Analytical and Finite Element Modelling of the Boring bar for Stability Studies

B. A. G. Yuvaraju*[1], B. K. Nanda[1] and J. Srinivas[1]

[1]Department of Mechanical Engineering, NIT, Rourkela-769 008, Odisha, India

Abstract–Tool Vibration induced in boring operations not only deteriorates the surface finish of workpiece but also adversely affects tool life and produce noise during the machining operation. Thus, it is essential to suppress these unwanted vibrations. Before applying any damping technique to suppress the chatter, it is necessary to know the behavior of the conventional tool. Therefore, in this paper, a traditional boring bar is modelled as a cantilever Euler-Bernoulli beam for which the first mode of vibration is considered. In addition, simulation of the system is performed in frequency and time domain and obtained stability lobe diagrams and time domain plots respectively. Further, these results are validated with finite element model. It is observed that the results of beam theory and finite element model are in good agreement. For minimizing the vibration levels, the concept of constrained layer damping is implemented with hybrid composite material as damping layer. The resultant vibration levels of such sandwiched boring tool with hybrid composite layer have drastically reduced in comparison with conventional tool.

Keywords: Boring Bar, Chatter, Euler-Bernoulli Beam, Stability Lobes, Time Domain, Tool vibrations.

1. INTRODUCTION

Today, the manufacturing industries are producing a variety of components despite of the difficulties in achieving more accurate and precise products. The uncontrolled vibrations and noise produced during the machining operations often lead to imprecise products and frequent machine breakdowns. Over the last two decades, several researchers focused on the analysis and control of these so called regenerative chatter oscillations produced in different machining operations [1]. Boring is one of the oldest machining operations to enlarge the pre-drilled or cast holes in a component, where highly accurate surface finish of the product is required. Here, the boring tool or bar is the weakest part of the machining system and its motion is time dependent. Therefore, the workpiece deformation during boring operation produces dynamic motion or vibration in the tool. These vibrations deteriorate the surface finish of machined part and tool life. The recent concern is to attenuate the regenerative oscillations induced in the tool during machining. Therefore, it is necessary to suppress these vibrations by adopting some control techniques.

There are different techniques of reducing vibration levels such as designing semi-active vibration absorber systems [2], active control techniques [3] to alter the motor currents and velocity/feed rates etc. However, passive damping is the most economical and popular method among all other methods for vibration minimization due to its feasibility in practical conditions. Polymers and fiber reinforced (FR) composites have gained attention in damping studies and widely used as passive dampers for cutting tools.

Damping of the composite materials had been extensively studied by Lazan [4] and found that the logarithmic decrement values are dynamic stress dependent. Later, Bert [5] and Nashif [6] have investigated the damping capacity of fiber-reinforced composites and found to be higher than metals. Therefore, Rivin and Kang [7] developed a boring tool with viscoelastic layers and observed improved stability, stiffness and damping. Further, Hwang and Kim [8] reported that boring bar made with two or more material increases the stiffness and results in reduced vibrations during boring process. Biju and Shunmugam [9] and Du et al. [10] developed a boring bar with granular particle impact dampers (PID) and found higher energy dissipation compared with conventional

*Corresponding Author: B. A. G. Yuvaraju,
Email: bagyuvaraju@gmail.com
Mobile: +91-8338849852

tools. Further, with advancement to the PIDs, Mei et al. [11] and Pour and Behbahani [12] used smart MR dampers on the boring bar for tuning the natural frequency of the tool, which results reduced chatter and improved stability of machining system. In recent years, piezoelectric damping is an advanced semi-active technique used for suppression of regenerative vibrations. Miguelez et al. [13] and Yigit et al. [14] modelled boring bar with piezoelectric shunt damping, presented an approach to suppress the vibrations and validated them by conducting experiments.

However, constrained layer damping gained popularity compared to other damping methods because easy to implement and economic cost. Therefore, the researchers in some works [15-18] have employed the constrained layered (or sandwiched) tool in boring as well as turning operations and observed a reduction in vibration levels with improved stiffness. Moreover, limited work is noticed on the stability studies of boring operations with sandwiched boring bar.

In the present work, first, the modelling procedure of a conventional boring bar using Euler-Bernoulli beam theory is presented. From the tool tip frequency responses, the stability lobe diagrams are generated for different overhang lengths. The results are validated with the finite element modelling (FEM) of the tool. A hybrid composite material (e.g. glass/SiC-epoxy) is used as damping materials (sandwich material) over the tool surface and modelling is further carried-out with equivalent material properties. The remaining part of paper is organized as follows: analytical and numerical modelling are first presented for the boring operations with regeneration. Also, the modelling of constrained layer damped tool is presented. The results are presented for a test case of boring bar employed for short length internal turning operation.

2. DYNAMIC MODELLING OF THE BORING BAR

The dynamic modelling of the conventional as well as sandwiched (constrained layered) boring bar are elaborated below.

2.1 Euler-Bernoulli beam theory

The bending of the boring bar is considered such that the stiffness of workpiece is much higher than the boring bar. Tangential and radial vibrations are the result of this bending mode. Tangential vibrations are in speed direction, which will not affect the regenerative nature of chip whereas radial vibrations change with cutting depth, which will produce the regenerative effect. On the other hand, the axial and torsional stiffness values of the boring tool are significantly larger than in bending. Hence, the analysis is carried out with the assumption that the boring bar is rigid in the axial and torsional directions. Therefore, the tangential, axial and torsional vibrations are ignored in the present modelling process. Figure 1 shows the cantilevered boring bar in bending with the Euler-Bernoulli beam element.

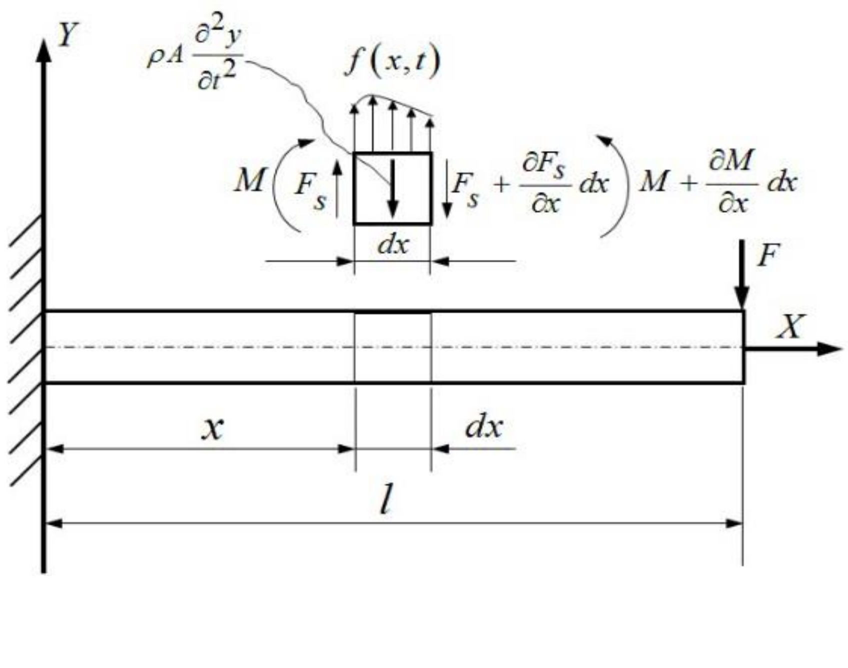


Figure 1. Dynamic model of the boring bar for Euler-Bernoulli beam theory

Equation of motion of the boring bar describing dynamics is given by

$$\rho A\frac{\partial^2 y(x,t)}{\partial t^2}+c\frac{\partial y(x,t)}{\partial t}+EI\frac{\partial^4 y(x,t)}{\partial x^4}=f(x,t) \tag{1}$$

where, E, A, c, ρ and $y(x,t)$ are elastic modulus, area of cross section, damping coefficient, material density and deflection or displacement of the boring tool. $f(x,t)=K_c d\left[y(x,t)-y(x,t-T)\right]$ is the distributed force acting on the boring bar. Equation (1) can be solved using mode superposition principle, where the beam deflection is assumed as:

$$y(x,t)=\sum_{i=1}^{\infty}\chi_i(x)\,q_i(t) \tag{2}$$

where, $\chi_i(x)$ and $q_i(t)$ are the mode or characteristic function and generalized coordinates respectively for the given system. The boundary conditions at the fixed end, i.e., $(x_0=0)$ are $\chi(x_0)=0$ and $\chi'(x_0)=0$ whereas at free end, i.e., $(x_0=l)$ are $\chi''(x_0)=0$ and $\chi'''(x_0)=0$. Substituting (2) in (1) and multiply with $\eta_i(x)$, integrating it over *0* to *1* by applying orthogonality conditions, which are given below:

$$\int_0^l \rho\chi_i(x)\chi_j(x)dx=\delta_{ij} \tag{3}$$

$$\int_0^l EI\chi_i''(x)\chi_j''(x)dx=\omega_{ni}^2\delta_{ij} \tag{4}$$

Using the generalized coordinate system, (1) can be further modified as follows

$$\frac{\partial^2 q_i(t)}{\partial t^2}+2\zeta\omega_{ni}\frac{\partial q(t)}{\partial t}+\omega_{ni}^2 q_i(t)=f_i(t) \tag{5}$$

Equation (5) is used for simulating the process with various values of speed and depth of cut. This time domain simulation is useful for identification of chatter so that one can adopt some damping technique to control the vibration in boring bar.

2.2 Finite element modelling

In the present analysis, the boring tool is considered as cantilever beam and Euler-Bernoulli beam theory is applied for determining the natural frequency. Further, time domain model is established and simulated to investigate vibrations in the boring bar. The tool is discretized into five nodded beam elements with four degrees of freedom at each node are taken, which is shown in Figure 2.

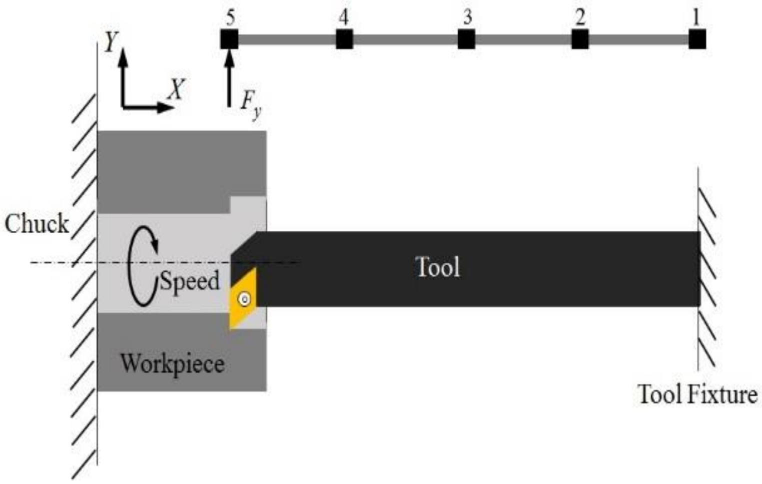


Figure 2. Finite element model of the boring bar

Tobias [19] presented a classical cutting force model for cutting operation in radial direction, which is given by

$$F_y = K_c d\left[y(t) - y(t-T)\right] \quad (6)$$

where, d and K_c are depth of cut and cutting force coefficient respectively. $y(t)$ and $y(t-T)$ are the displacements of current and previous pass of the tool respectively.

The equation of the boring tool in radial cutting direction is given by:

$$[M]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \left\{F_y\right\} \quad (7)$$

where, $[K]$ and $[M]$ are $2n \times 2n$ global stiffness and mass matrices respectively obtained by accumulating the elemental matrices and the boundary conditions of the tool, i.e., for clamped-free are applied. n is number of beam elements are considered along the length of the tool. y_1 and θ_1 are, respectively, displacement and slope, which are equal to zero at node 1 since the boundary condition is clamped. In addition, the dynamic cutting force $\left(F_y\right)$ is set to the $(n+1)^{th}$ node of the force matrix. Damping matrix, i.e., $[C]$ is determined from equation given by

$$[C] = [\phi][M][diag(2\zeta_i\omega_i][M][\phi]^T \quad (8)$$

where, $[\phi]$ is normalized modal matrix which satisfies the conditions like $[\phi]^T[K][\phi] = \left[\text{diag}\left(\omega^2\right)\right]$ and $[\phi]^T[M][\phi] = [I]$. ω_i is natural frequency of i^{th} mode of the system.

Both $[\phi]$ and ω_i are evaluated by solving the Eigen function characteristic equation of the system, i.e., $\left([K] - [\phi]\omega_i^2\right)\{\phi_i\} = [0]$ where $\left\{\phi_i\right\}$ is i^{th} mode shape vector. ζ_i represents system's damping ratio for i^{th} mode and found by performing experimental modal analysis (EMA). Moreover, EMA is used to validate finite element model. Figure 3 shows impact hammer test setup to find out the modal parameters of the boring bar.

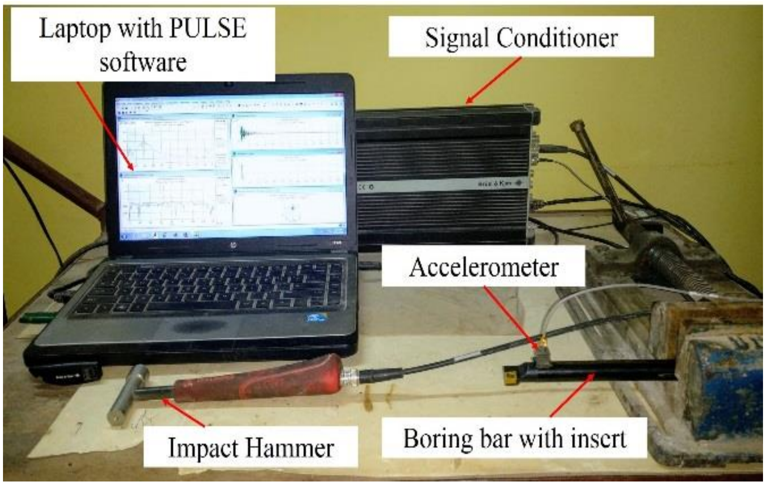


Figure 3. Impact hammer test setup

2.3 Modelling of sandwiched boring bar

Viscoelastic materials are widely used in vibration control due to its high damping capacity. However, these cannot be directly used as structural material because of their low elastic modulus. Therefore, a sandwiched boring bar with three layers is proposed in this paper. It mainly consists of core material (CM), damping layer (DL) and constrained layer (CL). 40cr steel, hybrid composite (glass/SiC-epoxy), 317 stainless steel are respectively considered as core, damping and constrained layer materials. The primary role of damping layer is to improve the fundamental natural frequency by reducing the mean boring bar density and other is to improve the damping capacity. Whereas, the role of constrained layer is to improve natural frequency and stiffness of the boring bar and to protect the damping layer. The thickness of the damping and constrained layers are taken as 2 mm and 1 mm respectively. The diameter of the base material used for the analysis is 20mm. The sandwiched boring bar is shown in Figure 4.

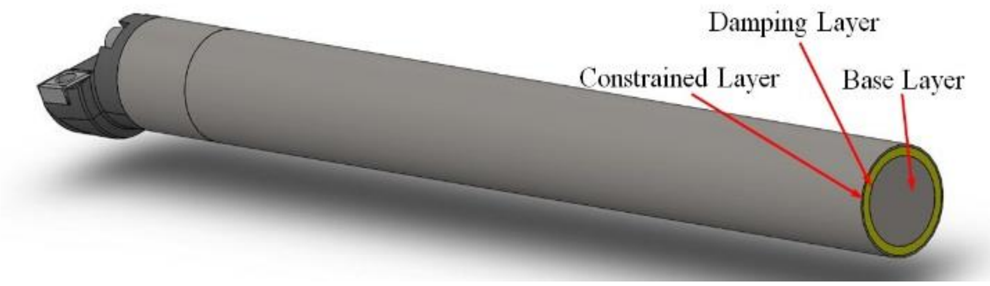


Figure 4. Sandwiched type boring bar

The general methods used in dynamic analysis of composite structure are modal strain energy (MSE), direct frequency response and complex egen value methods. The Last two methods calculate the parameters in complex domain, application of same in complex structure is very costly. Therefore, MSE is adopted in the present analysis and is based on strain energy and loss factor. This method is used to analyze and optimize the damping parameters in finite element analysis.

The strain energy of base material (U_b), damping layer (U_d), constrained layer (U_c) and composite (U_e) are expressed resepectively as

$$U_b = \frac{1}{2}\{\psi\}^T [K]_b \{\psi\} \tag{9}$$

$$U_d = \frac{1}{2}\{\psi\}^T [K]_d \{\psi\} \tag{10}$$

$$U_c = \frac{1}{2}\{\psi\}^T [K]_c \{\psi\} \tag{11}$$

$$U_e = \frac{1}{2}\{\psi\}^T [K]_e \{\psi\} \tag{12}$$

where, $\{\psi\}$ is a real vector of undamped system, $[K]_b$, $[K]_d$, $[K]_c$ are the stiffness matrices of base, damping, constrained layers respectively and $[K]_e$ is an equivalent stiffness matrix of the sandwiched boring bar.

The equivalent stiffness matrix of the sandwiched boring bar can be expressed as a relationship betweeen base, damping and constrained layer stiffness matrices as follows

$$[K]_e = [K]_b + [K]_d + [K]_c \tag{13}$$

The dissipation of energy by the damping layer in cycle is expressed as

$$W_c = \frac{1}{2}\gamma\{\psi\}^T [K]_d \{\psi\} \tag{14}$$

where, γ is the loss factor of damping material.

The loss factor of sandwiched composite boring bar is obtained by

$$\eta = \frac{W_c}{U} = \frac{\gamma[U]_d}{[U]_b + [U]_d + [U]_c} \tag{15}$$

The equivalent density and bending stiffness values are determined from the following equations

$$[\rho S]_e = [\rho S]_b + [\rho S]_d + [\rho S]_c \tag{16}$$

$$[EI]_e = [EI]_b + [EI]_d + [EI]_c \tag{17}$$

After determining the equiavalnet properties, mass and stiffness matrices of the sandwiched boring bar, finite element analysis is performed for evaluating the modal parameters. Further, the time domain (i.e., the tool tip displacement w.r.t time) plots are simulated using the evaluated data.

3. RESULTS AND DISCUSSIONS

3.1 Results of beam theory

The stability state of the system changes if the stiffness of the boring tool is adjusted. For studying the effect of variation of stiffness, the stability lobes are plotted for natural frequencies at 360Hz, 365 Hz and 370Hz, which are shown in Figure 5. The stability lobes of the system shift to the right with an increase in natural frequency, providing an enhanced stability region.

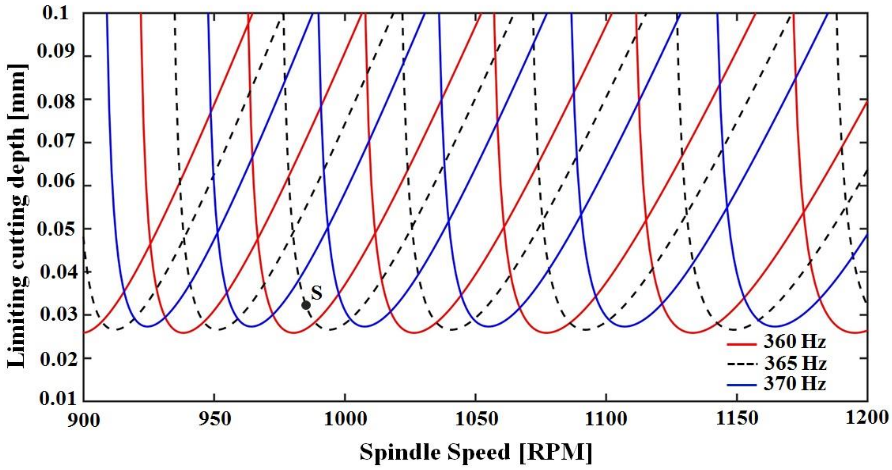
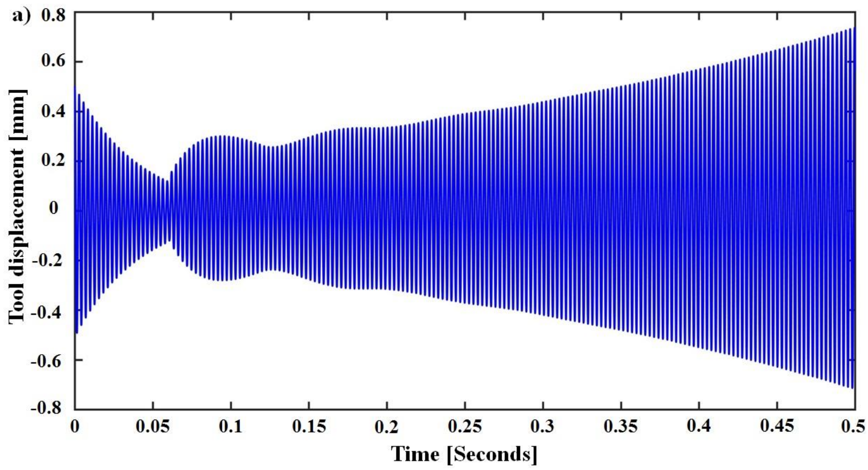
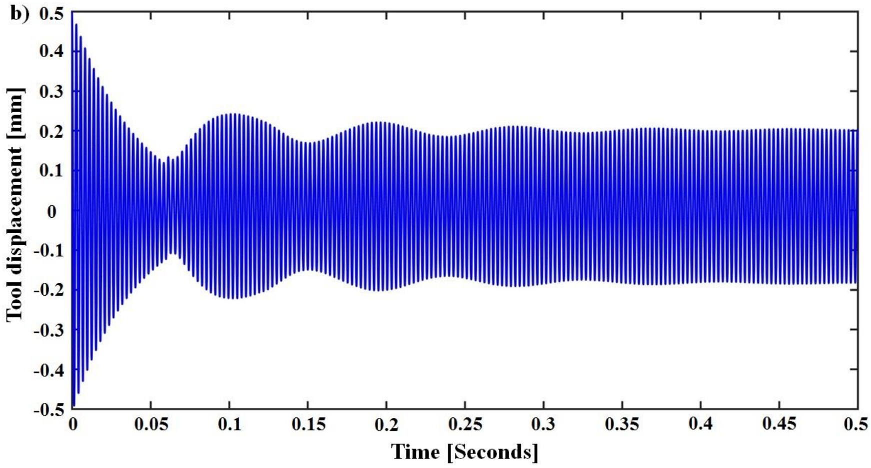


Figure 5. Stability lobes for various natural frequency values

A point S is considered in Figure 5 for the demonstration of the suppression of the boring tool vibrations with speed and cutting depth of 985 rpm and 0.033 mm respectively. From the Figure 5, the point S is in the unstable region at a frequency of 360 Hz and corresponding time domain plot is given in Figure 6a. This means the vibration of the tool increased without boundary resulting chatter in the machining process.





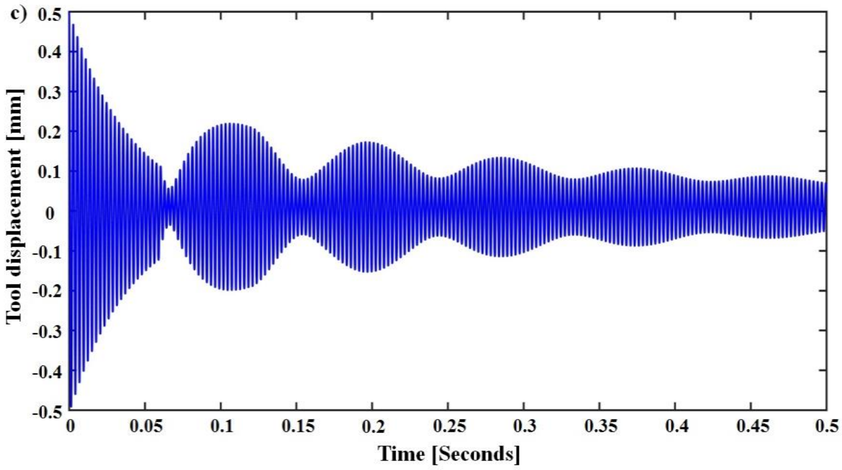


Figure 6. Time domain simulations of the boring bar found form beam theory for a natural frequency equals to a) 360 Hz, b) 365 Hz and c) 370 Hz

S is located on the stability lobes when the natural frequency increased to 365 Hz and respective tool vibration is shown in Figure 6b. This results in a constant vibration level at the tool-tip, which shows the system is critically stable. Similarly, S is in the conditional stable state at a frequency of 370 Hz and the tool-tip vibration is shown in Figure 6c. It is observed that the vibration decays quickly, which indicates the machining process is stable.

It can be concluded from the simulations that the vibrations occur during boring process in the tool can be attenuated by improving the system's natural frequency. This can be achieved by employing the proper vibration control technique.

3.2 Results of finite element modelling

In this section, the results obtained from the finite modelling of the tool are elaborated. The global stiffness and mass matrices are built. Further, a convergence analysis is carried out to find the pre-dominant natural frequency of conventional and sandwiched boring bar at first vibration mode. Moreover, EMA is conducted for determining the modal parameters, which validates the results of FEM model. Figure 7 shows the frequency response plot for the conventional as well as sandwiched boring tool obtained from the impact hammer test and the fundamental natural frequencies are found as 368 Hz and 476 Hz respectively. These frequencies are approximately equals to the frequency determined from FEM. This indicates the results of FEM are in good agreement with experiments.

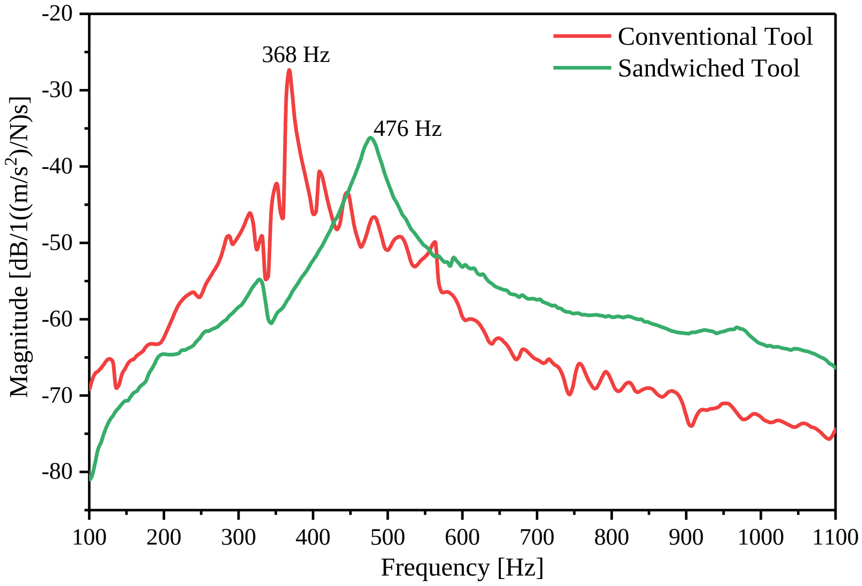


Figure 7. Frequency response plot obtained from impact hammer test for conventional and sandwiched boring bar

In addition, the damping ratios are evaluated from modal test and (8) is used for determining the damping matrix $[C]$. Further, time domain simulation are performed using MATLAB for the equation of the motion shown in (7). This simulation is used to determine the limiting cutting depth in boring operation for all values of speed, which gives the stability lobe plot. Finding stability lobes in the frequency domain is easier whereas time taking process in time domain. Even though, time domain simulations are more desirable than the frequency domain (stability lobes) plots because condition monitoring of tool is more comfortable in the time domain. The time domain plots obtained from the FEM for both conventional as well as sandwiched tool are shown in Figure 8. The cutting speed and depth of cut used for the simulation are 985 rpm and 0.03 mm respectively. The cutting stiffness coefficient is taken as 1680 N/mm. The simulation is performed using the delay differential equations using MATLAB.

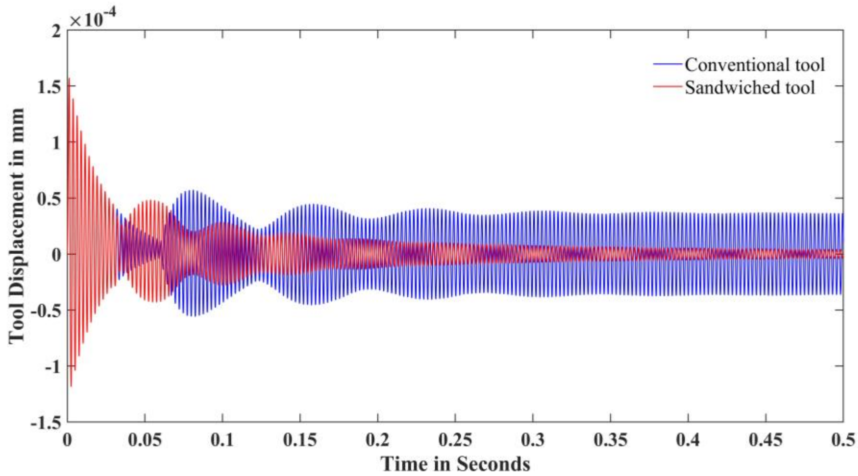


Figure 8. Tool tip displacements vs time plot for both the conventional and sandwiched boring tool using finite element method

From Figure 8, it has been observed that the time domain obtained for conventional boring tool from FEM shows constant vibration level which indicates critically stable state of the machining process for a given set of cutting conditions. It is also found that the time domain of sandwiched boring tool shows vibration decay results a stable cutting process for a set of parameters. This is due to the damping layer introduced in the sandwiched boring bar has the high damping capacity, which can dissipate the vibration energy produced in the tool within the volume of the material. Therefore, the vibrations can be controlled with the provision of damping materials. The sandwiched boring bar changes the dynamic stiffness through constrained layer and improve the damping performance through damping layer. Hence, it is concluded that the results of beam theory are very useful to study the FEM results. It is found that the sandwiched boring bar shows better results compared to conventional tool. Thus, the vibrations in the tool are controlled with use of sandwiched boring tool. It is suggested that change of damping material in the sandwiched tool may further improves the tool stiffness as well as damping capacity and results in reduced vibrations.

4 CONCLUSION

The boring bar is modelled as cantilevered Euler-Bernoulli beam and equation of motion is expressed in generalized coordinates. This equation has solved to obtain the time domain simulation. The stability lobes are plotted for different frequencies and a point 'S' is located on it for further analysis. In addition, the time domain plots are simulated using MATLAB by varying speed and depth of cut to observe the vibration level in the boring bar and carried out the stability analysis. Further, finite element modelling of the conventional and sandwiched boring bars has been performed and results of beam theory are used for analyzing the FEM results. It has been observed that the time domain (tool tip displacement) of conventional tool is critically stable (constant vibration) whereas stable (quick decay) for sandwiched boring tool. Therefore, it is concluded that the sandwiched boring bar shows better results compared to conventional tool for reducing the vibrations. This is mainly due to the damping layer provided in the sandwiched boring bar has high damping capacity, which dissipates the vibration energy produced in the tool within the volume of the material. Also, the sandwiched boring bar changes the dynamic stiffness through constrained layer and improve the damping performance through damping layer. Therefore, the sandwiched boring tool is very helpful for controlling tool vibrations in boring operation. Further, it is suggested that the damping performance and tool stiffness may improve respectively by changing the damping and constrained layer material of the sandwiched boring bar.

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