ISME

The Iranian Journal of Mechanical Engineering Transactions of ISME
Journal homepage: https://jmee.isme.ir/
Vol. 26, No. 1, 2025, pp. 31-43

Research Paper

DOI: https://doi.org/10.30506/jmee.2024.2027496.1345

The Optimum Design of an Anti-vibration Boring Bar by Improving Dynamic Characteristics

Mahdi Hosseinpour*
M.Sc. Student

Abbas Rahi†
Associate Professor

Dynamic absorbers have many applications in reducing the undesirable vibration of the systems and can play a key role in suppressing the vibration but their performance is limited to a specific range of frequencies so they need to be optimized based on operation conditions. Due to the low stiffness of the long cantilever boring bar, vibration often occurs during the boring process. Vibration suppression by dynamic absorbers permits higher productivity and a better surface finish. To improve the performance of the boring bar operation, a variable stiffness dynamic absorber is added inside the boring bar to control and reduce the vibration. A boring bar equipped with a dynamic absorber can be tuned with different values of absorber length, diameter, stiffness, and installation position. This study modeled a boring bar with a long overhang equipped with a dynamic absorber and investigated the effect of each parameter on the stable cutting range of the boring process. By defining the stable cutting range of the boring bar as an objective function and the constraints of each parameter like the length, thickness, and chatter frequency, the optimum values of each parameter are determined by developing and using generated numerical analysis.

Keywords: Boring bar, Dynamic absorber, Transverse vibration, Natural frequency, Stiffness

1 Introduction

Boring bars with high length-to-diameter ratios are widely used in industry. However, when the boring bar has a large length-to-diameter ratio, vibration often occurs, which leads to a negative effect on surface quality, tool life, and processing performance so the new methods needs to be used to control the vibration of the boring bar and improve vibration condition of the tool [1], [2].

*M.Sc. Student, Faculty of Mechanical and Energy Engineering, Shahid Beheshti University, Tehran, Iran, ma.hoseinpour@mail.sbu.ac.ir

†Corresponding author, Associate Professor, Faculty of Mechanical and Energy Engineering, Shahid Beheshti University, Tehran, Iran, a_rahi@sbu.ac.ir

Received: 2024/05/15, Revised: 2024/08/09, Accepted: 2024/09/01

The vibration of the machining tool is an interesting topic for researchers and some researches are written on the vibration control of the machining tools [3]. The scholars mainly focused on the mechanism structures, boring materials, and the theory of passive and active methods. Jingmin et al. [4] used a conic boring bar with layered composite material for improving the stiffness of the boring operation and showed that by decreasing the layer angle and increasing the conic angle the natural frequency of the tool increases. Soyama et al. [5] used a carbide rod in the inner hole of the body for improving the boring bar stiffness and reached higher length-to-diameter ratios for a specific boring operation. Mihic [6] proposed a novel boring bar with a peripheral elastic layer for increasing damping and achieved higher damping in the mechanism and a more stable cutting operation. Sayuti et al. [7] used coolant to decrease the friction between the tool and workpiece to generate smaller forces during boring operation. Xia et al. [8] proposed a multi-layer boring bar to increase the natural frequency and damping ratio of the tool to increase the stability of the operation. Daghini et al. [9] proposed a novel configuration for a boring bar holder to guide the vibration energy through the dampers and achieved 10 times more damping for the configuration and higher productivity. Garcia et al. [10] used a dynamic absorber to control the vibration of a system and showed that by increasing the mass of the absorber the optimum value of the natural frequency ratio decreases. Tunc et al. [11] investigated the effect of process damping on the boring operation and showed that as the cutting speed decreases, the process damping increases. Beregovenko [12] investigated the nitriding effect on increasing the damping ratio of steel ck 45 and showed that nitriding can increase the damping ratio of steel up to 6 times. Sortino et al. [13] investigated the effect of cutting parameters on the stability lobe diagram and showed that tool length to diameter ratio is the most important parameter affecting the stability lobe diagram. Rahi et al. [14] investigated the parameters of a long overhang boring bar with an outer dynamic absorber and showed the natural frequency of the model is the highest with equal stiffness distribution between dynamic absorber springs. Filho [15] used a suspended mass on two elastic materials with an adjustable mechanism to change the compression force of the system. He used this passive dynamic absorber to control the vibration of the boring bar in a specific range of frequencies. Li et al. [16] investigated the vibration characteristic of a boring bar with a variable stiffness dynamic absorber and measured the effect of axial compression on the radial stiffness of the dynamic absorber spring. Caslstedt [17] used a dynamic absorber consisting of many heavy rings wasting vibration energy based on friction to reduce the vibration of the boring bar. Abele et al. [18] proposed the active method to control and reduce the vibration of a long overhang boring bar and used an inertia mass actuator to actuate the boring bar according to the excitation forces during the boring operation to increase stability.

As a summary, the literature review can be defined in a few steps. The first step includes a few methods to increase boring bar static stiffness like studies done with Jingmin around improving boring bar vibration condition. The second step includes methods focused on damping vibration energy by adding damping layers on the boring bar body like studies done with Peter and the third step uses some metallurgical methods to increase the mechanical properties of the boring bar steel body to result in better cutting condition like efforts made by cuplon. The last step includes methods that use dynamic absorbers to dampen the vibrating energy of the boring bar like studies done with Ebel. By considering the time of studies it is clear that new emerging articles are written around using dynamic absorbers to dampen vibrating energy and all of them modeled the dynamic absorber as a single degree of freedom body.

So,This study investigated the optimum design of a boring bar equipped with an inner two-degree-freedom dynamic absorber. For this purpose, firstly the governing equation of the analytical model is extracted by the finite element method and the effect of each parameter on the boring bar is studied then by defining optimization variables, limitations, and objective functions, the optimum values of the modeled boring bar with inner dynamic absorber are extracted by developing and using generated numerical analysis.

2 Dynamic model of boring bar and governing equations

Normally the boring operation is performed with a simple structure consists of a rod and an insert at the end and this structure normally is modeled as a long cantilever beam and a pointed force at the end. Because of the long overhand of boring tools, the stiffness of the tool during the operation is low and the tool is susceptible to vibrating during the cutting operation. This vibration excites the first mode of vibration of the tools and needs to be controlled. Moreover, the operating condition of the boring bar does not allow for improving the stiffness of the tool by increasing the tool diameter, so embedding a dynamic absorber in the body would be a good idea.

The view of the boring bar with an embedded dynamic absorber and tool holder is shown in Figures (1) and (2). the dynamic absorber consists of a cylindrical mass and two rubber bushes are placed inside the boring bar. It should be noted that axial compression could be adjusted for the used rubber bush to reach the required radial stiffness and the rubber bush used in the proposed model has specifications of an outer diameter of 24.2 mm, an inner diameter of 11.5, and a thickness of 7.1.

When the exciting force occurs, the vibration energy is transmitted from the tooltip to the tool shank and embedded dynamic absorber. As shown in Figure (2), the boring bar is subjected to three forces during the cutting process, namely axial force, tangential force, and radial force. Also, The structure of the used dynamic absorber is shown in Figure (3).

The rigidity of the bar is much higher along the feed or axial direction than in the tangential and radial bending directions. Also, the boring bar exhibits higher stiffness in torsion than in bending. Therefore, the bending vibrations caused by the tangential force and radial force should be considered in the analysis [19]. So the exciting force is expressed as equation (1):

$$\overrightarrow{F_0} = \overrightarrow{F_c} + \overrightarrow{F_p} \tag{1}$$

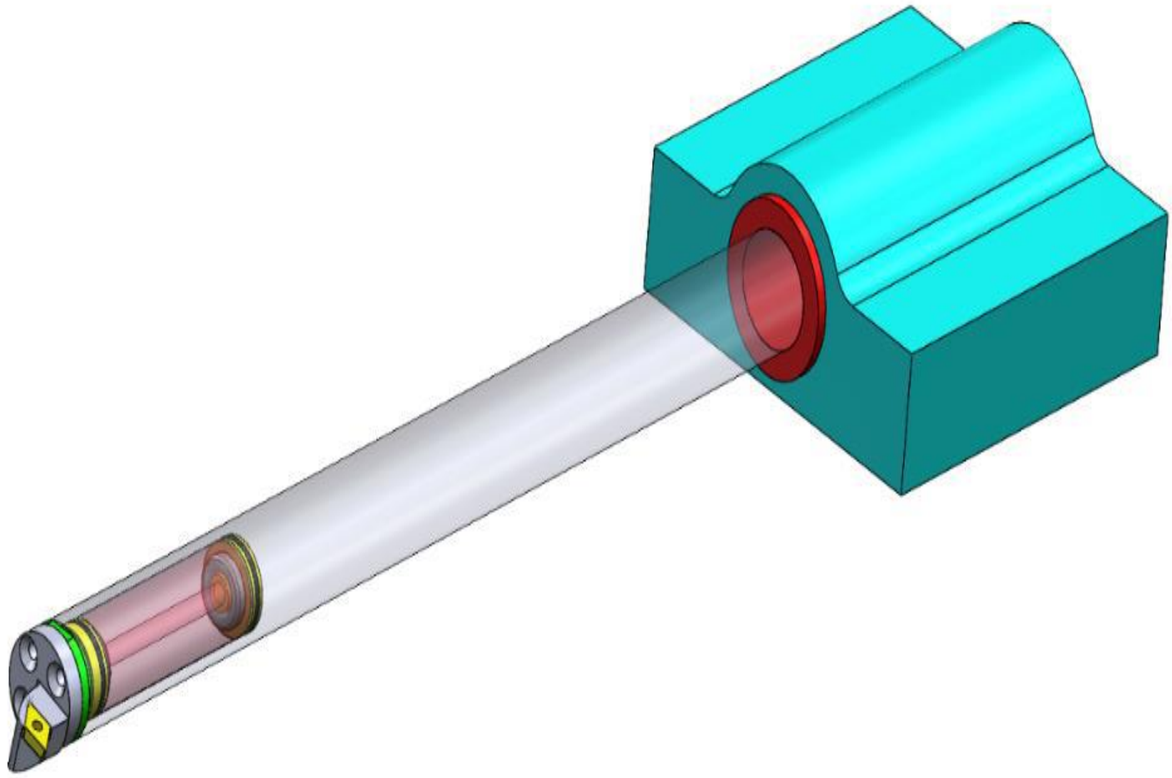

Figure 1 The boring bar with embedded dynamic absorber

To simplify the calculation, the excitation force is considered a sine wave.

$$\overrightarrow{F_0} = F\sin(\omega t) \tag{2}$$

And F and ω are the amplitude and frequency of the excitation forces, respectively.
The analytical model of the modeled boring bar with an embedded dynamic absorber is shown in Figure (4). The model is discretized by 5 nodes and 5 elements. The first, second and fourth elements are considered beams and the third and fifth elements are regarded as truss elements.

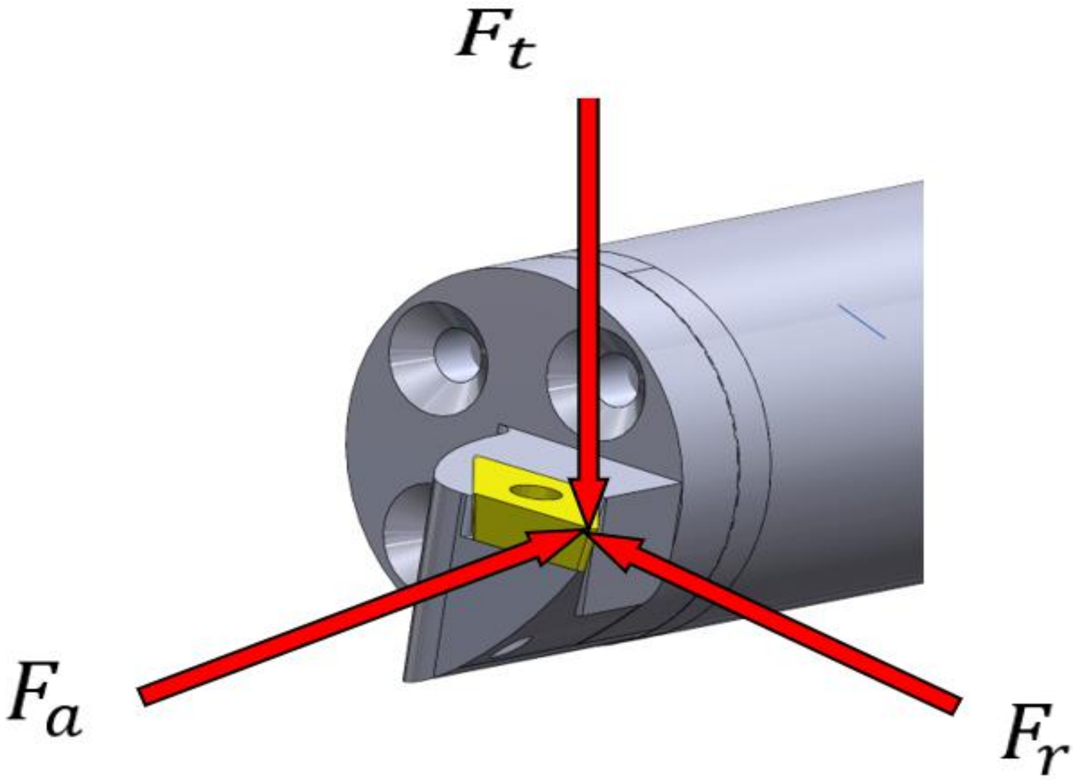

Figure 2 Tangential, radial, and axial forces on cutting insert

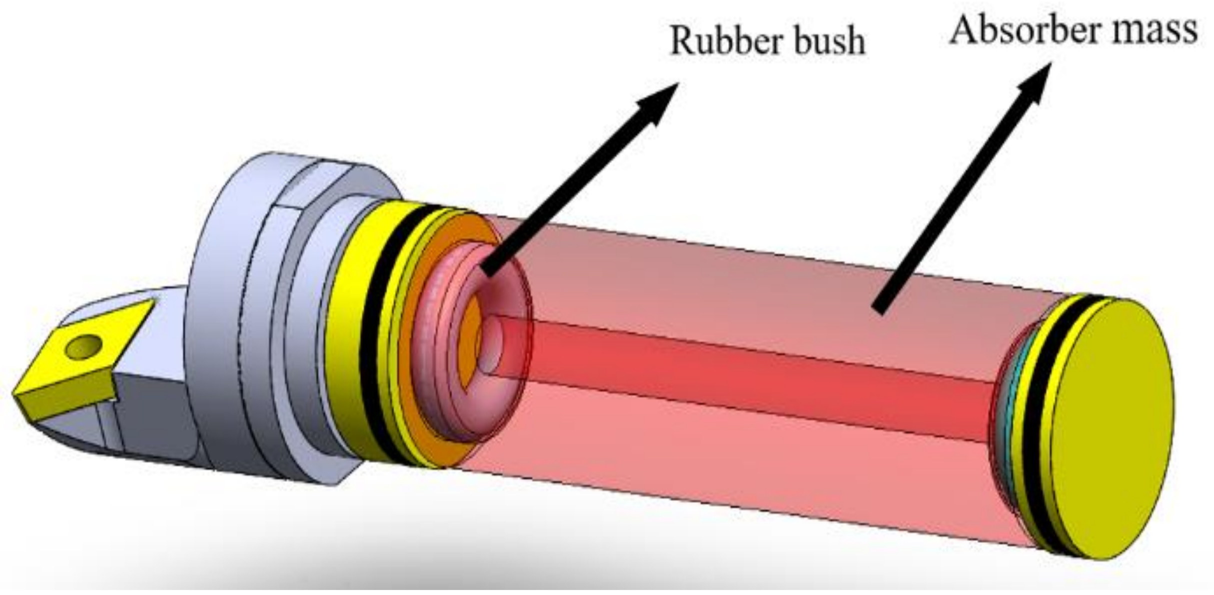

Figure 3 The structure of the used dynamic absorber

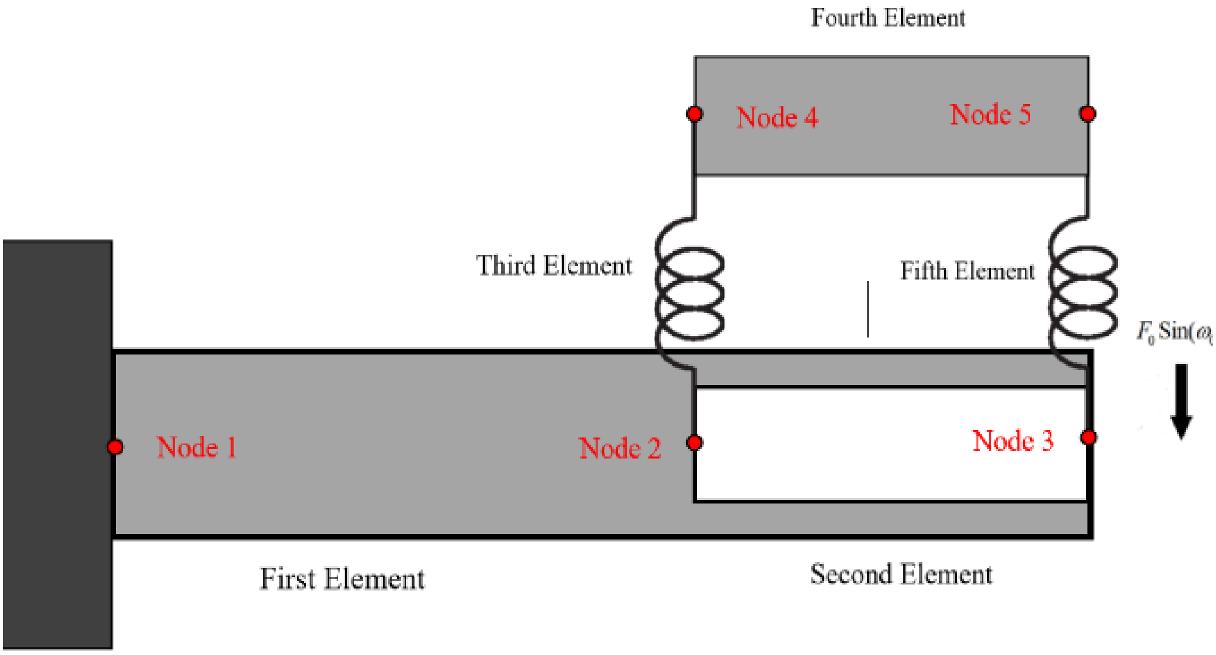

Figure 4 The analytical model of the boring bar with an embedded dynamic absorber

The equivalent mass and stiffness for a beam element with a hollow are defined in equation (3):

$$\boldsymbol{k}_{e,i} = \frac{EI_i}{l_i^3(1+\phi_i)}\begin{bmatrix} 12 & 6l_i & -12 & 6l_i \\ 6l_i & (4+\phi_i)l_i^2 & -6l_i & (2-\phi_i)l_i^2 \\ -12 & -6l_i & 12 & -6l_i \\ 6l_i & (2-\phi_i)l_i^2 & -6l_i & (4+\phi_i)l_i^2 \end{bmatrix} \quad (3)$$

$$m_{A,i} = \frac{\rho A_i l_i}{210(1+\phi_i)^2} \times \begin{bmatrix} (70\phi_i^2+147\phi_i+78) & \frac{l_i}{4}(35\phi_i^2+77\phi_i+44) & (35\phi_i^2+63\phi_i+27) & \frac{-l_i}{4}(35\phi_i^2+63\phi_i+26) \\ \frac{l_i}{4}(35\phi_i^2+77\phi_i+44) & \frac{l_i^2}{4}(7\phi_i^2+14\phi_i+8) & \frac{l_i}{4}(35\phi_i^2+63\phi_i+26) & \frac{-l_i^2}{4}(7\phi_i^2+14\phi_i+6) \\ (35\phi_i^2+63\phi_i+27) & \frac{l_i}{4}(35\phi_i^2+63\phi_i+26) & (70\phi_i^2+147\phi_i+78) & \frac{-l_i}{4}(35\phi_i^2+77\phi_i+44) \\ \frac{-l_i}{4}(35\phi_i^2+63\phi_i+26) & \frac{-l_i^2}{4}(7\phi_i^2+14\phi_i+6) & \frac{-l_i}{4}(35\phi_i^2+77\phi_i+44) & \frac{l_i^2}{4}(7\phi_i^2+14\phi_i+8) \end{bmatrix} \quad (4)$$

where ϕ_i is the shear correction term defined in equation (5):

$$\phi_i = \frac{12EI_i}{k_{si}A_iGl_i^2} \quad (5)$$

and k_{si} is equivalent with:

$$k_{si} = \frac{6(a_i^2+b_i^2)^2(1+\nu^2)}{7a_i^4+34a_i^2b_i^2+7b_i^4+\nu(12a_i^4+48a_i^2b_i^2+12b_i^4)+\nu^2(4a_i^2+16a_i^2b_i^2+4b_i^4)} \quad (6)$$

where a_i and b_i as shown in Figure (5) are the inner and outer diameters of the beam cross-section with an inner hollow.

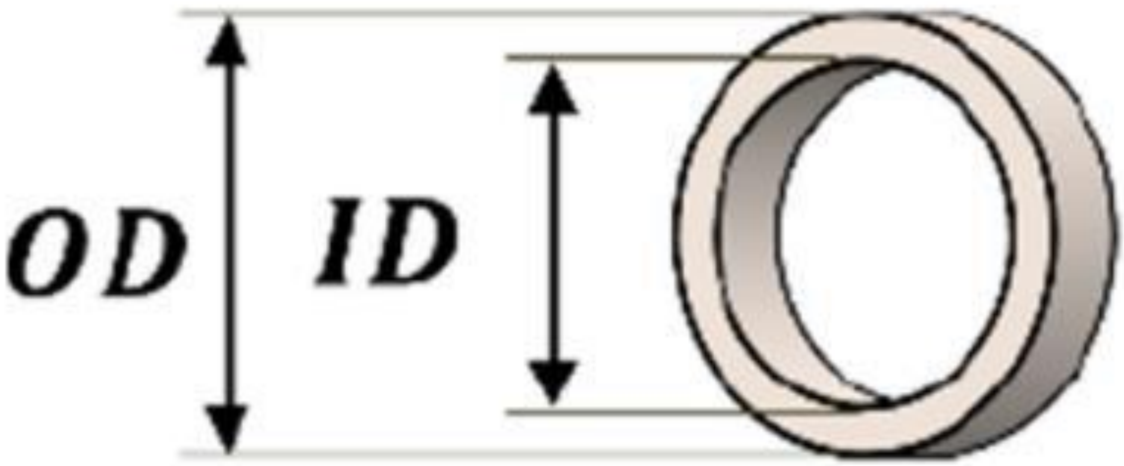

Figure 5 The cross-section of a beam with inner and outer diameter

The mass and stiffness matrix for truss elements is defined as:

$$[K_i] = \frac{EI_i}{l_i^3}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} \tag{7}$$

$$[M_i] = \frac{\rho AL}{6}\begin{bmatrix} 2 & 1 \\ 1 & 2 \end{bmatrix} \tag{8}$$

By considering each element's connection with other elements and forming the whole mass and stiffness matrix is extracted as:

$$[M_i] = \begin{bmatrix} M_{ai} & \dots & M_{bi} \\ \dots & \dots & \dots \\ M_{bi}^T & \dots & M_{ci} \end{bmatrix} \tag{9}$$

$$[K_i] = \begin{bmatrix} K_{ai} & \dots & K_{bi} \\ \dots & \dots & \dots \\ K_{bi}^T & \dots & K_{ci} \end{bmatrix} \tag{10}$$

$$[M] = \begin{bmatrix} M_{a1} & M_{b1} & \square & \square & \square \\ M_{b1}^T & M_{c1}+M_{a2} & M_{b2} & \square & \square \\ \square & M_{b2}^T & M_{c2}+M_{a3} & M_{b3} & \square \\ \square & \square & M_{b3}^T & M_{c3}+M_{a4}M_{b4} & \\ \square & \square & \square & M_{b4}^T & M_{c4} \end{bmatrix} \tag{11}$$

$$[K] = \begin{bmatrix} K_{a1} & K_{b1} & \square & \square & \square \\ K_{b1}^T & K_{c1}+K_{a2} & K_{b2} & \square & \square \\ \square & K_{b2}^T & K_{c2}+K_{a3} & K_{b3} & \square \\ \square & \square & K_{b3}^T & K_{c3}+K_{a4}K_{b4} & \\ \square & \square & \square & K_{b4}^T & K_{c4} \end{bmatrix} \tag{12}$$

By substituting equation (11) and equation (12) in the momentum equation, the equation is found as:

$$[M]\{\ddot{x}\} + [K]\{x\} = \{f\} \tag{13}$$

where $\{x\}$ and $\{f\}$ are displacement and force matrixes. By extracting the displacement matrix, the implemented matrix is found as equation (14):

$$\{x\} = (-\omega^2[M] + [K])^{-1}\,\{f\} \tag{14}$$

By considering the system boundary conditions, equation (14) can be solved for different exciting frequencies.

3 Analysis of the vibration characteristics of the dynamic absorber

The modeled boring bar has the specification shown in Table (1). The vibration behavior of the shown boring bar with an inner dynamic absorber is under the effect of key parameters. The parameters affecting the vibration of the modeled boring bar are the hole and absorber diameter, hole and absorber length, the stiffness of the rubber bushes, and the location of the dynamic absorber installation. The behavior of each parameter will be investigated.

Table 1 Specification of the modeled boring bar

Tool name	Shank diameter (mm)	Total overhang (mm)	Insert name
S32T SDUCR 11	32	384	DCMT 11

Table 2 The natural frequencies of the boring bar before and after adding the dynamic absorber

The mode number	Frequency with dynamic absorber (rad/sec)	Frequency without dynamic absorber (rad/sec)
1	3410	1395
2	15100	7015
3	22044	16710

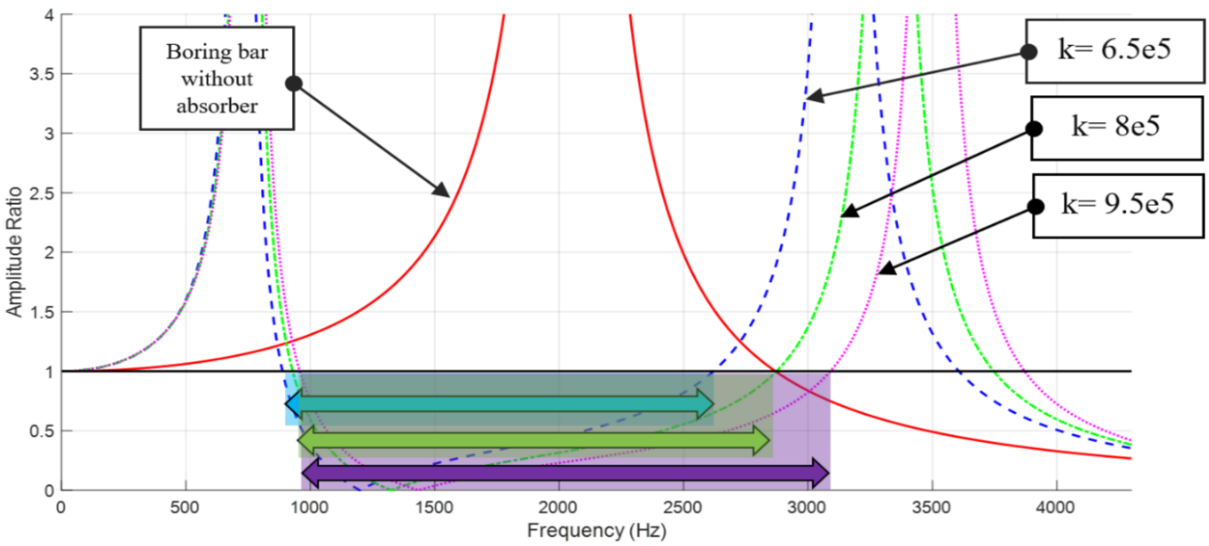

Figure 6 Investigation of the stiffness value on the stable cutting range frequency

By extracting the natural frequency of the tool, before and after adding the dynamic absorber, the natural frequencies of the tool can be expressed as the values shown in Table 2). As it is clear, the dynamic absorber has a significant effect on the first natural frequency of the boring bar and increased it from 1395 rad/sec to 3410 rad/sec so the tool will start to vibrate in higher frequencies.

The investigation of the effect of the dynamic absorber stiffness on the vibration behavior of the tool is shown in

Figure (6). In comparison to the boring bar without the dynamic absorber, the preferred operating frequencies are colored in the bellow figure and it is preferred to be increased as much as possible because the amplitude ratio is lower than the conventional boring bar without the dynamic absorber.

It should be noted that the high-frequency values have also a lower amplitude ratio but it should be pointed out that reaching high-frequency values is almost impossible due to the low frequency of the boring operation. Moreover, for reaching the high-frequency values it is needed to surpass the natural frequency of the tool which decreases the surface quality and tool life.

The effect of the dynamic absorber position on the amplitude ratio of the tool based on the second spring location is shown in Figure (7). As it is clear by changing the position of the second spring, the amplitude ratio of the tooltip decreases in different applied forces.

It should be noted as the dynamic absorber gets close to the tooltip, the amplitude ratio of the tool decreases because the dynamic absorber gets close to the first mode's high amplitude displacements.

As shown in the analytical model, the dynamic absorber is located inside the inner hole with a specific diameter and length. It should be pointed out that by changing the hole diameter and length, the natural frequency of the tool changes as shown in Figure (8).

As it is shown in Figure (8) by increasing the hole length, the natural frequency of the tool without the absorber increases up to a specific value and then decreases. The shown behavior is caused because at the beginning, decreasing the mass is much more than decreasing the natural frequency and after a specific value, the decrease of the stiffness is much more than decreasing the mass value.

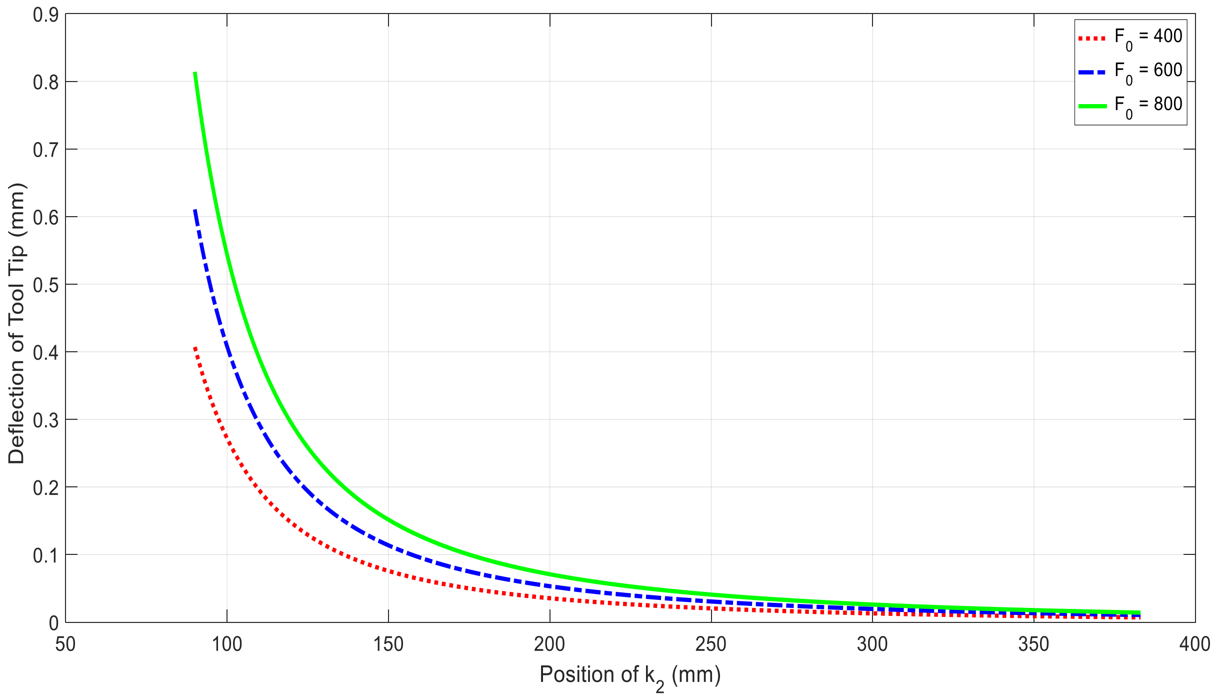

Figure 7 The position of the boring bar installation based on the second spring location

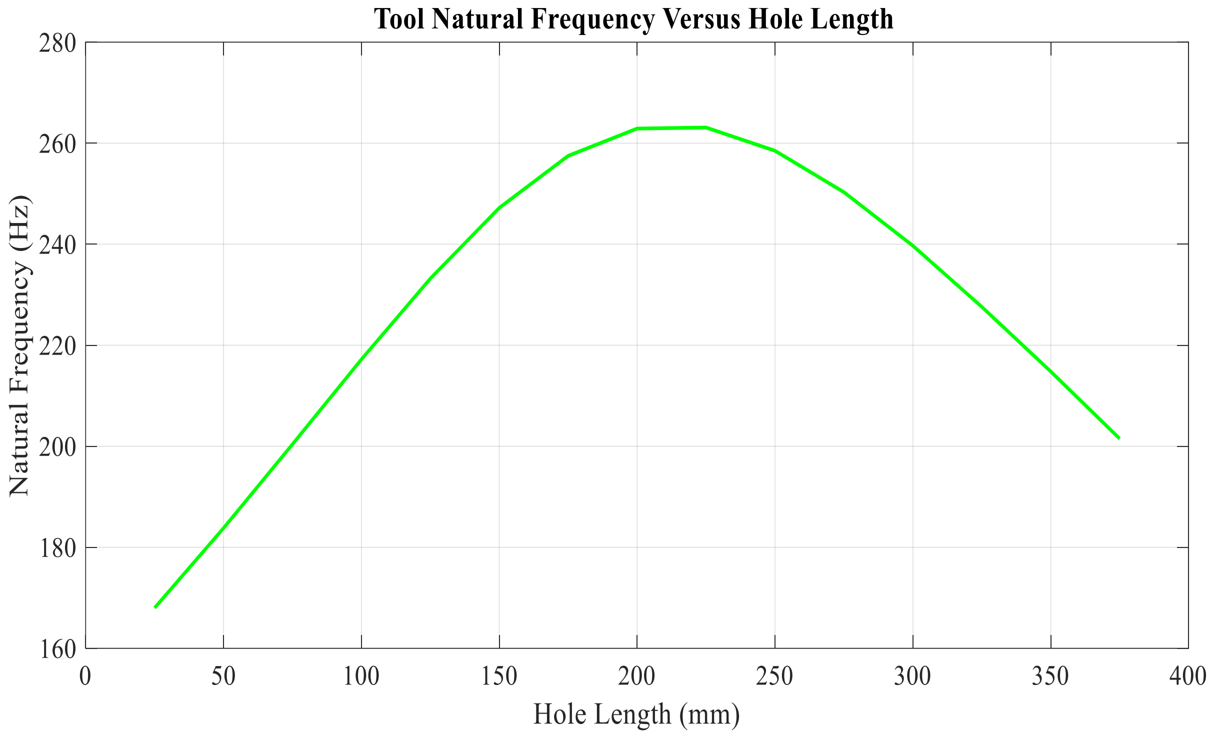

Figure 8 Changing the natural frequency of the tool by increasing the hole length

According to Figure (4) and Figure (8) as the absorber length increases, from one side the exerted point of the dynamic absorber gets close to the beginning of the beam which decreases the effectiveness of the dynamic absorber from another side the absorber mass increases which increases the effectiveness of the dynamic absorber significantly. Moreover by increasing the absorber length, the natural frequency of the tool increases and then decreases respectively. So the absorber length needs to be optimized in trade-off with other parameters.

4 Optimizing the parameters

The effect of each parameter on the tool vibrating behavior was investigated in the previous section. For designing an optimized boring bar with the dynamic absorber, the parameters like the absorber position, absorber length, stiffness, and frequency ratio should be optimized.
The analysis of the dynamic absorber characteristics showed as the absorber gets close to the tooltip, the vibration of the tool improves significantly. Moreover, increasing the mass improves the function of the dynamic absorber. It also showed that increasing the absorber length is not always a good idea and the absorber length should be optimized. It should be noted that the absorber length and the hole length are always assumed to be equal and the hole length can not exceed the length-to-diameter ratio of more than 4 because of the production limitation in the industry.
The ratio of dynamic absorber natural frequency to the tool natural frequency ratio is called natural frequency ratio. It should be noted that the rubber bush stiffness and the natural frequency ratio are related parameters and the selection of one of them determines the value of the other parameter. So, the natural frequency ratio of the dynamic absorber to the tool should be selected.
Li[15] showed that the radial stiffness of the modeled boring bar is under the effect of the axial load of the rubber bush and calculated the relationship between axial load and the resultant radial stiffness of the used rubber bush. He showed that the maximum reachable radial stiffness for the proposed rubber bush is 100 kilo Newton per meter so the required radial stiffness of the optimized dynamic absorber should not exceed the maximum value.

It is shown that the best installation position for the dynamic absorber is the closest place to the tooltip so in the following optimization the absorber is fixed at the closest distance to the tooltip. As the diameter of the hole increases the performance improves but the body gets vulnerable to failure so by considering the fatigue limit of the steel 4140 with oil quenched and tempered at 650 degrees, the maximum allowable hole diameter is calculated as 27 mm so the maximum hole length would be 108 mm. moreover the clearance between the hole and the dynamic absorber is assumed to be 1 mm.
The objective function is assumed to be the frequency range of the stable cutting range in the frequency response diagram and it is tried to maximize this frequency range.
By calculating the stable frequency range of the tool for different absorber lengths and natural frequency ratios, the surface is shown in Figure (9). As shown in the bellow surface, the objective function does have different maximum values based on the different absorber lengths and natural frequency rations.
The vibration of the boring operation consists of three different vibration categories. The first category is the free vibration of the tool due to the jamming of the chip during the operation. The second category is forced vibration which comes from the spindle speed and corresponding frequency and the third category is the self-excited frequency. The first and the second categories are controlled and reduced easily and the third category is the most important group with higher complexity. Because in the boring operation, the self-excited phenomena are the most effective drawback for increasing productivity and the main reason behind decreasing the surface quality, this vibration is considered as the main problem in the boring operation.
Because the self-excited vibration is always exciting the first natural frequency of the tool, the dynamic absorber should be focused on controlling and reducing this frequency range so the natural frequency ratio of the proposed boring bar with dynamic absorber should be equal to 1. The effect of the self-excited vibration of the tool on the surface quality is shown in Figure (10). The resultant objective function for the natural frequency ratio of 1 is shown in Figure (11). It is clear that for the hole range up to 108 mm, the maximum value belongs to the length of 108 mm and the corresponding stiffness is 800 kilo Newton per meter which is in the allowable range.
The specification of the optimized point of the diagram is shown inTable (3).

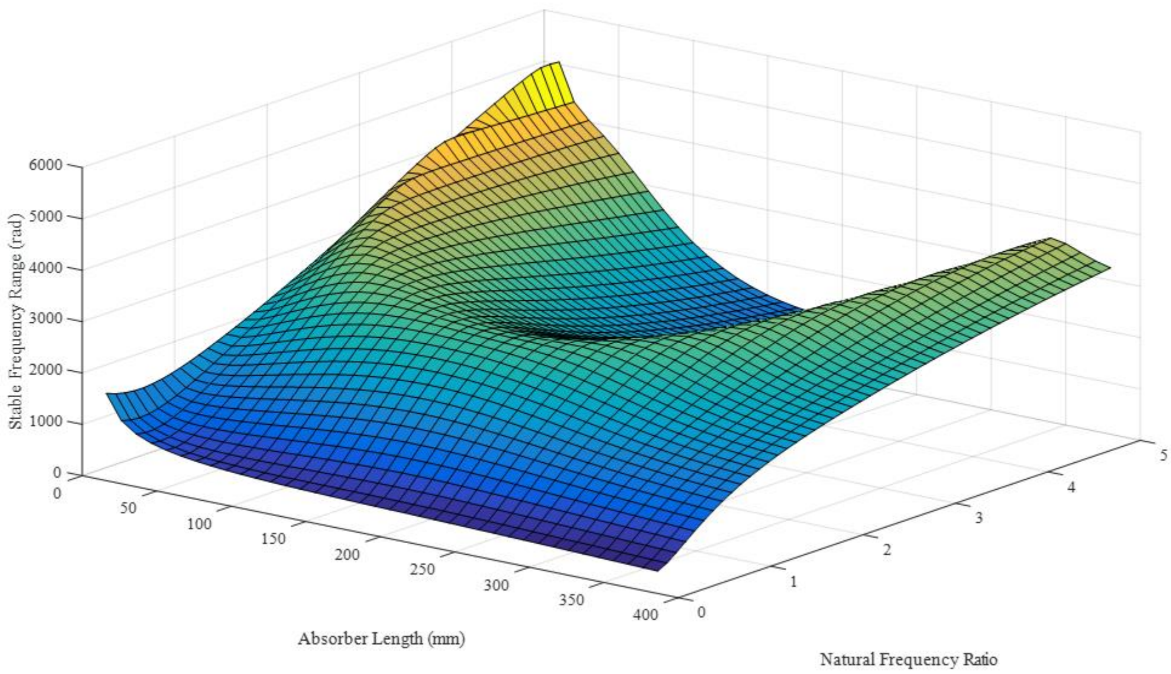

Figure 9 The stable frequency range of the proposed boring bar

Figure 10 The surface quality of the workpiece with self-excited vibration (**a**) and without self-excited vibration (**b**)

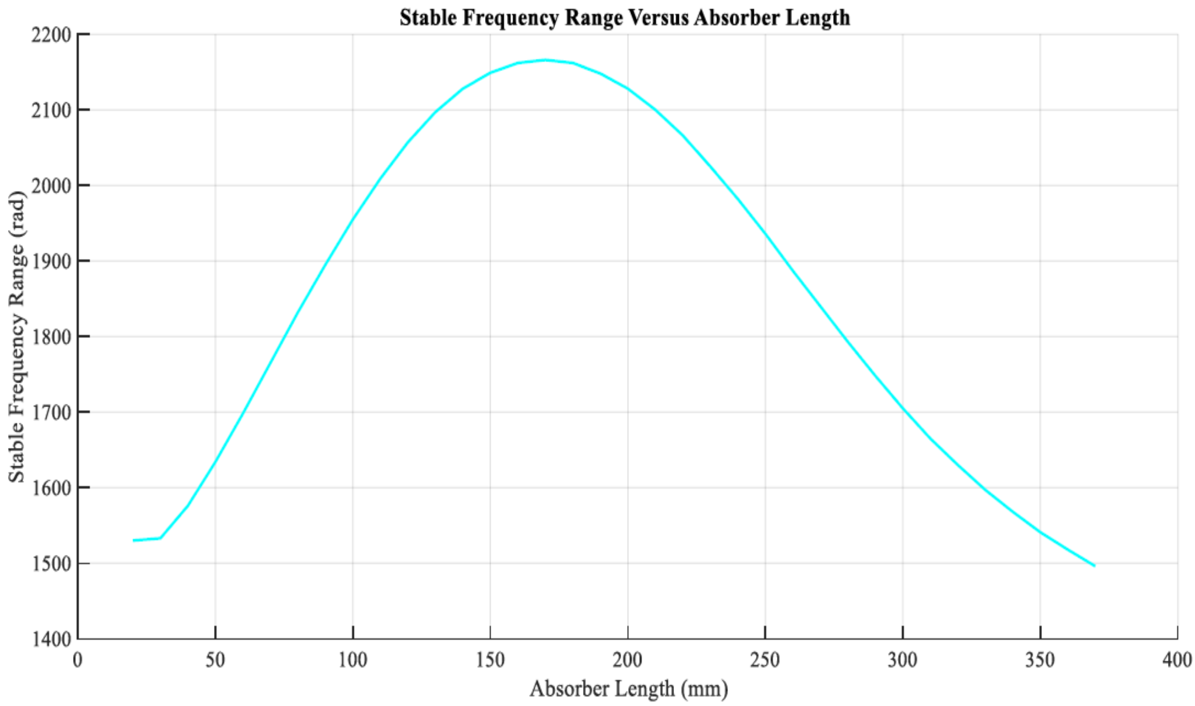

Figure 11 Stable cutting range versus different absorber lengths in the natural frequency ratio 1

Table 3 The values of optimized boring bar

Natural frequency ratio	Radial stiffness (N/m)	Hole length (mm)	Absorber length (mm)	Hole diameter (mm)	Absorber diameter (mm)	Absorber length (mm)
1	800000	108	108	27	25	175

It should be considered that as the absorber length increases, from one side the mass of the absorber increases which is a positive point, and from another side by increasing the length of the absorber one of the forces gets close to the beginning of the tool and loses effectiveness on controlling and reducing the vibration which is a negative item. Up until 175 mm in length, the final behavior is under the effect of increasing the mass, and from 175 mm up until 400 the negative item is the most dominant item.

5 Conclusion

The vibration of a boring bar with a long overhang equipped with an inner dynamic absorber investigated. The analytical model of the boring bar with an attached mass established and the effect of each parameter on the tool vibration behaviour studied. Changing the natural frequency of the tool before and after adding the dynamic absorber compared and the results show that the boring bar with a dynamic absorber has better stiffness and higher natural frequency so the tool with a dynamic absorber starts to vibrate at a higher frequency. In addition, changing the stiffness of the dynamic absorber changes greatly the stable cutting range of the tool equipped with a dynamic absorber. The effect of the installation of the dynamic absorber on the vibration of the tooltip investigated and shows that as the dynamic absorber gets close to the tooltip, the performance improves significantly. The effect of the hole on the natural frequency of the tool studied and shows that the natural frequency of the tool increases and then decreases drastically. After investigating the effect of each parameter on the tool vibration and determining the values of some parameters, the stable cutting range of the tool according to the tool natural frequency ratio and the absorber length extracted. After defining the objective function and the limitations, the optimum values of the boring bar with embedded dynamic absorber extracted and the frequency response function of the optimum tool presented.

References

[1] G. Quintana, and J. Ciurana, "Chatter in Machining Processes: A Review," *International Journal of Machine Tools and Manufacture*, Vol. 51, No. 5, pp. 363–376, May 2011, doi:10.1016/J.IJMACHTOOLS.2011.01.001.

[2] M. Siddhpura, and R. Paurobally, "A Review of Chatter Vibration Research in Turning," *International Journal of Machine Tools and Manufacture*, Vol. 61, pp. 27–47, Oct. 2012, doi: 10.1016/J.IJMACHTOOLS.2012.05.007.

[3] C. Yue, H. Gao, X. Liu, S. Y. Liang, and L. Wang, "A Review of Chatter Vibration Research in Milling," *Chinese Journal of Aeronautics*, Vol. 32, No. 2, pp. 215–242, 2019, doi: 10.1016/j.cja.2018.11.007.

[4] J. Ma, and Y. Ren, "Free Vibration and Chatter Stability of a Rotating Thin-walled Composite Bar," *Advances in Mechanical Engineering*, Vol. 10, No. 9, pp. 1–10, 2018, doi: 10.1177/1687814018798265.

[5] D. I. Suyama, A. E. Diniz, and R. Pederiva, "The Use of Carbide and Particle-pamped Bars to Increase Tool Overhang in the Internal Turning of Hardened Steel," *International Journal of Advanced Manufacturing Technology*, Vol. 86, pp. 2083–2092, Sep. 2016, doi: 10.1007/s00170-015-8328-z.

[6] P. Mihic, "*Vibration - damped Tool Holder*," United States patent, U.S. Patent No. 8,020,474, Washington, DC: U.S. Patent and Trademark Office, Sep. 20, 2011, Available: https://patents.google.com/patent/US8020474B2.

[7] M. Sayuti, A. A. D. Sarhan, T. Tanaka, M. Hamdi, and Y. Saito, "Cutting Force Reduction and Surface Quality Improvement in Machining of Aerospace Duralumin AL-2017-T4 using Carbon Onion Nanolubrication System," *International Journal of Advanced Manufacturing Technology*, Vol. 65, pp. 1493–1500, 2013, doi: 10.1007/s00170-012-4273-2.

[8] F. Xia, Z. Liu, and Q. Song, "Boring Bar with Constrained Damping," *Hangkong Xuebao/Acta Aeronautica et Astronautica Sinica*, Vol. 35, No. 9, pp. 2652–2659, Sep. 2014, doi: 10.7527/S1000-6893.2013.0484.

[9] L. Daghini, A. Archenti, and C. M. Nicolescu, "Design, Implementation and Analysis of Composite Material Dampers for Turning Operations," *World Academy of Science, Engineering and Technology, International Journal of Mechanical and Mechatronics Engineering*, Vol. 3, No. 5, pp. 613–620, 2009, https://www.researchgate.net/publication/263887736_Design_Implementation_and_analyis_of_composite_material_dampers_for_turning_operations.

[10] V. J. Garcia, E. P. Duque, J. A. Inaudi, C. O. Márquez, J. D. Mera, and A. C. Rios, "Pendulum Tuned Mass Damper: Optimization and Performance Assessment in Structures with Elastoplastic Behavior," *Heliyon*, Vol. 7, No. 6, Jun. 2021, doi: 10.1016/j.heliyon.2021.e07221.

[11] L. T. Tunc, and E. Budak, "Effect of Cutting Conditions and Tool Geometry on Process Damping in Machining," *International Journal of Machine Tools and Manufacture*, Vol. 57, pp. 10–19, Jun. 2012, doi: 10.1016/j.ijmachtools.2012.01.009.

[12] A. Yu. Beregovenko, V. G. Kaplun, A. P. Yakovlev, and I. N. Pastukh, "Influence of Nitriding on Damping Properties of Some Metallic Materials," *Strength of Materials*, Vol. 25, No. 8, pp. 604–609, Aug. 1993, doi: 10.1007/BF01151130.

[13] M. Sortino, G. Totis, and F. Prosperi, "Development of a Practical Model for Selection of Stable Tooling System Configurations in Internal Turning," *International Journal of Machine Tools and Manufacture*, Vol. 61, pp. 58–70, 2012, doi: 10.1016/j.ijmachtools.2012.05.010.

[14] A. Rahi, and M. Hosseinpour, "Investigation of Dynamic Absorber Parameters to Improve the Vibration Behavior of Boring Bar," *Iranian Journal of Manufacturing Engineering*, Vol. 9, pp. 1–8, 2022, doi: 10.22034/ijme.2022.163347.

[15] R. F. de Souza Filho, "*Boring Bar with Improved Stiffness*," Inventor; Kennametal Inc, Assignee, U.S. Patent No. 10,040,127, Washington, DC: U.S. Patent and Trademark Office, Aug 7, 2018, Accessed: Jan. 20, 2023, Available: https://patents.google.com/patent/US20150258612.

[16] L. Li, B. Sun, and H. Hua, "Analysis of the Vibration Characteristics of a Boring Bar with a Variable Stiffness Dynamic Vibration Absorber," *Shock and Vibration*, Vol. 2019, No. 1, 2019, doi: 10.1155/2019/5284194.

[17] R. L. Caslstedt, "*Boring Bar*," U.S. Patent 3164041, Jan. 5, 1965, Available: https://patents.google.com/patent/US3164041A/en?oq=US3164041A.

[18] E. Abele, M. Haydn, and T. Grosch, "Adaptronic Approach for Modular Long Projecting Boring Tools," *Cirp Annals-Manufacturing Technology*, Vol. 65, No. 1, pp. 393–396, 2016, doi: 10.1016/j.cirp.2016.04.104.

[19] I. Lazoglu, F. Atabey, and Y. Altintas, "Dynamics of Boring Processes: Part III-time Domain Modeling," *International Journal of Machine Tools and Manufacture*, Vol. 42, No. 14, pp. 1567-1576, 2002, doi: https://doi.org/10.1016/S0890-6955(02)00067-6.