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Analytical investigation of boring bar for chatter

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Abstract—*In this paper, design and analysis of turnable holder for boring bar is described. Analytical receptance coupling method is used to couple the boring bar and holder assembly. The flexible holder natural frequency is matched to the clamped natural frequency of the tool, a new dynamic system is obtained with reduced possibility of chatter. The flexible holder supports the boring bar and acts as a dynamic absorber for the boring bar.*

Keywords—boring bar, chatter, flexible holder , receptanccoupling.

I. INTRODUCTION

During metal cutting operations, vibratory motion between a cutting tool and work piece can lead to reduce cutting performance accuracy. Such vibrations can cause the cutting tool, work piece, and/or machine to become damaged. Self-excited vibrations, or chatter, between the cutting tool and work piece can cause poor surface finish, tool breakage, and other unwanted effects. When chatter does occur, the machining parameters must be changed and, as a result, productivity may be adversely affected.

One example of tools that may encounter excessive vibration is boring bars, which are typically used to fabricate deep holes. A primary difficulty in their use is that the holes tend to be deep and narrow so boring bars must be long and have small diameters. Therefore, during machining, the variable cutting force causes the tool to deflect and leave a wavy surface behind. When the cutting edge encounters this wavy surface in the next revolution, additional forces and deflections may be caused which can lead to chatter.

Various methods for reducing boring bar vibration are currently used for example,

- Internal vibration absorbers.
- Reduction of cutting force
- Periodic variation of cutting condition
- Enhancement of structural stiffness
- Passive vibration absorber
- Active dampers

Here, we describe a new method to reduce tool vibrations by providing a flexible holder with dynamics tuned to match the boring bar dynamics. The flexible holder supports the boring bar and acts as a dynamic absorber for the boring bar. The flexible holder natural frequency is matched to the clamped natural frequency of the tool, thereby reducing the amplitude of vibration at the free (cutting) end of the bar. In this paper we present both an analytical solution, which applies Euler-Bernoulli beam theory [1], combined with receptance coupling techniques [4].

The advantage of implementation of new method is that it does not require the tool changing characteristics to match the holder frequency and clamped bar natural frequencies. The "modal mass effect" is realized by adjusting the position of a mass attached to the tool that enables the tool dynamics to be tuned with the holder dynamics. The overall goal of the providing new method, flexible holder with dynamics tuned to match the boring bar dynamics (modal mass effect) is to use a single holder for a set of varying length and diameter of boring bars. The holder can then be quickly and efficiently tuned for use (through the modal mass effect) for the current boring bar with pre-determined mass positions.

In this paper we present an analytical solution, which applies Euler-Bernoulli beam theory and receptance coupling techniques..A holder-boring bar is designed and frequency response measurements of the boring bar alone are compared to the measured response of a prototype holder-boring bar assembly.

II. RECEPTANCE COUPLING METHOD

Closed-form, Euler-Bernoulli beam receptances[1] were used to describe an ISO A10-SCLPR2 NE4 boring bar with a length to diameter (L: D) ratio of 6:1. This high L:D ratio was selected since the focus of this work is the improvement of the dynamic stiffness for these inherently low stiffness situations. A diameter of 15.9 mm was chosen because this is the smallest diameter forcommercially available "tunable" boring bars (with dynamic absorbers located inside the bar).

Due to external force applied at the free end for the clamped free beam model of steel boring bar.it was developed using equation(1).

$$h_{jj} = h_{kk} = \frac{-F_5}{EI(1+i\eta)\lambda^3 F_3} \qquad (1)$$

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$$h_{jk}=h_{kj}=\frac{-F_8}{EI(1+i\eta)\lambda^3F_3} \qquad (2)$$

Where,

$$F_3=\cos\lambda L\cdot\cosh\lambda L-1$$
$$F_5=\cos\lambda L\cdot\sinh\lambda L-\sin\lambda L\cdot\cosh\lambda L$$
$$F_8=\sin\lambda L-\sinh\lambda L$$

$$l_{jj}=-l_{kk}=\frac{-F_1}{EI(1+i\eta)\lambda^2F_3} \qquad (3)$$

$$l_{jk}=-l_{kj}=\frac{-F_{10}}{EI(1+i\eta)\lambda^2F_3} \qquad (4)$$

$$n_{jj}=-n_{kk}=\frac{-F_1}{EI(1+i\eta)\lambda^2F_3} \qquad (5)$$

$$n_{jk}=-n_{kj}=\frac{-F_{10}}{EI(1+i\eta)\lambda^2F_3} \qquad (6)$$

$$p_{jj}=-p_{kk}=\frac{F_6}{EI(1+i\eta)\lambda F_3} \qquad (7)$$

$$p_{jj}=-p_{kk}=\frac{F_6}{EI(1+i\eta)\lambda F_3} \qquad (8)$$

Where,

$$F_1=\sin\lambda L\cdot\sinh\lambda L$$
$$F_6=\cos\lambda L\cdot\sinh\lambda L+\sin\lambda L\cdot\cosh\lambda L$$
$$F_7=\sin\lambda L+\sinh\lambda L$$
$$F_{10}=\cos\lambda L-\cosh\lambda L$$
$$m=\frac{\pi(d_0^2-d_0^2)L\rho}{4}$$
$$\lambda^4=\frac{\omega^2 m}{EI(1+i\eta)L}$$

E= Elastic modulus
I =2nd area moment of inertia
η = frequency-independent damping coefficient
d_o =outer diameter,
d_i = inner diameter (set equal to zero if the beam is not hollow),
L = length
ρ = density
ω= frequency (in rad/s)
i=1, 2
j=1, 2

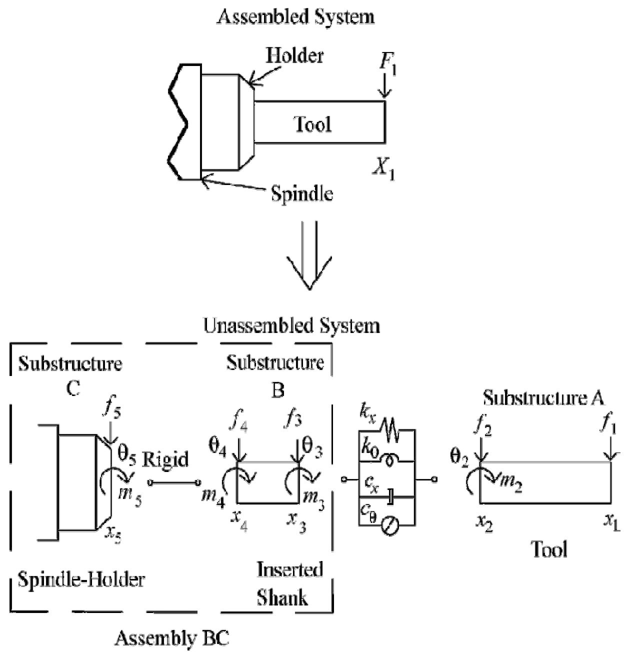

Fig. 1 Schematic representation of receptance Coupling Model boring bar

The sub assembly BC receptance ($R_{33(BC)}(\omega)$) coupling according to Euler beam theory is given by

$$R_{33(BC)}(\omega)=R_{33}-R_{34}(R_{44}+R_{55})^{-1}R_{43} \qquad (9)$$

Where,

$$R_{jk}(\omega)=\begin{pmatrix} h_{jk} & l_{jk} \\ n_{jk} & p_{jk} \end{pmatrix}=\begin{pmatrix} \frac{x_j}{f_k} & \frac{x_j}{m_k} \\ \frac{\theta_j}{f_k} & \frac{\theta_j}{m_k} \end{pmatrix} \qquad (10)$$

Take j,k=3,4,5 inequation 1 to 8equation10& find receptance components $R_{33}, R_{34}, R_{44}, R_{43}, R_{55}$ and then put values in equation 9 to get receptance of sub assembly $R_{33(BC)}(\omega)$

Equation 9 gives the receptance of the sub assembly (tool holder, sleeve and boring bar) in math lab from that we can predict vibration amplitude of subassembly The receptance matrix of the assembly $G_{(ABC)}(\omega)$) according to Euler beam theory is given by

$$G_{(ABC)}(\omega)=R_{11}+R_{12}(R_{22}+R_{33(BC)}+K^{-1})^{-1}R_{21} \qquad (11)$$

Where,

K= 6.35×10^{5} N/m K=Stiffness [1]

η= 6.5×10^{-4} η=damping ratio [1]

$G_{(jk)}$ = Receptance of assembly(sub structure are coupled to produce assembly)

$R_{jk}(\omega)$ =Substructurereceptances.

x_j = Assembly displacement and rotation at coordinate j

f_k, m_k = force and moment applied to assambly at coordinate k

$$G_{jk}(\omega)=\begin{pmatrix} H_{jk} & L_{jk} \\ N_{jk} & P_{jk} \end{pmatrix}=\begin{pmatrix} \frac{X_j}{F_k} & \frac{X_j}{M_k} \\ \frac{\theta_j}{F_k} & \frac{\theta_j}{M_k} \end{pmatrix}$$
$$R_{jk}(\omega)=\begin{pmatrix} h_{jk} & l_{jk} \\ n_{jk} & p_{jk} \end{pmatrix}=\begin{pmatrix} \frac{x_j}{f_k} & \frac{x_j}{m_k} \\ \frac{\theta_j}{f_k} & \frac{\theta_j}{m_k} \end{pmatrix} \quad (12)$$

Take j,k=1,2,3in equation 1 to 8, equation 12 &find receptance components R_{11}, R_{12}, R_{22} R_{21} and then put values in equation 11 to get receptance of assembly ($G_{(ABC)}(\omega)$)

Equation 11 gives the receptance of the assembly (tool holder, sleeve and boring bar) in math lab from that we can predict vibration amplitude.

III. VIBRATION ANALYSIS

The implementation of MATLAB in chatter suppression considers the design parameters like mass stiffness of tool, damping of tool and diameter of tool, length of tool, direct and cross receptance and various boring bar conditions like clamped free and an values of the parameter can be predetermined by simulation and analysis of the required model. The above design parameters mentioned parameters are very well expressed in the form of equations in above section.

In this section the effect of various parameters on the vibration amplitude are studied.

3.1 Frequency Response Function for cantilever Beam

Figure 2 shows the analytical frequency response function (FRF) for lateral vibration x at the free end due to an external force F applied at free end for clamped free model of steel boring bar. Its developed using equation (1) where j,k=1, E=200GPa, $I=\frac{\pi}{64}d^4$ is the area of moment of inertia, $\eta=0.0015$ is the unit less solid damping factor.

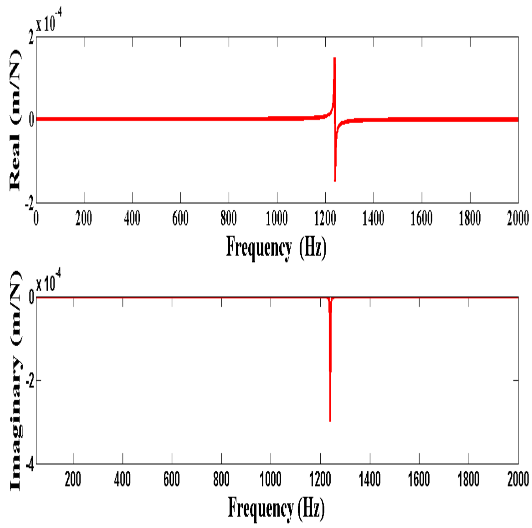

Fig. 2 Cantilever response of 6:1 boring bar

3.2 Effect of Peak ratio for stiffness and damping Factors on vibration amplitude

The role of the holder stiffness and damping characteristics on the system are shown in Figure 3. The top graph shows the ratio of the maximum holder-bar assembly FRF magnitude to the maximum fixed-free boring bar magnitude as a function of a stiffness factor, $factor=\frac{k_{holder}}{k_{bar}}$, where$k_i$is the modal stiffness value, while damping is held constant. The
Bottom is the reverse; damping is varied while stiffness remains constant. In this case, $factor=\frac{c_{holder}}{c_{bar}}$ where $c_i$is the modal damping value In both instances, the bar and holder natural frequencies were matched for all factor values.

A decrease in amplitude is observed for an increase in the stiffness/damping factors. Since the decrease is approximately logarithmic, the present change varies inversely with an increase in the factor. The percent change is less than one for both stiffness and damping factors of 10, which leads to a design goal of a holder with a first naturalfrequency of 306 Hz and a minimum stiffness value of 10 times the stiffness of the bar.

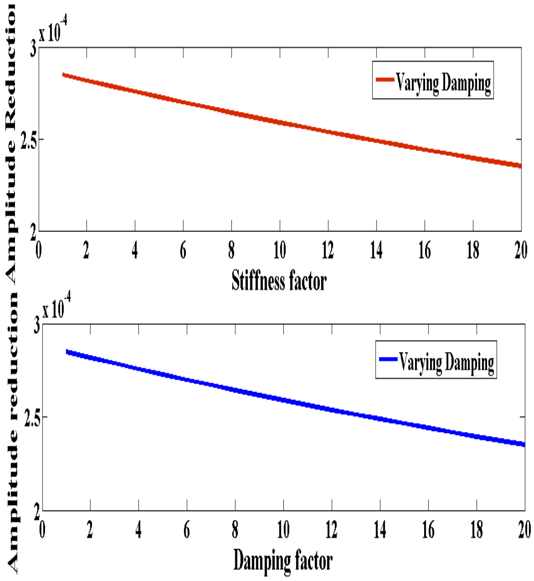

Fig. 3 Peak ratio for stiffness (top) and damping (bottom) factors

3.3 Sensitivity study

The sensitivity of the reduction in amplitude for mismatches in the bar and holder natural frequencies were determined analytically. In the analytical models the natural frequency of the bar and the holder were assumed to be equal. Since its difficult to produce a holder whose natural frequency matches with the boring bar's natural frequency.

The sensitivity analysis was done in order to provide an initial assessment of the feasibility and accuracy required for prototype manufacturing. For this analysis, the natural frequency for the holder modeled analytically in Fig. 4 (with a stiffness of 100 times the bar and the same damping value) was varied between 70% and 130% of the boring bar's fixed-free natural frequency (306 Hz). The ranges of natural frequencies are 214.2 Hz to 397.8 Hz in steps of 5.75 Hz. The heavy solid lines represent the nominal natural frequencies. Each holder was analytically coupled with the boring bar.

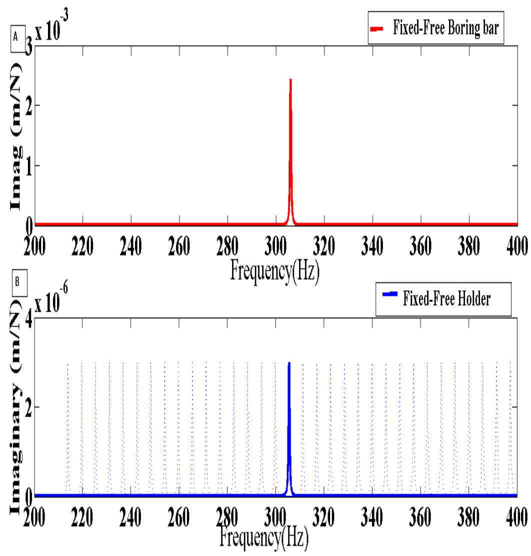

Fig. 4 Sensitivity study

(A) Boring bar dynamics.
(B) Various holder dynamics. The heavy solid lines represent Thenominal natural frequencies, while the dotted lines represent the various non-ideal holder responses.

3.4 Frequency Response Function for Boring bar holder assembly &clamped free boring bar.

Using the peak picking method [2], a stiffness value of $6.35x10^5$ N/m and a damping ratio of $6.5x10^{-4}$ were determined for the boring bar. A single degree of freedom (SDOF) representation of the cantilever holder was then defined with a stiffness value 20 times greater than the boring bar, but the same natural frequency and damping ratio. Next, the holder model wascoupled to a free-free model of the boring bar using the receptance coupling approach [3-8].

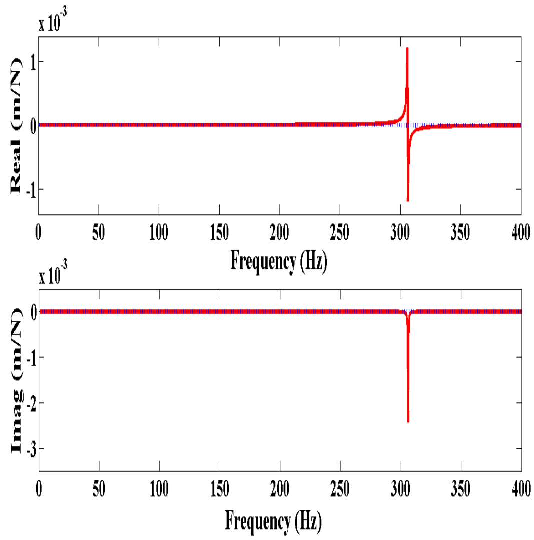

Fig. 5 Boring bar holder assembly (solid line)Clamped free boring bar (dotted line)

The FRF of the combined holder and boring bar is shown in Fig. 5. A 68% reduction is amplitude is observed for the holder assembly as compared to clamped free boring bar.

IV. RESULTS

The above simulation revealed the following.
Vibration amplitude of boring bar with different frequency and for different condition (clamped free and holder assembly) can be predicted using MATLAB.

[1] As shown in fig. 3 decrease in vibration amplitude is observed with increase in the stiffness/damping factors.
[2] The change in vibration amplitude is less thanone for both stiffness and damping factors of10. This leads to a design goal of a holder witha first naturalfrequency of 306 Hz and a minimumstiffness value of 10 times the stiffness of the bar.
[3] The sensitivity analysis,the natural frequency for the holder modeled analytically in Fig. 4 with a stiffness of 100 times the boring bar andthe dampingvalue same. The natural frequency for the holder modeled analytically was varied between 70% and 130% of the boring bar's fixed-freenaturalfrequency (306 Hz). Theindividual cases are shown in Fig.4, where eachdotted line represents a different holder.The ranges of natural frequencies are 214.2 Hz to 397.8 Hz in steps of 5.75 Hz.
[4] As shown in fig. 5 the dynamic stiffness of the Holder-boring bar assembly is higher than thestiffness of the cantilever boring bar alone, the stiffness improvements up to 68% are observed for the holder-boring bar assembly.

V. CONCLUSION

This paper described a flexible tool holder which acts as a dynamic absorber for a boring bar. By introducing flexibility into the holder (using notched flexure geometry) and matching its fundamental natural frequency to the first cantilever natural frequency of the boring bar, the holder effectively served as a dynamic absorber for the boring bar. An analytical approach was used to select the nominal holder response for an ISO A10-SCLPR2 NE4 boring bar with a 6:1 length to diameter ratio (15.9 mm diameter). The dynamic stiffness of the holder-boring bar assembly was compared to the stiffness of the cantilever boring bar alone; Stiffness improvements up to 68% were observed for the holder-boring bar assembly.

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