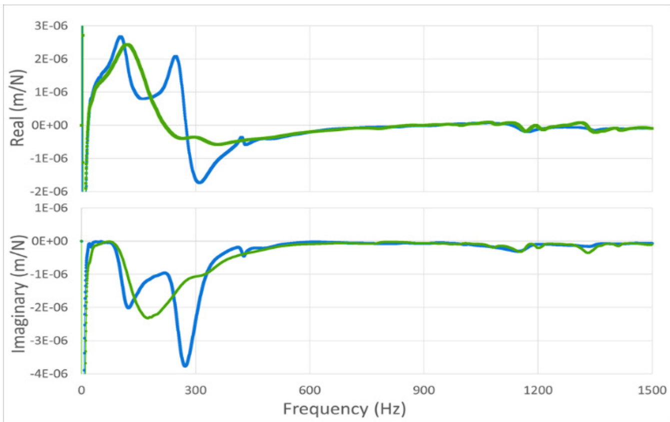

Fig. 5. Tuning process- blue: non-tuned FRF; green: tuned FRF

series of experimental investigations to highlight the various aspects of the product or to identify the causes of issues.

2. Results and discussion

The dynamic behavior of the toolholder is the key to vibration mitigation (passive control). For this reason, the detailed experimentation is important to understand all aspects of dynamic behavior. We first proposed a different way to obtain the FRFs, using electrodynamic shakers and frequency sweeping techniques (continuous or discrete sine-sweep). An LDS shaker device was used to generate the sweep sine periodic deterministic signals for estimating frequency response function. The stimulus (excitation force) and response (acceleration) were measured with a PCB 288D01 impedance head (see Fig. 7). We used the National Instruments data acquisition and processing hardware and software to achieve FRF (LabVIEW Development System with Sound & Vibration Toolbox, NI USB 4431 dynamic signal acquisition card). An example of the magnitude and phase plot for 0-1500 Hz range is shown in Fig. 8.

The first resonance frequency was at 210Hz, an anti-resonance frequency at 755Hz, and a double peak at about 1273Hz, which is probably from the coupling of the second modes from each plane. The natural frequencies after 1500Hz are believed to be out of interest, as their influence in the machining process can be neglected in classical machining operations.

2.1. Deflection Shape analysis

Operational deflection shapes (ODS) are spatial representations of relative information about the motion (or

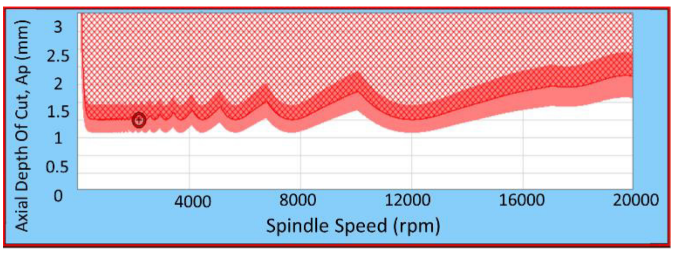

Fig. 6. SLD of the boring bar with best TMD parameters

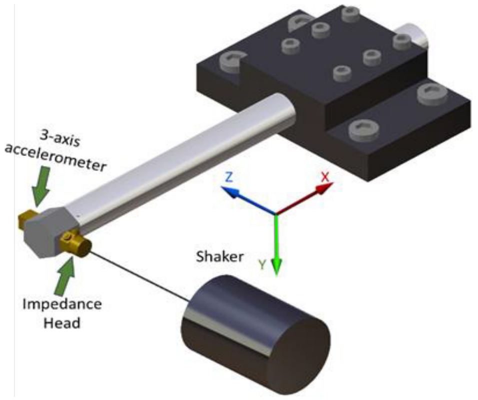

Fig. 7. Experimental setup

deformation) of a structure under a natural process excitation, at a given frequency. Here we use a slightly different notion, the deflection shape (DS), which is the same spatial representation of the structure deflection, but under sinusoidal excitation applied by a shaker. In some classical assumptions, this could be very close to the modal shape (MS) of the structure when the frequency is located at a given natural frequency. DS provides extremely useful results for comprehending and analyzing a component's, machine's, or structure's absolute dynamic behavior. This visual representation of the way that the structure behaves is precious for understanding the system dynamics. It can lead to the discovery of the best modifications to manage noise and vibration problems [23].

One or more frequency response measurements can be used to check for the presence of resonance conditions at the critical points discovered with the DS. As they are close to the modal vectors (modal shapes), the DSs can also be predicted from analytical models (modal analysis) by defining the boundary conditions. There are no unnecessary errors introduced as a result of geometric issues, inappropriate boundary conditions, or linearity issues. DS can be considered as the image that would be obtained if a stroboscope was used to freeze a vibrating object at the desired frequency. Therefore, a DS is an observation or visualization of particular dynamic behavior that does not provide the characteristic dynamic properties of a specific structure. DS is typically computed from a set of sampled time-domain responses acquired concurrently or using pairs of frequency domain data sets. An overview of the most widespread DS measurement methods in the time and frequency domain has been given [24]. Our main interest in using DS instead of MS is the simplicity and rapidity compared to a complete experimental modal analysis.

In our case, we used the Polytec PSV-500 Xtra for obtaining the DSs of our toolholders. PSV-500 Xtra is a Doppler effect scanning vibrometer with an integrated camera that visualizes the measuring object and makes the measuring process and post-processing much easier.

The measuring points on the structure are selected on the bar (Fig. 9). The vibrometer controller drives the shaker and for each point; a sine sweep excitation has been carried out. Time-domain data has been recorded and FRFs were computed. Near natural frequencies, the DS is very close to the modal shape of the corresponding natural frequency. A collection of FRF measurements can also be used to obtain modal parameters (natural frequency, damping, and mode shape) [24].

For obtaining the DS of the bar, 18 scanning points were chosen on its length. The first scanning point was on the fixture to make sure the clamping was stiff enough and it did not move. Excitation and the scanning points were in the same plane. The average of 10 sine sweeps was carried out from 100Hz to 3000Hz with the duration of 3.2 seconds and we move on to the next scanning point. In the post-processing stage, FRF was calculated by applying FFT to the signal.
Under 1500Hz, there are three interesting frequencies to investigate 210 Hz (which seems to be the first resonance, Fig. 10. (a)), 1273 Hz (second resonance of the toolholder, Fig. 10. (b/green)), and 755 Hz (seeming like an anti-resonance, see Fig. 10. (b/red)).

A triaxial PCB 356B21 accelerometer was installed on the bar in addition to the scanning vibrometer to acquire accelerations during the scanning operations (see Fig. 7). The spatial movement of the last scanning point on the free end of the bar after integrating the signal twice to have the displacement is presented in Fig. 11. This result demonstrates that at 210 Hz, the bar is almost entirely moving in the direction of excitation and the out-of-plane motion is insignificant. Therefore, we concluded that the scanning vibrometer gives reliable data regarding the in-plane dynamics of the bar. We confirmed that at the natural frequency, the DS could be a reliable representation of the mode shape for this structure. We believe that the machining process stability is influenced only by the first mode of the bar and we focus only on the first mode investigation.

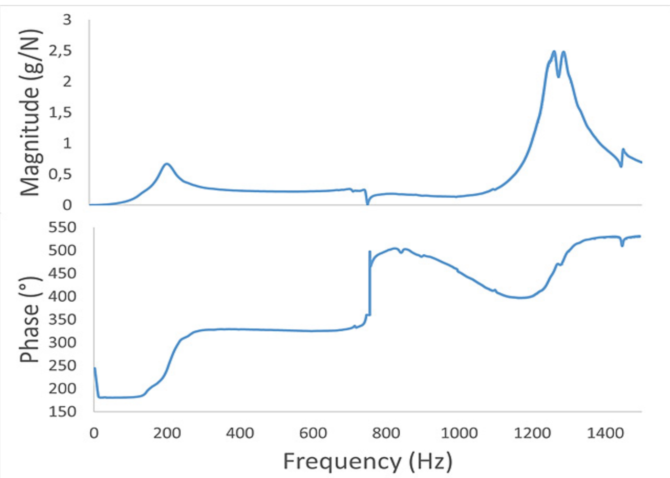

Fig. 8. Gain and phase of Steadyline with shaker test

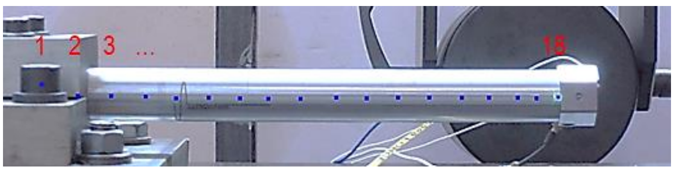

Fig. 9. Scanning points of Vibrometer on the boring bar

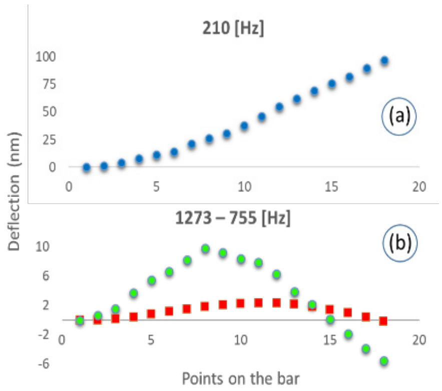

Fig. 10. Deflection shape: (a) 210 [Hz]; (b) 1273 (green)-755 (red) [Hz]

2.2. Amplitude dependency

Excitation in the cutting process is a function of several variables, including the material, cutting tool, cut depth, tool wear, temperature, feed, and angle. It can vary from time to time. The test procedure was replicated with various levels of excitation force. We used the amplitude (voltage level in [V]) of the signal that drove the amplifier of the shaker to compare these different levels. Indeed, the force signal coming from the sensor depends on the system's dynamic response. It is not a suitable or reliable way to compare different excitation levels. The highest excitation signal amplitude was 3 Volt that resulted in a maximum of ±25N force and the lowest amplitude was 0.01 Volt that resulted in a maximum of ±0.1N force at the shaker tip.

The toolholder's response was highly dependent on the amplitude of the excitation force (Fig. 12). One could easily observe that the magnitude peak was higher for lower excitation levels. The difference between the natural frequency of the lowest curve and higher curve was about 35Hz, meaning that this system requires a different tuning for high and low amplitude.

This shows that the boring bar is effective for high amplitude of applied excitation (the lowest curve), which provides a high amount of damping and almost flattens the FRF. However, as the amplitude is reduced, the pick becomes sharper and sharper, to the point where the damper's effect is insignificant.

The explanation for this behavior can be traced back to the current solution's design. Rubbers are used as both damping and stiffness sources in this solution. When using rubber composites with fillers, especially carbon black, amplitude dependent behavior is unavoidable. The Payne effect is a

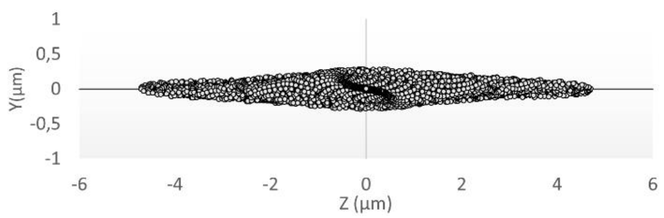

Fig. 11. The spatial movement of last scanning point on the bar in YZ plane

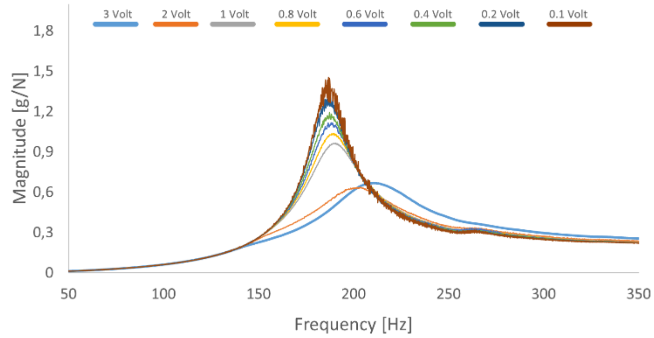

Fig. 12. Gain plot of Steadyline in multiple level of excitation

nonlinear viscoelastic behavior that occurs in carbon-filled elastomers [25]. It is manifested as the dependence of the storage and loss moduli on the amplitude of the applied strain. Above critical strain amplitude, the storage modulus decreases rapidly with increasing amplitude saturating at rather large deformations, while the loss modulus shows a maximum in the region where the storage modulus decreases. The Payne effect depends on the filler content of the material and vanishes for unfilled elastomers.

The impact hammer test is used in the tuning process to obtain optimal combination of design parameters. The impact hammer test is a quick and powerful method to investigate the dynamic behavior of structures if some assumptions are accepted and well-known rules are applied (the hammer size, material tip, sensors sensitivities concerning the frequency's range of interest). It is quick and simple to do, and it covers a wide frequency range. However, because our structure seemed to have nonlinear amplitude dependency, the shaker test was a superior alternative, despite the extra stiffness and mounting challenges. The critical mode is the first mode for this boring bar, as it is for most other anti-vibration boring bars on the market, and the dampers are designed to dampen it. The first mode in our case was around 200Hz, depending on the excitation amplitude. The hammer test excites all modes at the same time with an impulse, but the shaker allows one to focus on a specific frequency band, easily changing the amplitude (which is more difficult to do with a hammer test), and repeating the test with the same parameters.

2.3. Rubber-like material dynamic models

The presented experimental evidence clearly indicates a non-linear behavior. The dynamic behavior for rubber-like materials, which include all the known nonlinear effects, is difficult to model and simulate. Analytical modeling is useful for understanding the dynamic behavior of systems, despite all the required hypothesis. In some cases, a simple linear model can be good enough for some applications, but high accuracy is required for some uses. The more complex the model, the more realistic it can be. However, by increasing the complexity of the model, the calculation cost increases significantly and identification methods become more complex. Models could be based on a phenomenological approach or not. From the practical point of view, a good model should not necessarily be a phenomenological one.

Therefore, here we presented an alternative model, after a short state of the art in this field.

In recent decades, many studies have focused on the dynamic behavior of rubber-like material and tried to present an empirical model for them. Voigt-Kelvin and Maxwell models are the first attempts to model viscoelastic behavior of rubber material using spring and dashpot in parallel and on a series assembly, respectively [26]. Earlier research has revealed that basic linear viscoelastic models cannot accurately predict the rubber's behavior. Hence, we have to use nonlinear viscoelastic constitutive equations. Generalized Maxwell and nonlinear Voigt-Kelvin and their combination were later on carried out to increase the accuracy of modeling of nonlinear behavior of rubber-like material [26]. There are three approaches to modeling nonlinear material behavior [27].

- Empirical function fitting.
- Rational analysis based on a particular theory.
- A combination of the two.

Because the behavior of nonlinear materials is exceedingly complex, the first approach is unlikely to succeed except in very basic scenarios, while the second way is overly complicated. As a result, most authors have adopted the last strategy. Phenomenological and statistical theories were two main approaches to model rubber elasticity. Rubber-like material also shows an amplitude-dependent behavior, so-called Payne effect that is inherent and not avoidable for filler-reinforced elastomers. Rendek and Lion [26] introduced a method based on generalized Maxwell and implemented the method into FEM to model the amplitude-dependence behavior of rubber material. The constitutive model consists of an element responsible for hyperelasticity and a series of Maxwell elements with nonlinear dashpots for viscoelastic behavior. However, other parameters contribute to the sophistication of analytical modeling of rubber-like materials, such as thermal behavior, high deformability, and Mullins effect.

Identifying the material's constitutive model parameters also requires dedicated instruments and increases costs. Meanwhile, the O-rings also include uncertainties, geometrically or materially, which necessitates that the identification procedure must be done on a large number of samples so that the outlier data can be excluded to have a valid identification process. The overall behavior of the boring bar, as well as the influence of each parameter on the process's stability and reliability, has been the purpose of this study. Considering the complexity of constitutive equations drastically increases the computation time, and this is in contradiction with the need of SECO regarding the tuning process.

Therefore, we tried to find a more efficient way to account for the main nonlinear effect that we observed, even if there was no underlying phenomenological analysis. To simulate the nonlinear behavior, the linear equation of motion for the spring-mass-damper system can be adjusted. The equation of motion (EOM) for linear spring-mass-damper system can be represented by Eq.1

$$m\ddot{x}(t)+c\dot{x}(t)+kx(t)=F(t) \tag{1}$$

where m, c, and k are mass, damping constant, and spring constant respectively. For having the nonlinear behavior, the equation needs to contain some nonlinear terms such as Eq. 2.

$$\begin{gathered} m\ddot{x}(t)+c_1\dot{x}(t)+c_2\dot{x}^2(t)\,\mathrm{sgn}\left[\dot{x}^2(t)\right]+c_2\dot{x}^3(t)+ \\ k_1x(t)+k_2x^2(t)\,\mathrm{sgn}\left[x^2(t)\right]+k_3x^3(t)=F(t) \end{gathered} \tag{2}$$

where c_2 and k_2 are coefficients for quadratic terms and c_3 and k_3 are coefficients for cubic terms. This could be seen as a generalization of the Duffing equation (when only k_3 is presented); so, combining these four terms could be used to fit the resulted behavior on experimental data.

Using a chirp force signal as excitation, we simulated what we do on our testbeds in real life, using the shaker. The time-domain response of the Eq. 2 showed that the maximum amplitude of the response depends on amplitude of excitation. The applied force in Fig. 13 (a) is four times smaller than Fig. 13 (b). An estimate of the transfer functions for the difference is also shown in Fig. 14.

The linear spring-mass-damper model cannot represent the nonlinear behavior of the polymer based TMD. However, by adding some nonlinear terms to the EOM, the amplitude dependency was shown in the results. Having a model for the current system can make the tuning process much more reliable.

3. Conclusion

A long overhang boring bar poses a crucial issue in boring operations due to its low dynamic stiffness. Chatter occurs subsequently, which leads to tool breakage and poor surface. Furthermore, it is preferable to maintain a consistent depth of cut and cutting speed in boring operations to reduce rapid

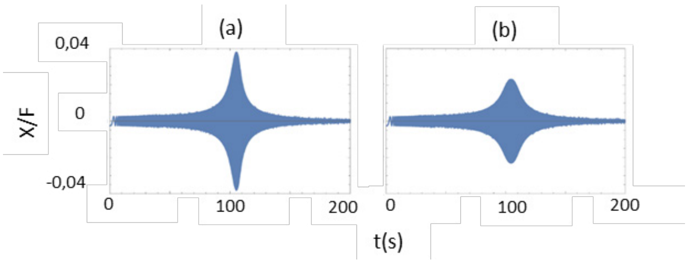

Fig. 13. Time-domain response of nonlinear model with different amplitude

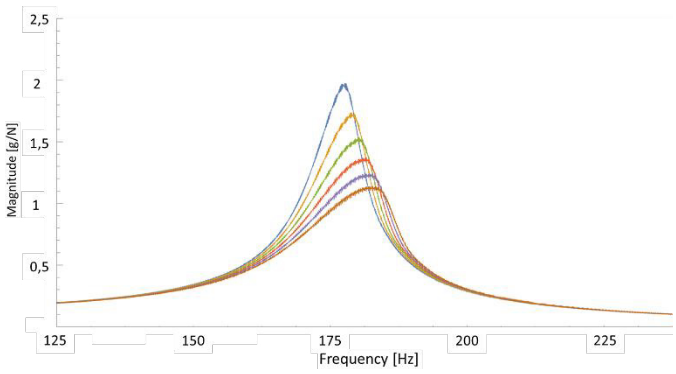

Fig. 14. Transfer function of nonlinear model with different amplitude

changes in inertia and cutting forces. Chatter mitigation is critical for achieving high efficiency in boring operations. In this study, the advantages and disadvantages of a well-known anti-vibration toolholder on the market, called Steadyline, were evaluated using EMA from multiple perspectives to highlight its benefits and limitations. The deflection shape at the natural frequencies and the anti-resonance in the frequency range of interest, was studied that helped to explain the dynamic behavior of the bar. Steadyline technology consists of a heavy mass inside the boring bar, supported by tunable rubber O-rings. The findings showed that Steadyline has an amplitude-dependent behavior because of inherent visco-hyperelastic behavior of rubber-like materials with carbon black fillers. The impact hammer test was used in the tuning process as a criterion for determining the optimal parameter configuration, although it is not a reliable method for nonlinear amplitude-dependent systems. Rubber O-rings' geometrical tolerances and material uncertainties also resulted in unrepeatable dynamic behavior for two identical bars. A simple analytical approach for nonlinear amplitude-dependent behavior was also developed. Identifying the parameters is needed to fit the proposed model to the system to find innovative solutions with better behavior in low-level excitations.

Acknowledgements

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