

Numerical Analysis of Boring Bar Vibration Response in Internal Turning with Spherical Mass–Rubber Dynamic Vibration Absorber (MR-DVA) [†]

Aufa Ulin Nuha *  and Wiwiek Hendrowati

Department of Mechanical Engineering, Institut Teknologi Sepuluh Nopember, Surabaya 60111, Indonesia

* Correspondence: aufaulinn@gmail.com; Tel.: +62-857-3299-5003

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Abstract: Vibration control in boring processes is essential to ensure machining accuracy and stability. This study analyzed the vibration response of a boring bar fitted with a combination of three pairs of spherical Mass–Rubber Dynamic Vibration Absorbers (MR-DVAs) with different stiffness constants during internal turning operations on a lathe machine. A customized boring bar with an internal cavity designed to house the spherical MR-DVA was employed. Modal analysis was performed using ANSYS software version 19.2 to determine natural frequencies, which were then applied in harmonic response simulations to analyze vibration behavior under an excitation force derived from cutting parameters including spindle speed, federate, and depth of cut. The dynamic response was evaluated in three axes (axial, tangential, and radial), highlighting the effectiveness of different rubber combinations. The results demonstrated that the integration of the spherical MR-DVA significantly reduced vibration amplitudes. These findings contribute to optimizing vibration control in deep-hole boring applications.



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1. Introduction

In the machining process for enlarging holes in workpieces, or the boring process, there are several factors that affect the quality of the machining results. One aspect that influences the quality of the machining is vibration. Vibration often occurs during machining processes and can have a negative impact on quality and processing performance [1,2]. Due to this issue, vibration control is regarded as crucial for improving efficiency and accuracy in machining processes [3]. In deep hole boring, a large length-to-diameter (L/D) ratio of the boring bar can be used to control the vibrations that occur [4]. A longer boring bar has lower stiffness and greater mass, which reduces its natural frequency. When the natural frequency approaches the chatter frequency, resonance and high deflection vibration can occur, which may lead to failure in the boring process [5]. Another solution is the use of a Dynamic Vibration Absorber (DVA), which can be employed to dampen vibrations during machining without increasing the L/D ratio of the boring bar [6].

The use of a Dynamic Vibration Absorber as a damper can generate forces that neutralize dynamic responses caused by excessive vibrations or chatter [7]. There are two types of vibration dampers: active and passive. Active dampers involve a system that actively responds to vibration using sensors and actuators for detection and opposing force generation. On the other hand, passive dampers do not require active control but instead

rely on materials or structures with physical properties such as stiffness, mass, and rigidity to reduce vibration amplitude [8]. In the studies by Lu et al. and Grossi et al. [9,10], an active Dynamic Vibration Absorber (DVA) was implemented with optimized actuators to control vibrations during boring operations. Zhang et al. [11] developed a tapered boring cylinder aimed at increasing chatter stability in CNC boring processes. Hayati et al. [12] employed a passive DVA consisting of two pins mounted inside in the boring bar, which resulted in a smoother surface finish. Liu et al. [13] utilized a passive damper in the form of a mass block installed inside the boring bar. Their findings indicated that a higher damper mass could lower the natural frequency. By reducing the natural frequency and moving it away from chatter or excitation frequencies, vibrations were effectively mitigated. Another damping design was introduced by Thomas et al. [14], and used spherical particle dampers capable of attenuating vibrations in multiple force directions. Chaari et al. [15] investigated the influence of ball size in damping, finding that balls with dimensions matching the boring diameter provided superior vibration reduction. These studies collectively highlight diverse approaches to vibration damping in boring applications for both active and passive dampers.

In this study, a passive Dynamic Vibration Absorber system was placed inside a boring bar. The customized boring bar incorporated three pairs of spherical Mass–Rubber Dynamic Vibration Absorbers (MR-DVAs) specifically designed to enhance vibration control. The addition of a spherical MR-DVA was investigated as a vibration control mechanism on a lathe machine. The addition of a mass–rubber system as a Dynamic Vibration Absorber (DVA) can cause a shift in the natural frequency, bringing it closer to the excitation frequency of the system [16]. The spherical shape of the DVA was chosen because it effectively dampens vibrations in multiple directions. Simulations were carried out to evaluate the effects of various spherical MR-DVA placement combinations. The vibration response of the boring bar was analyzed with a spherical DVA mechanism, where stiffness was adjusted by altering the rubber combinations. The investigation was focused on optimizing damping performance by utilizing MR-DVAs with varying rubber stiffness combinations in boring operations. This method of vibration reduction was expected to minimize the vibrations under different cutting parameters, leading to improved quality and performance in the boring process.

2. Research Methodology

This study referred to the parameter of the boring process with lathe machine CDL 6241. Simulations were conducted using ANSYS software. The primary system under investigation was a boring bar, which featured an overhang length-to-diameter ratio (L/D) of 8, indicating a longer overhang compared to the diameter. The primary system of the boring bar comprises three main components, which are the body, the insert, and the bolt that secures the insert. The body acts as the main structure of the boring bar and was the focus of this research. In this study, two types of boring bars were used, regular and customized boring bars. The regular boring bar is a standard configuration without a DVA, while the customized boring bar includes an integrated DVA. The boring bar material is AISI 1045 steel with a density of 7850 kg/m^3 and a modulus of elasticity of 210 GPa. The insert is made of carbide, and the bolt is an M5 size constructed from carbon steel. The boring bar has a total length of 250 mm and a diameter of 32 mm. For the customized design, a cavity measuring 100 mm in length and 25 mm in diameter was created to accommodate the placement of the DVA. An illustration of the primary system is shown in Figure 1. The DVA consisted of a spherical mass made of stainless steel (serving as the absorber mass) and a rubber component (providing absorber stiffness). The DVA was mounted within a housing located inside the customized boring bar. The material of the spherical

DVA operates in a radial direction, mirroring the movement of the main system but in the opposite direction. Positioned within the cavity at the front of the boring bar, the DVA responds to excitation forces generated by the friction between the rotating workpiece and the cutting tool. These excitation forces are continuously applied throughout the machining process. The design of the DVA is detailed in Figure 2.

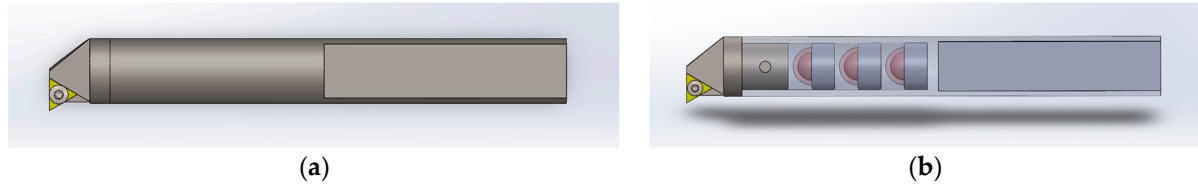


Figure 1. Boring bar: (a) regular; (b) customized.

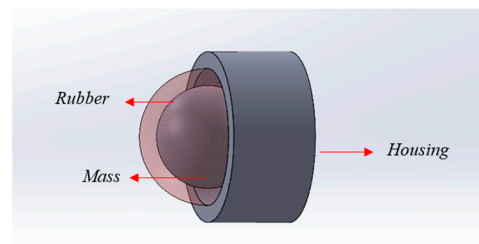


Figure 2. Spherical mass–rubber dynamic vibration absorber.

During the machining process, the primary system is subjected to excitation forces (F_m) generated by the friction between the cutting tool and the workpiece surface. These excitation forces, when decomposed along the x-axis, y-axis, and z-axis, result in thrust force (F_t), cutting force (F_c), and radial force (F_r). These force components are illustrated in Figure 3. These three forces can be expressed as the excitation force of the boring bar as

$$\vec{F}_m = \vec{F}_c + \vec{F}_r + \vec{F}_t \quad (1)$$

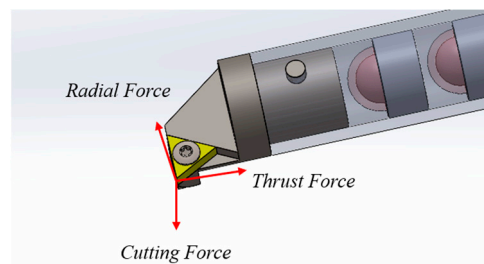


Figure 3. Forces on the tip of the boring bars in the machining process.

The dynamic system model for the primary system with the spherical MR-DVA is shown in Figure 4. The main system's dynamic model without spherical MR-DVA was 1DoF, and the one with spherical MR-DVA was 12DoF. In this model, M_1 , C_1 , and K_1 represent the mass of the primary boring bar system, the damping constant of the primary boring bar system, and the stiffness, respectively. Meanwhile, M_2 , M_3 , M_4 , C_2 , C_3 , C_4 , K_2 , K_3 , and K_4 correspond to the masses of the first, second, and third spherical MR-DVAs, the damping constants of the first, second, and third spherical MR-DVAs, and the stiffness values of the first, second, and third spherical MR-DVAs, respectively. Based on

the dynamic system model of the main system with spherical MR-DVAs added in Figure 4, Equations (2)–(13) were obtained.

$$\begin{aligned} M_1\ddot{x}_1 + C_{1x}\dot{x}_1 + 2C_{2x}(\dot{x}_1 - \dot{x}_2) + 2C_{3x}(\dot{x}_1 - \dot{x}_3) + 2C_{4x}(\dot{x}_1 - \dot{x}_4) + K_{1x}x_1 \\ + 2K_{2x}(x_1 - x_2) + 2K_{3x}(x_1 - x_3) + 2K_{4x}(x_1 - x_4) \\ = F_t + 2C_{2x}(\dot{x}_2 - \dot{x}_1) + 2K_{2x}(x_2 - x_1) + 2C_{3x}(\dot{x}_3 - \dot{x}_1) \\ + 2K_{3x}(x_3 - x_1) + 2C_{4x}(\dot{x}_4 - \dot{x}_1) + 2K_{4x}(x_4 - x_1) \end{aligned} \quad (2)$$

$$\begin{aligned} M_1\ddot{y}_1 + C_{1y}\dot{y}_1 + 2C_{2y}(\dot{y}_1 - \dot{y}_2) + 2C_{3y}(\dot{y}_1 - \dot{y}_3) + 2C_{4y}(\dot{y}_1 - \dot{y}_4) + K_{1y}y_1 \\ + 2K_{2y}(y_1 - y_2) + 2K_{3y}(y_1 - y_3) + 2K_{4y}(y_1 - y_4) \\ = F_c + 2C_{2y}(\dot{y}_2 - \dot{y}_1) + 2K_{2y}(y_2 - y_1) + 2C_{3y}(\dot{y}_3 - \dot{y}_1) \\ + 2K_{3y}(y_3 - y_1) + 2C_{4y}(\dot{y}_4 - \dot{y}_1) + 2K_{4y}(y_4 - y_1) \end{aligned} \quad (3)$$

$$\begin{aligned} M_1\ddot{z}_1 + C_{1z}\dot{z}_1 + 2C_{2z}(\dot{z}_1 - \dot{z}_2) + 2C_{3z}(\dot{z}_1 - \dot{z}_3) + 2C_{4z}(\dot{z}_1 - \dot{z}_4) + K_{1z}z_1 \\ + 2K_{2z}(z_1 - z_2) + 2K_{3z}(z_1 - z_3) + 2K_{4z}(z_1 - z_4) \\ = F_r + 2C_{2z}(\dot{z}_2 - \dot{z}_1) + 2K_{2z}(z_2 - z_1) + 2C_{3z}(\dot{z}_3 - \dot{z}_1) \\ + 2K_{3z}(z_3 - z_1) + 2C_{4z}(\dot{z}_4 - \dot{z}_1) + 2K_{4z}(z_4 - z_1) \end{aligned} \quad (4)$$

$$M_2\ddot{x}_2 + 2C_{2x}(\dot{x}_2 - \dot{x}_1) + 2K_{2x}(x_2 - x_1) = 0 \quad (5)$$

$$M_2\ddot{y}_2 + 2C_{2y}(\dot{y}_2 - \dot{y}_1) + 2K_{2y}(y_2 - y_1) = 0 \quad (6)$$

$$M_2\ddot{z}_2 + 2C_{2z}(\dot{z}_2 - \dot{z}_1) + 2K_{2z}(z_2 - z_1) = 0 \quad (7)$$

$$M_3\ddot{x}_3 + 2C_{3x}(\dot{x}_3 - \dot{x}_1) + 2K_{3x}(x_3 - x_1) = 0 \quad (8)$$

$$M_3\ddot{y}_3 + 2C_{3y}(\dot{y}_3 - \dot{y}_1) + 2K_{3y}(y_3 - y_1) = 0 \quad (9)$$

$$M_3\ddot{z}_3 + 2C_{3z}(\dot{z}_3 - \dot{z}_1) + 2K_{3z}(z_3 - z_1) = 0 \quad (10)$$

$$M_4\ddot{x}_4 + 2C_{4x}(\dot{x}_4 - \dot{x}_1) + 2K_{4x}(x_4 - x_1) = 0 \quad (11)$$

$$M_4\ddot{y}_4 + 2C_{4y}(\dot{y}_4 - \dot{y}_1) + 2K_{4y}(y_4 - y_1) = 0 \quad (12)$$

$$M_4\ddot{z}_4 + 2C_{4z}(\dot{z}_4 - \dot{z}_1) + 2K_{4z}(z_4 - z_1) = 0 \quad (13)$$

The excitation force can be expressed as

$$\vec{F}_m(t) = Fe^{i\omega t} \quad (14)$$

So that the excitation of the boring processes was a sinusoidal force with $Fe^{i\omega t}$, the displacement was expressed as

$$X_j(t) = X_j e^{i\omega t}; j = 1, 2 \quad (15)$$

$$Y_j(t) = Y_j e^{i\omega t}; j = 1, 2 \quad (16)$$

$$Z_j(t) = Z_j e^{i\omega t}; j = 1, 2 \quad (17)$$

The simulation design utilized in this study was categorized into two main types, regular boring bars and customized boring bars with variations in rubber combinations used in the spherical MR-DVAs. The customized boring bars were equipped with three pairs of spherical MR-DVAs placed in an internal cavity. In these variations, natural rubber is represented by “A” and neoprene rubber is represented by “N”. The configurations are illustrated in Figure 5.

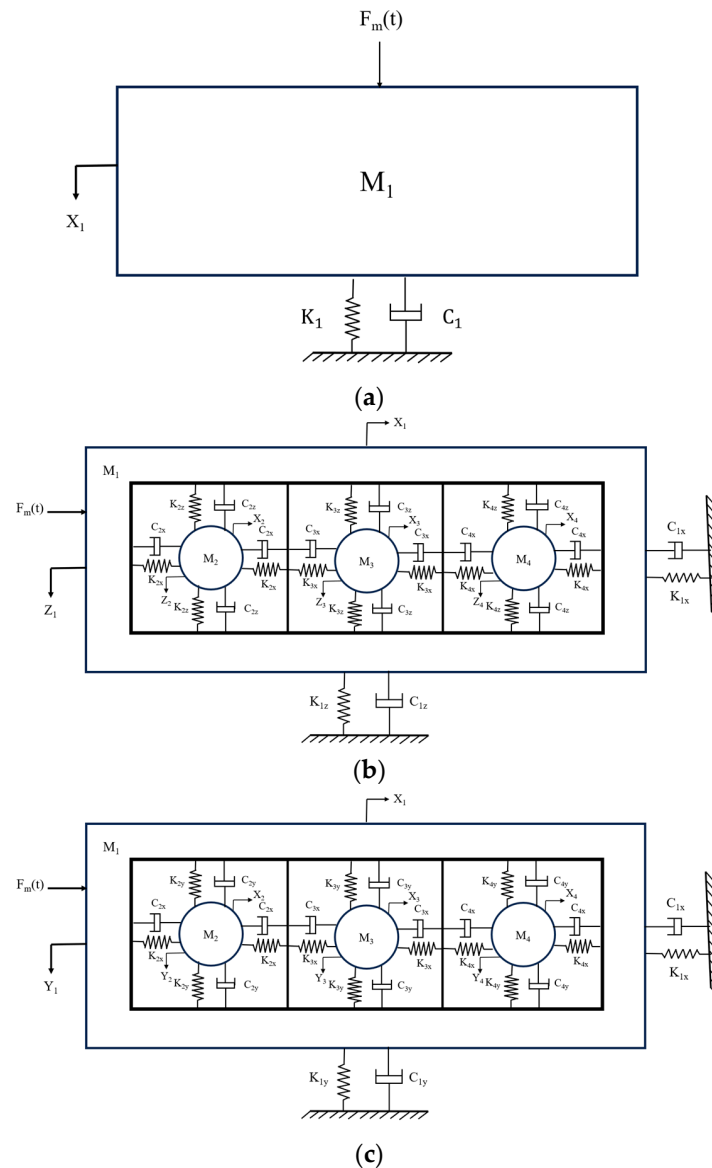


Figure 4. Dynamic model of main system (a) without spherical MR-DVAs; (b) with spherical MR-DVAs (top view); and (c) with spherical MR-DVAs (side view).

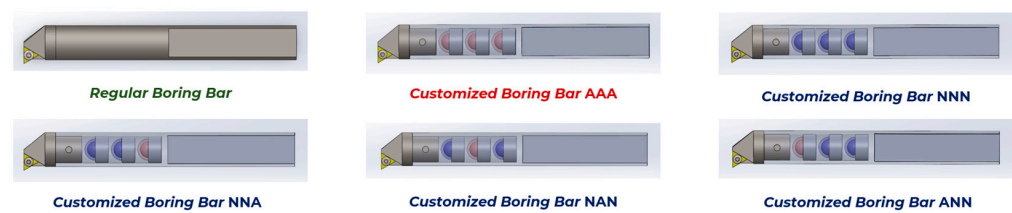


Figure 5. Geometry of boring bars for simulation.

In this study, the cutting parameters for the boring process included spindle speed, feed rate, and depth of cut. The analysis focused on the chatter frequency, which was calculated using the ω_c formula derived from Nwoke et al. [17], as shown in Equation (18). The excitation force for the research was determined based on the cutting parameters outlined in Table 1. The cutting parameters were selected based on the specifications of the lathe machine CDL 6241. The parameters of the boring bar were determined through simulations conducted using ANSYS software version 19.2 (Pittsburgh, USA), as illustrated in Figure 6. In this simulation, a concentrated force of 150 N in the vertical direction (y -axis)

was applied to the nose of the insert, as shown in Figure 7a. The equivalent stiffness constant of the boring bar at the insert edge was obtained by performing a static structural simulation. The resulting static deformation (K_{st}) was used to calculate the equivalent stiffness constant, with the process depicted in Figure 8. The equivalent stiffness constant of the main system was calculated by Equation (19).

$$\omega_c = 17.0 - 1.566v + 3971f + 155d \quad (18)$$

$$K_1 = \frac{F}{K_{st}} \quad (19)$$

Table 1. Cutting parameters.

Spindle Speed (rpm)	Feed Rate (mm/rev)	Depth of Cut (mm)	Chatter Frequency (ω_c)
320	0.1	0.2	388
450	0.1	0.2	366
720	0.1	0.2	318

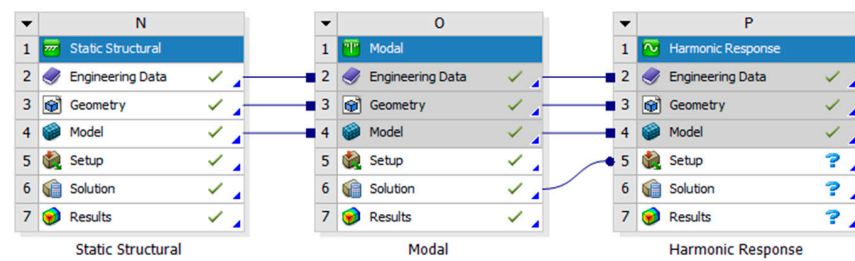


Figure 6. Simulation scheme with ANSYS software.

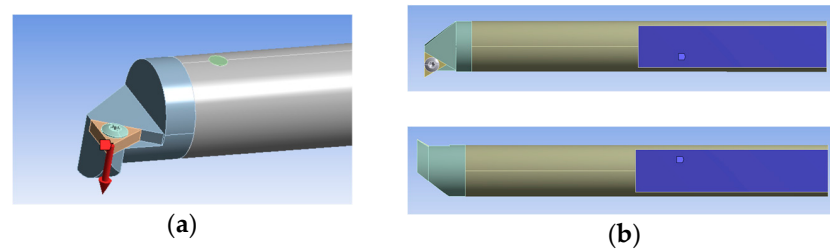


Figure 7. (a) Applied force and (b) fixed support.

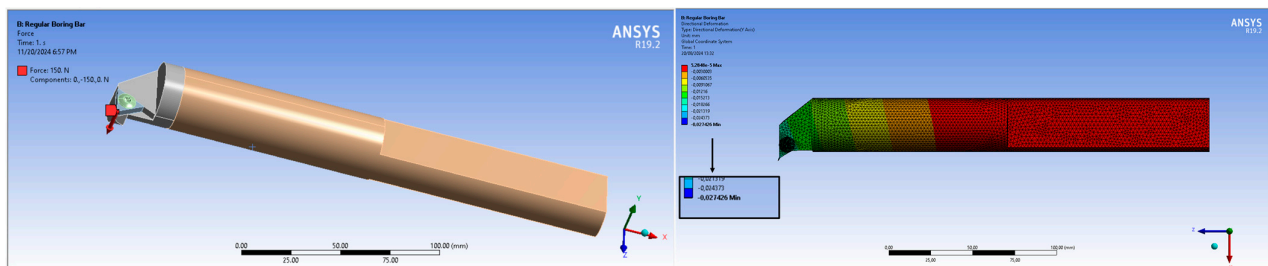


Figure 8. Static deformation simulation.

After that, to determine the equivalent mass of the boring bar at the insert edge, a modal analysis was carried out as shown in Figure 9. Fixed support was applied to the rear section of the boring bar, as shown in Figure 7b. The equivalent mass was determined using Equation (20).

$$M_1 = \frac{K_1}{\omega^2} \quad (20)$$

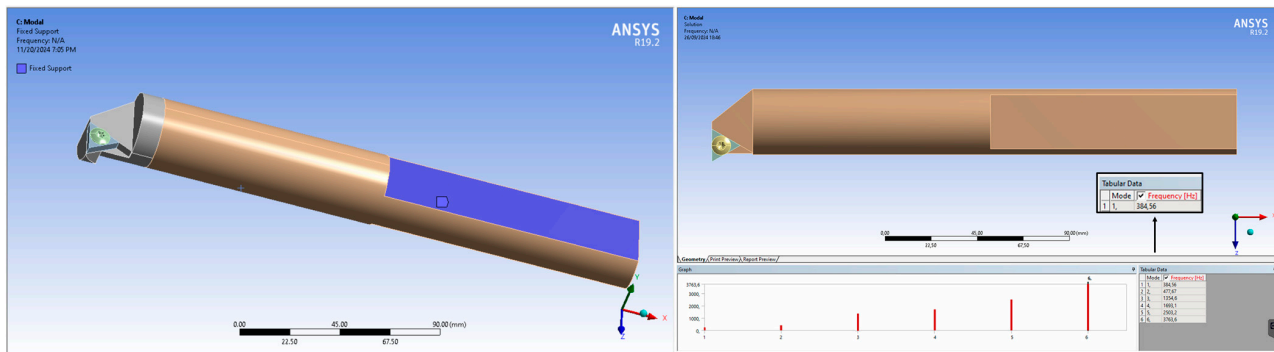


Figure 9. Natural frequency simulation.

The equivalent damping constant of the boring bar was obtained by using Equation (21), where the damping ratio (ζ_1) depended on the material workpieces. In this study, the value of the damping ratio (ζ_1) is 0.005. The same process was conducted to obtain the spherical MR-DVA parameter as shown in Table 2.

$$C_1 = 2\zeta_1 \sqrt{M_1 K_1} \quad (21)$$

Table 2. Parameters of boring bars and MR-DVAs.

Parameter	Stiffness (N/m)	Mass (kg)	Damping (Ns/m)	Natural Frequency (Hz)
Regular boring bar	0.55×10^7	0.942	22.76	384.56
Customized boring bar	0.42×10^7	0.701	17.16	389.44
DVA natural	0.39×10^7	0.00149	1.335	3571.2
DVA neoprene	0.49×10^7	0.00154	3.651	9426.6

The research involved simulating the related systems using a meshing method. Meshing involved using a finite element method as a geometric approximation of the system's original form. In this study, three simple meshing techniques were utilized, such as body sizing, tetrahedron meshing, and face meshing. Body sizing was employed for rigid components which were divided in the boring bar, insert, bolt, pin, and DVA to ensure consistent element distribution and computational efficiency. In the boring bar, mesh elements were set with a size of 2 mm. For the insert, pin, and DVA, a size of 1 mm was applied, while the bolt utilized a mesh size of 0.5 mm. Tetrahedron meshing was applied to cylindrical profiles such as rubber, spherical mass, and bolts as it provides better conformity to complex geometries and ensures accurate stress representation. Face meshing was implemented on the curved surfaces of the boring bar to align with the body sizing technique, enhancing the simulation's precision in these regions. The meshing process resulted in 292,348 nodes and 202,254 elements for the geometry of the main system with four DVAs. The mesh demonstrated a minimum element quality of 0.00026139 and a maximum quality of 0.99745, and the average quality of mesh using the orthogonal quality was 0.77603. The meshing results and quality are illustrated in Figure 10.

In this study, the vibration response of the boring bar was analyzed using ANSYS harmonic response as shown in Figure 11. The simulation employed the 3D geometry and engineering data of the system, incorporating the material properties of each component. The harmonic response analysis was conducted along the x -axis, y -axis, and z -axis directions of the boring bar. From this analysis, mode diagrams were generated showing the amplitude response of the main system in relation to the solution's frequency response deformation.

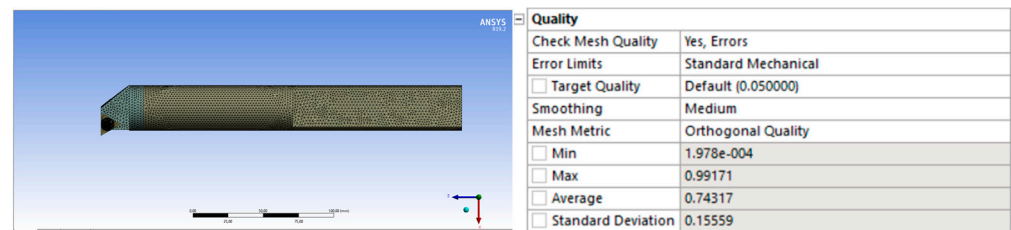


Figure 10. Mesh.

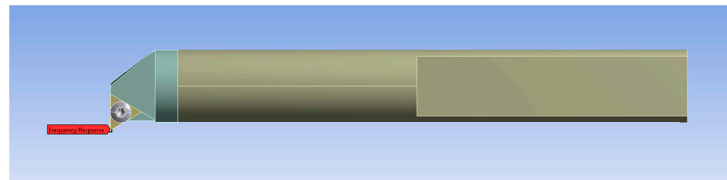


Figure 11. Harmonic response.

3. Results and Discussion

The simulation was conducted to evaluate the dynamic deformation response of the boring bar focusing on the edge of the insert along the x -axis (axial), y -axis (tangential), and z -axis (radial) directions. The simulation covered frequency data ranging from 2 to 1000 Hz for each boring bar geometry and processed using Ms. Excel software. The dynamic response was normalized by dividing it by the static response for each axis, represented as X_1/X_{st} , Y_1/Y_{st} , and Z_1/Z_{st} . The resulting dimensionless frequency responses were plotted using MATLAB software. Simulations were performed on the main system both before and after incorporating the spherical MR-DVA with variations in rubber combinations within the spherical MR-DVA.

3.1. Analysis of the Relationship Between Modal Analysis and Harmonic Response

In this study, simulations were conducted using ANSYS software starting with modal analysis to identify the natural frequencies of the system. Figure 12 shows the natural frequency results, specifically at 384 Hz for the first mode and 478 Hz for the second mode, that were incorporated as input for the harmonic response analysis. The system's vibration response was then evaluated in terms of deformation across a range of frequencies. The results, depicted as frequency versus amplitude graphs, demonstrated that peaks in the harmonic response occurred at the same frequencies as the natural frequencies identified in the modal analysis. The first peak of harmonic response appeared at 384 Hz and the second peak appeared at 478 Hz. This confirmed the correlation between the natural frequencies from modal analysis and the resonance observed in harmonic response analysis.

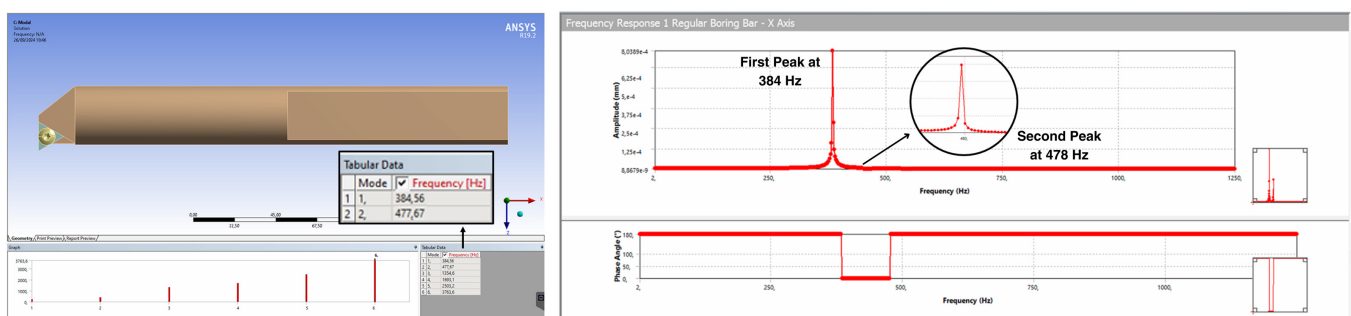


Figure 12. Modal analysis and harmonic response in ANSYS.

3.2. Analysis of Main System Vibration Response

Figure 13 depicts the bode diagram representing the vibration response of the boring bar geometries utilized in this study. The customized boring bar was designed with three pairs of spherical MR-DVAs. The stiffness constant of the spherical MR-DVA using natural rubber was 0.39×10^7 N/m, while that of neoprene rubber was 0.49×10^7 N/m. The vibration responses along the x -axis, y -axis, and z -axis are displayed. Within the frequency range of 2–1000 Hz, the regular boring bar exhibits its first peak amplitude ratio at 384 Hz. In comparison, the customized boring bar with spherical MR-DVAs demonstrates two distinct peak amplitude ratios within the same frequency range. The lowest peak amplitude ratio is observed in the customized boring bar configured with the NNA combination with a peak range spanning from 226 Hz to 718 Hz. This configuration effectively reduced vibrations due to the high stiffness of neoprene rubber, which dampened initial tool tip vibrations. The lower stiffness and damping properties of natural rubber then adjusted to the resonant frequency, further enhancing vibration absorption.

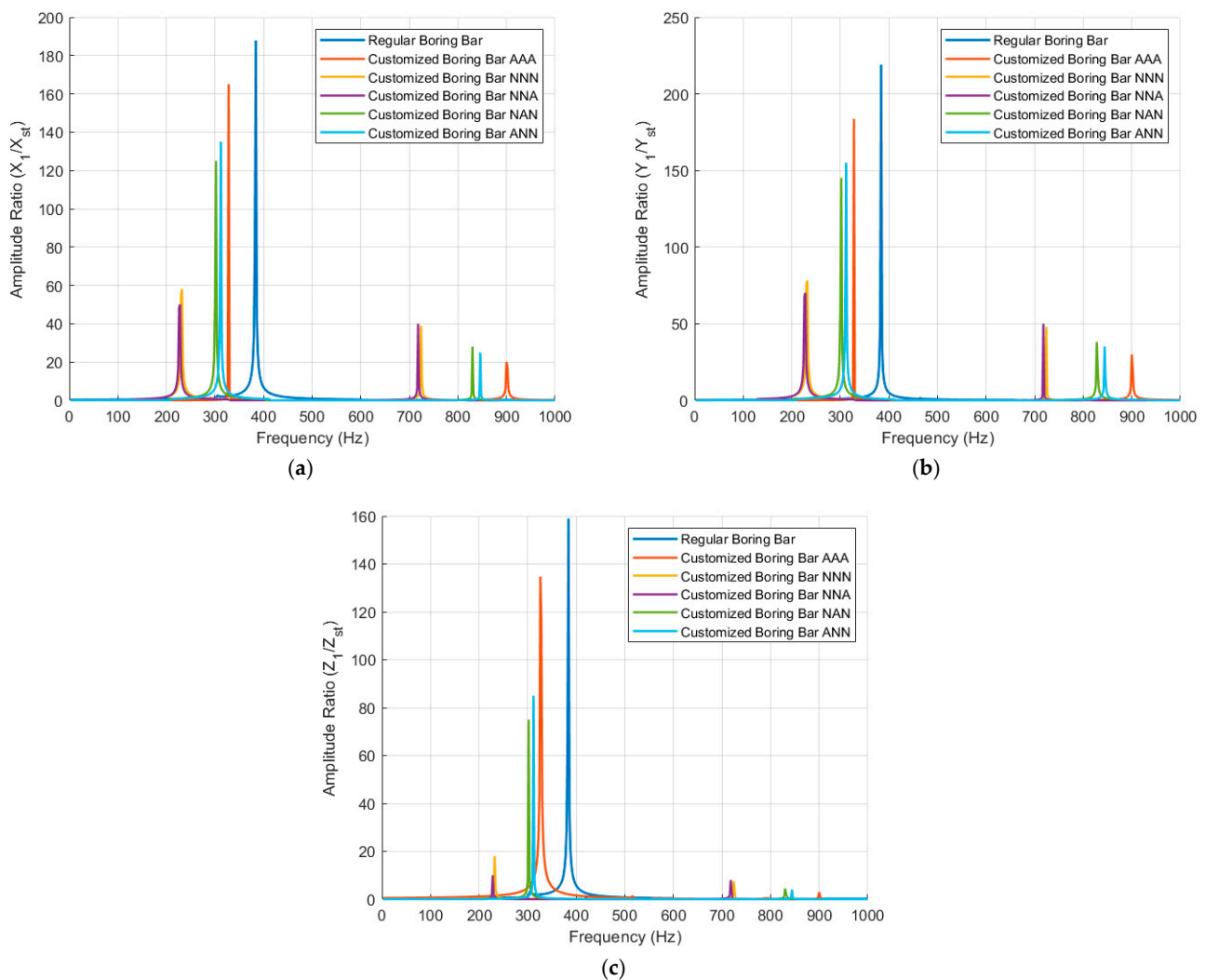


Figure 13. Bode diagram of boring bars: (a) x -axis; (b) y -axis; (c) z -axis.

Experiments with the customized boring bar with spherical MR-DVAs found that the first peak was observed leftward of the regular boring bar's peak and the second peak shifted rightward with a reduced amplitude ratio. This outcome indicated that the customized boring bar with spherical MR-DVAs successfully divided the single peak into two means, expanding the stable operating range during machining processes.

3.3. Vibration Reduction Analysis with Spherical MR-DVAs

Figure 14 illustrates the chatter phenomenon observed across variations in spindle speed listed in Table 1. Vibration reduction values were determined at each chatter frequency point and are presented in Figure 15. These reductions are expressed as percentages, comparing the regular boring bar to the customized boring bar equipped with spherical MR-DVAs. The configurations of NNN and AAA customized boring bars consistently showed the highest reduction percentages across all force axes. At a chatter frequency of 388 Hz, all customized boring bar configurations effectively reduced vibrations across all axes with an average reduction of 98%. Similarly, at 366 Hz, all configurations dampened vibrations across all axes with an average reduction of 89%. However, at 318 Hz, only the NNN and NNA configurations successfully dampened vibrations in all axes. The AAA configuration reduced vibrations in the x - and y -axes, the NAN configuration was effective only in the z -axis, and the ANN configuration did not provide damping in any axis.

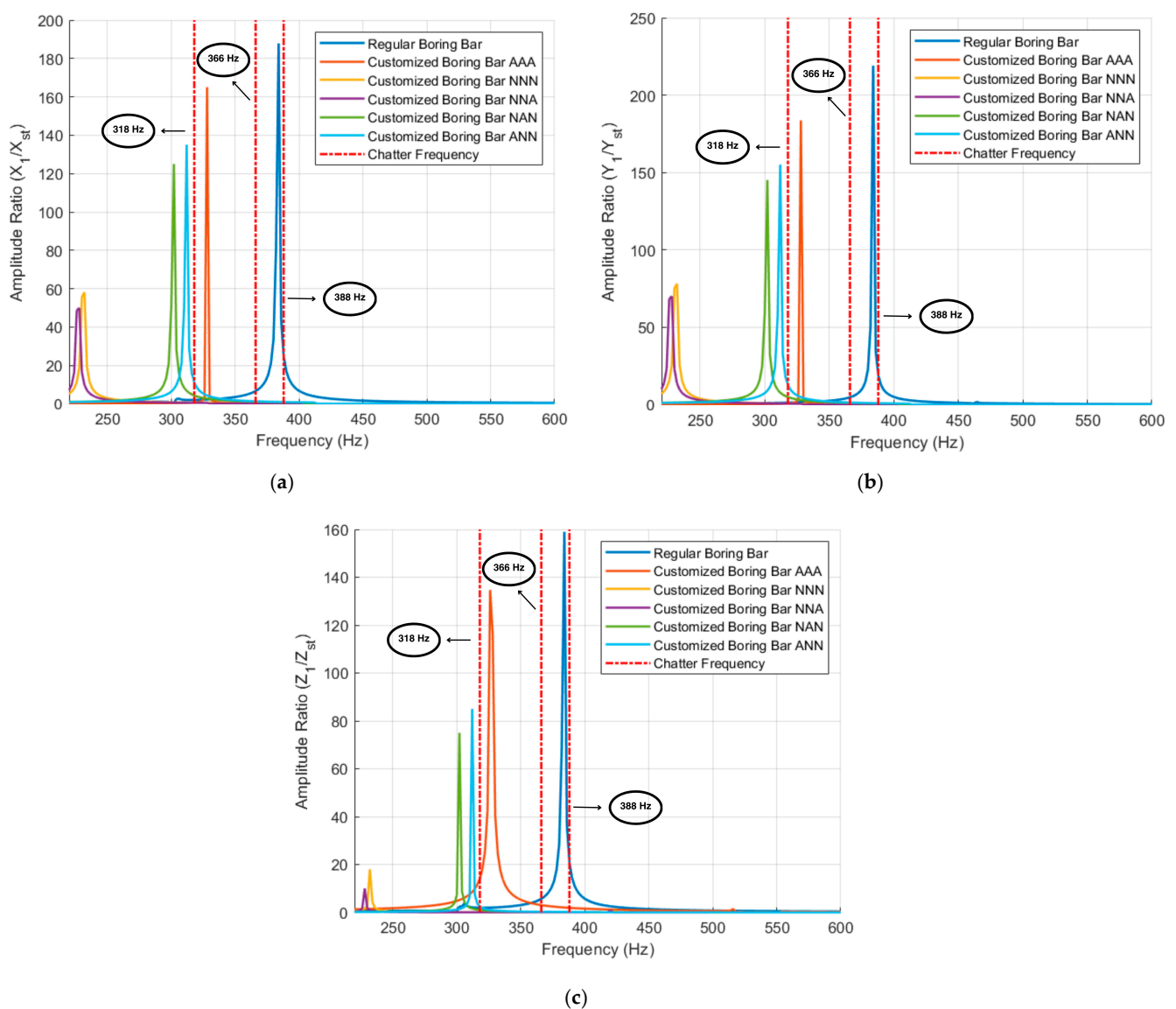


Figure 14. Chatter frequency. (a) 320 rpm; (b) 450 rpm; (c) 720 rpm.

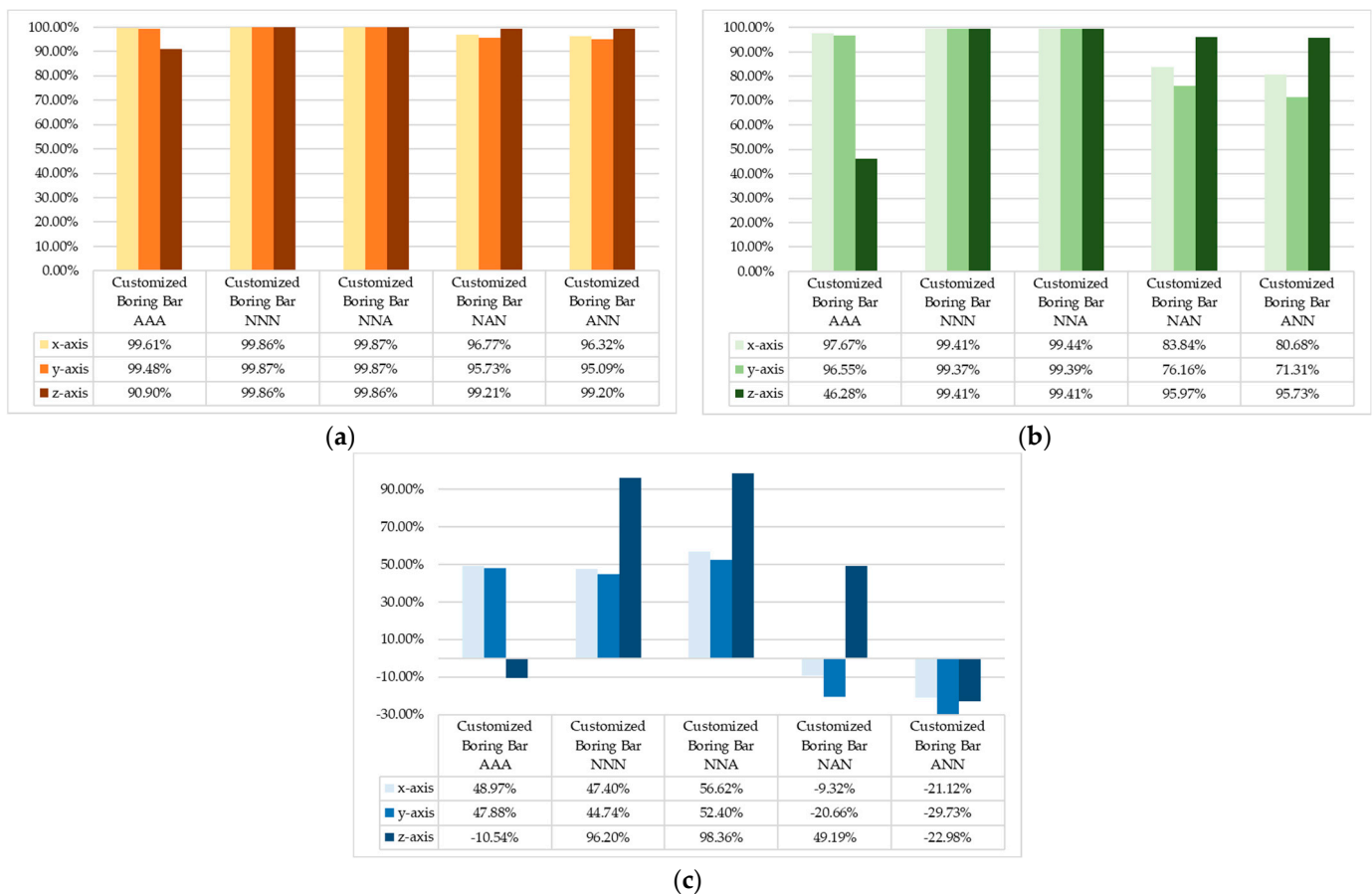


Figure 15. Vibration reduction in boring processes with spherical MR-DVAs at chatter frequency of (a) 388 Hz; (b) 366 Hz; and (c) 318 Hz.

The phenomenon depicted in Figure 15 was observed to indicate that neoprene rubber dampened vibrations more effectively at high chatter frequencies but was less effective at lower chatter frequencies. Optimal vibration control across various frequencies was achieved by combining neoprene and natural rubber, depending on whether neoprene or natural rubber dominated the damping configuration.

4. Conclusions

As a result of the simulation, integrated three pairs of spherical Mass–Rubber Dynamic Vibration Absorbers (MR-DVAs) into the boring bar effectively reduced vibration amplitudes and improved machining quality. The customized boring bar with the NNA configuration demonstrated the lowest amplitude values and consistently provided effective damping during the boring process across the utilized chatter frequency range. The spherical MR-DVAs shifted the natural frequencies, dividing the vibration peaks and extending the stable operating range during machining. The combination of neoprene and natural rubber provided optimal vibration control, with neoprene excelling at higher chatter frequencies and natural rubber contributing at lower frequencies.

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Conflicts of Interest: The authors declare no conflicts of interest.

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