A Receptance Coupling Approach to Design Damped Boring Bars

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Abstract

Damped boring bars are necessary in deep hole boring operations as regular boring bars are prone to severe chatter. Boring bars are often damped by integrating a tuned vibration absorber that will damp the flexible mode(s) and minimize chatter. This paper introduces a new method to design damped boring bars using the receptance coupling approach. The boring bar is modeled as a cantilevered Euler-Bernoulli beam. Receptances, i.e. frequency response functions of the boring bar are obtained at its free (cutting) end and at potential locations where the receptances of a tunable mass damper can be synthesized. The objective of this kind of coupling is to maximize the dynamic stiffness of the boring bar at the free end. Treating the boring bar and the tunable mass damper as separate substructures facilitates sensitivity analysis for identifying optimal parameters of the tunable mass. Sensitivity analysis is carried out to investigate influence of mass, stiffness, damping properties. Absorber effectiveness was observed to saturate beyond certain levels of stiffness and damping values. For boring of an Aluminum alloy, simulated chatter-free depth of cut was found to increase to 0.58 mm with the optimally tuned and damped bar as compared to being 0.15 mm for the original bar, a significant improvement. The methods presented are simple and technically elegant and can be used to optimally design damped boring bars to increase their resistance against chatter vibrations.

Keywords: Receptance coupling, Boring bars, Tuned mass damped

1. INTRODUCTION

Deep hole boring necessitates the use of long slender boring bars with length to diameter ratios ranging from six to ten or higher. These boring bars are essentially cantilevered with cutting forces acting on the free end. On account of being long, slender and cantilevered, and because of their inherently low structural damping, the boring bars tend to vibrate with large amplitudes under the action of cutting forces. Such process induced large amplitude vibrations often result in machining instabilities, i.e. chatter. Chatter deteriorates surface finish and results in high tool wear or breakage; thereby limiting productivity, precision, and material removal rates. To avoid chatter, boring bars must possess improved dynamic stiffness and damping behavior.

Dynamic stiffness and damping are often characterized using frequency response functions (FRFs), i.e. receptances. Moreover, following the classical chatter model of Tobias and Fishwick [1], it is common to relate stable cutting conditions to the inverse of the minimum real part of tool point FRF. Hence, the approach adopted in this paper is to focus on reducing the magnitude of the peak in real part of the tool point FRF by presenting a strategy to design damped boring bars. It is hypothesized that reducing the magnitude of the peak in real part of the FRF will result in chatter free boring processes.

There exist many methods in the literature to maximize the dynamic stiffness and improve damping, and in turn obtain chatter free boring processes. These methods include solutions using tuned dampers – passive [2,3], semi-active [4], and/or active [5]. Though active systems can sometimes outperform passive ones, in most cases, a well-designed passive damper is preferred due to its simplicity, cheaper costs, and Industrial viability. Hence, this paper also focuses on articulating a solution using a passive tuned mass damper (TMD) to increase resistance against chatter.

Design of TMDs include the classical approach of Den Hartog [6], in which analytical closed form solutions were presented to integrate a tuned damper for the case of zero damping in the main simplified single degree of freedom (SDOF)vibrating system. Rivin and Kang [7] analyzed the more general case of damped original systems and different forcing functions. Sims [8] presented analytical solutions to tune vibration absorbers (dampers) to suppress chatter for the case of zero damping in main SDOF systems.

Though effective, these [6-8] tuning methods do not consider the effect of TMD location relative to the free end of tool and model continuous systems as simplified SDOF systems. To address the issue of designing dampers that can be positioned at any arbitrary distance away from the free end of the tool, we present an approach using receptance coupling [9]. Receptance coupling, a structural modification tool, is used to couple two separate subsystems in the frequency domain such that final response of the main system can be obtained as necessary. In this study, two subsystems: unmodified boring bar and tuned mass damper, as shown in Fig. 1, are coupled together to obtain the modified FRF of the damped bar. TMD parameters were varied in a feasible range and modified response was studied to obtain the optimized damper parameters.

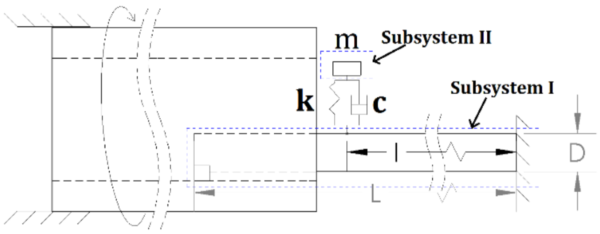

Figure 1: Schematic of long boring bars

The remainder of the paper is organized as follows: a chatter stability model is introduced in Section 2, followed by introducing the receptance coupling model in Section **3**. Boring bar and TMD dynamics have been defined in Section 4. Sensitivity analysis of tuned boring bar is carried out in Section **5**, and recommended tuning parameters are given in Section6, which is followed by the main conclusions in Section **7**.

ISBN: 978-93-80689-28-9

2. MACHINING STABILITY MODEL

Since the purpose of developing damped boring bars is to overcome chatter, the relationship between dynamic stiffness, damping, and the limiting stable, chatter-free cutting parameters (depth of cut) can be characterized as [1]:

$$a_{lim,crit} = \frac{-1}{2K_f\mu \min\left(Re\left(H(\omega_c)\right)\right)} \quad (1)$$

wherein $a_{lim,crit}$ is the critical limiting chatter-free depth of cut, K_f is cutting force coefficient which depends on the tool geometry and work piece material combination; μ is the directional factor; and $Re\left(H(\omega_c)\right)$ is the real part of the FRF, H at a chatter frequency, ω_c. It is evident that critical depth of cut is inversely proportional to the negative peak in real part of tool point FRF. Thus, a decrease in the magnitude of the negative peak in real component of tool point FRF is taken as the objective function for tuning of TMD parameter.

3. RECEPTANCE COUPLING

Receptance coupling (RC) methods facilitate analysis of complex systems by synthesizing the FRFs, i.e. receptances of their individual subsystems. In this study, modified damped boring bar is the final assembled system which has been modeled as two sub-systems: original boring bar as the 1st subsystem, and the TMD mass as the 2nd subsystem, with a spring and damper coupling the two, see Fig. 1.

Unlike the other classical approaches to the design of TMDs [6-8] which cannot account for placement considerations of the TMD along the length of the boring bar, and are limited to only placing the TMD at the free end, which is not feasible in boring, since the free end also corresponds to the cutting end, the RC method allows designing TMDs to be positioned at different sections along the boring bar. Though the RC method can also suggest an optimal placement location, in the present study, absorber (TMD) is attached at the viable distance of 0.8 times the total cantilevered length ($l = 0.8L$) as free end is not accessible.

If the direct receptances of the original boring bar at its free end and the TMD coupling location are known a priori, as are the cross receptances between the free end and the TMD coupling location, the receptance of the TMD can be tuned in such a way such that the synthesized receptance, H_{11}, for the combined model at the free end can be shown to be:

$$H_{11} = h_{11} - h_{12a}\left(h_{2a2a} + h_{2b2b} + \frac{1}{k'}\right)^{-1} h_{2a1} \quad (2)$$

$$k'(\omega) = k + iC\omega$$

wherein h_{11} is the receptance at the free end for the original boring bar; h_{2a2a} is the receptance at the coupling location of the TMD for the original boring bar; h_{12a} and h_{2a1} are the cross receptances between the free end and the coupling location of the TMD, again for the original boring bar, and since the structure is symmetric and linear, these cross receptances are the same and equal; h_{2b2b} is the tunable receptance of the TMD; and k' is the effective stiffness. Though receptances in Eq. (2) are frequency (ω) dependent, ω is omitted for brevity.

4. RECEPTANCES FOR SUBSYSTEMS

The subsystem level receptances for the original boring bar and the TMD are obtained as discussed below.

A. Original Boring Bar Model

A long slender boring bar with a length to diameter ratio of 12 is used in this study. Boring bar is modeled with Euler Bernoulli beams using the Finite Element (FE) method. Beam and FE parameters are given in Table 1. Note that the cutting insert and insert cartridge were ignored while modeling the boring bar.

Table 1: Boring Bar FE Model Details

Parameters	Values
Diameter (D)	25mm
Length (L)	300mm
Density (ρ)	7850 Kg/m^3(Mild Steel)
Young's Modulus (E)	200 GPa(Mild Steel)
Absorber location($l = 0.8L$)	240 mm
Number of elements	300 (equal length)
Element Type	Linear, 2 nodes per element
Node degree of freedom (DOF)	2 DOF per node $\left(u, \frac{\partial u}{\partial x}\right)$

An eigenvalue problem was formulated for the equation of motions using the mass and stiffness matrices. Solving the eigenvalue problem form of the equation of motion gives the eigen values (natural frequencies) and eigenvectors for the beam. Receptances can be evaluated using the eigenvalues and eigenvectors as:

$$h_{pq}(\omega) = \sum_{k=1}^{n} \frac{\Phi_{p_k}\Phi_{q_k}}{-\omega^2 + {\omega_k}^2 + 2i\zeta\omega\omega_k} \quad (3)$$

wherein h_{pq} is the receptance with response at location p and excitation at q. Φ_{p_k} and Φ_{q_k} are the mass normalized eigen vectors at p, q for mode 'k'; ω_k is eigenvalue for mode 'k'. Damping ratio, ζ, that includes the influence of structural damping is assumed to be uniform for all modes at 5%.

Real parts of the receptances obtained using Eq. (3) are shown in Fig. 2. Response is shown only up to 250 Hz, since the higher frequency modes are several orders of magnitude dynamically stiffer than the dominant low frequency response. As evident from Fig. 2, the real part of the receptances clearly show the response at the free end (h_{11}) to be greater (higher amplitude of the negative peak) than at the coupling end (h_{2a2a}) or between the free end and the coupling ends ($h_{12a} = h_{2a1}$).

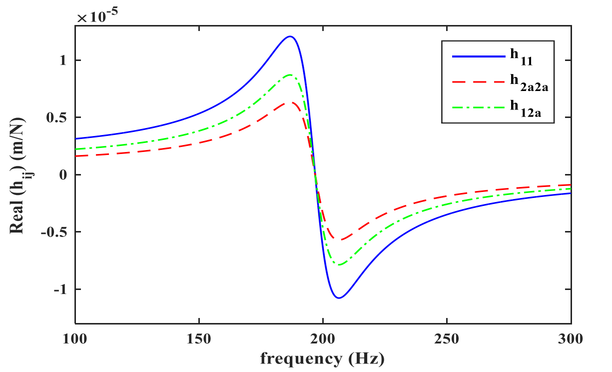

Fig.2: Direct and cross real parts of the receptances of the original boring bar model

B. Absorber (TMD) Model

The TMD mass is modelled as a single degree of freedom (SDOF) system with mass m. The receptance of this SDOF system can be shown to be:

$$h_{2b2b}(\omega) = \frac{-1}{m\omega^2} \quad (4)$$

Further, for a given absorber stiffness K and damping C, flexible element in the assembled system is used as effective stiffness$k'(\omega) = k + ic\omega$.

To tune the absorber to the optimal parameter set of m, k, c combination, all the parameters were varied over a range according to Table 2.Maximum free mass is selected to be ~10% of the beam mass (1.14 kg). Stiffness is selected such that TMDs natural frequency matches the1st and most flexible mode of the boring bar. Damping values are selected on the basis of physical feasibility.

Table 2: Absorber Parameters Used in Sensitivity Analysis

Parameter	Units	Value			
Free Mass (m)	kg	0.03	0.06	0.09	0.12
Stiffness (k)	*1e4 N/m	4	8	12	16
Damping (c)	N-s/m	10	30	50	70

5. INFLEUNCE OF TMD PARAMETERS ON DAMPED RESPONSE OF THE BORING BAR

Influence of m, k and c of the TMD on the damped response of the boring bar is investigated as below, by sequentially fixing two parameters and varying the third to check the sensitivity of the damped assembled response (H_{11}) on the TMD parameters.

A. Mass

Effect of varying free mass was studied by keeping the stiffness and damping in absorber constant. It is observed from Fig. 3 that increasing the mass may not always improve the resistance to chatter (resistance to chatter being inversely proportional to $\min(Re(H_{11}))$.

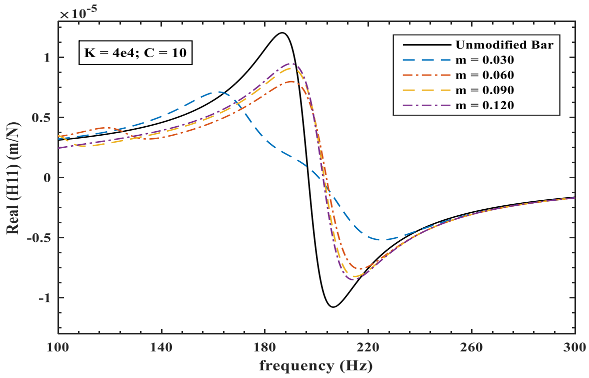

Fig 3: Effect of Free Mass on Real Part of Assembly Response for $k = 40\ kN/m$and $c = 10\ N-s/m$

B. Stiffness

For a fixed mass and damping, stiffness has to betuned to a particular (optimum) value to get the desired performance of absorber.Form Fig. 4 it is evident that $k = 8e4$ gives the best result, i.e. $\min(Re(H_{11}))$. Higher/lower stiffness has less than optimum results.

C. Damping

For a fixed mass and stiffness, continuously increasing damping in the absorber may result in inefficiency of the damper. This effect is similar to the fact that energy dissipation for a SDOF increases with the increase in damping, but after a threshold (damping ratio =1) this has adverse effects. Same can be inferred from the Fig. 5.

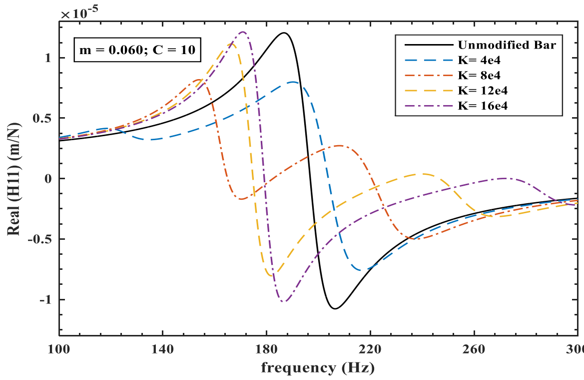

Fig 4: Effect of Stiffness on Real Part of Assembly Response for $m = 0.06\ kg$ and $c = 10\ N-s/m$

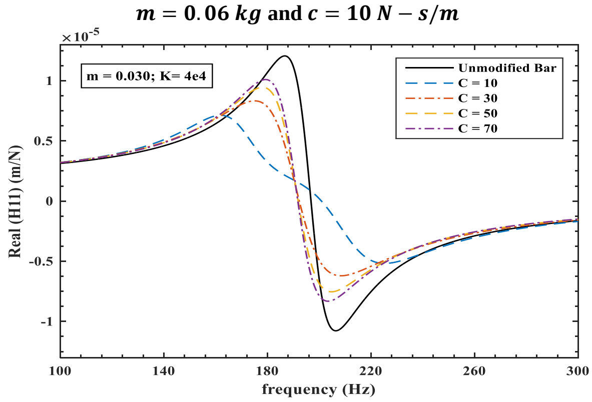

Fig 5: Effect of Damping on Real Part of Assembly Response for $m = 0.03\ kg$ and $k = 40\ kN/m$

6. RECOMMENDED TMD PARAMETERS

It is evident from discussions in Section 5, that each of the three TMD parameters, i.e. m, k, and c influence the damped response of the boring bar differently. Hence, to find the optimum set of parameters, a large 3D-matrix was created such that each of its data points stores the absolute value of minimum negative peak in the real part of assembled FRF. This matrix is then scanned to find the optimum set of k, and c corresponding to each value m.

The procedure to scan the 3D-matirx to find the optimum data set can be understood from Table 3, which reports the absolute minimum of the negative of the real partof the assembled FRF for a selective subset of absorber parameters given in Table 2. For example, if we consider the ***Bold Italicized***data point (***0.4997 X 10^{-5}m/N***), this data point represents the value of the negative peak in real part of assembly response for a massof 0.06 kg, stiffness of 80kN/m and damping of10N-s/m. Remaining entries in Table 3 can be similarly interpreted. Optimal stiffness and dampingwere found by varying $k$and $c$in the specified range for a fixed value of free mass, andthe real part of assembly response was analyzed to obtain the absolute minimum value of its negative peak. This procedure can be again explained using Table 3. For example, if we were to find the optimum parameter set for free mass of 0.06 kg, data points corresponding to the second row need be scanned. On scanning the second row, the absolute minimum of the real part is observed to be 0.3478 X 10^{-5}m/N, which correspond to a stiffness of 80 kN/m and a damping of 30N-s/m. Similarly other absolute minimum values of the negative part of the real part corresponding to each value of the free mass are obtained, and these data points have been underlined in Table 3.

Table 3: Absolute Value of Negative Peak in Real Part (m/N X 10^{-5}) of Modified Receptance for Different TMD Parameters

S. No.	Free Mass m(kg)	c = 10 N-m/s				c = 30 N-m/s				c = 50 N-m/s			
		Joint Stiffness (k, N/m)											
		4e4	8e4	12e4	16e4	4e4	8e4	12e4	16e4	4e4	8e4	12e4	16e4
1	0.030	0.5180	0.9190	1.0629	1.0922	0.6207	0.7861	0.9740	1.0435	0.7539	0.8180	0.9395	1.0118
2	0.060	0.7597	*0.4997*	0.8046	1.0206	0.5618	0.3478	0.4767	0.7932	0.5319	0.3839	0.4566	0.6943
3	0.090	0.8228	0.6700	0.5130	0.8498	0.6012	0.4441	0.3174	0.4418	0.5106	0.3754	0.2683	0.3119
4	0.120	0.8492	0.7438	0.6291	0.6906	0.6268	0.5068	0.3993	0.3160	0.5161	0.4075	0.3158	0.2473

a

b

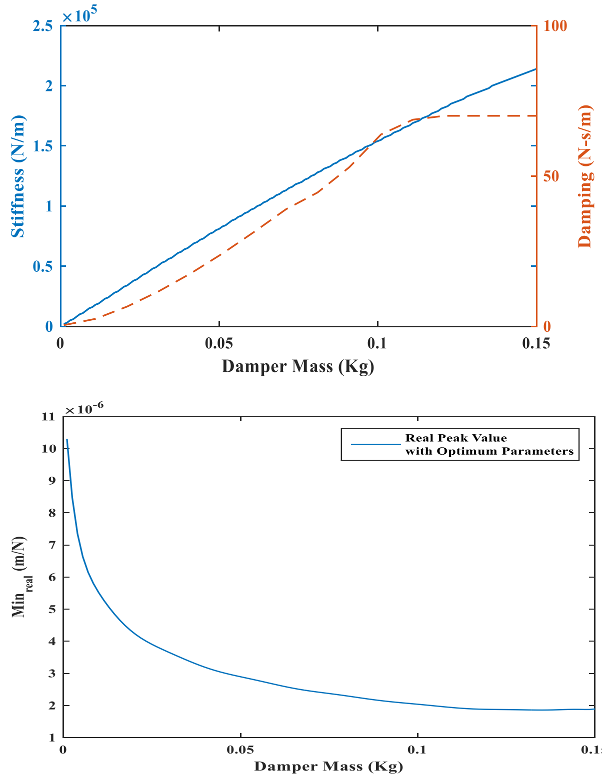

Fig 6: (a) Recommended Values of k and c for a given Mass; (b) Absolute value of negative peak in Real Part of Tool Point FRF with recommended k and $c$parameters.

The above procedure is followed for all parameter levels of Table 2, and the results are summarized in Fig.6. It is clear from Fig 6 that damping and stiffness should be increased as we increase the free mass of the absorber. Fig 6(b) shows the absolute value of real peak in assembly response given that recommended parameters are used. It is evident that beyond a threshold value of mass, increased damping does not necessarily translate to reduction in amplitude of the absolute minimum of the negative part of the real part of the FRF.

Finally, the influence of the optimized TMD on the critical chatter free stable depth of cut is evaluated for the case of cutting soft Aluminum with an effective cutting force coefficient (K_f) of 600e6 N/m^2 and with the directional factor (μ) of 1. For the original (unmodified) boring bar, the critical chatter free depth is evaluated to be 0.15mm (with the negative peak in real part of receptance being 1.0778e-05m/N); whereas for the boring bar with a TMD that has a mass of 0.05 kg, stiffness of 85 kN/m and damping of 25 N-s/m, the chatter free critical depth of cut is evaluated to be0.58 mm (with receptance peak magnitude of 2.8987e-06m/N). This translates to a ~400% improvement over the original case, a significant improvement.

7. CONCLUSION

This paper presents a new receptance coupling based approach to design damped boring bars. Boring bar was modeled as an Euler Bernoulli beam, and the absorber was modelled as a damped single degree of freedom spring-mass system. The receptance coupling model allows integration of the absorber at arbitrary locations along the length of the boring bar, something not possible with the earlier reported classical methods. Receptances of the separate substructures were synthesized to obtain the damped response of the boring bar at its free (cutting) end. Optimal absorber stiffness and damping parameters were obtained as a function of its free mass. Absorber effectiveness was observed to saturate beyond certain levels of stiffness and damping values. Chatter-free depth of cut for boring was found to increase to 0.58 mm with the tuned bar as compared to being 0.15 mm for the original bar, a significant improvement.

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