TUNABLE HOLDER FOR BORING BARS

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INTRODUCTION

During metal cutting operations, vibratory motion between a cutting tool and work piece can lead to non-beneficial cutting performance. 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 workpiece 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, because the holes tend to be deep and narrow, the 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.

Several methods for reducing boring bar vibration are currently used including, for example, internal vibration absorbers. 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 an analytical solution, which applies Euler-Bernoulli beam theory and receptance coupling techniques, and a finite element model for the assembly dynamics. 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.

RECEPTANCE COUPLING MODEL

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 for commercially available "tunable" boring bars (with dynamic absorbers located inside the bar). Figure 1 shows the analytical frequency response function (FRF) for the fixed-free (cantilever) beam model representing our steel, cylindrical boring bar.

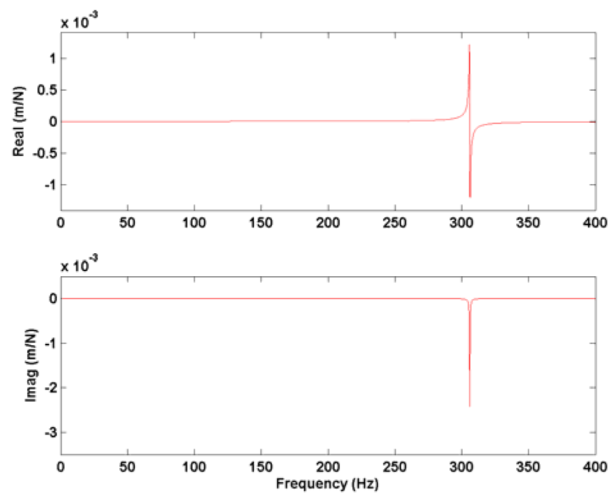

Figure 1: Cantilever response of 6:1 boring bar.

Using the peak picking method [2], a stiffness value of 6.35×10^5 N/m and a damping ratio of 6.5×10^{-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 was coupled to a free-free model of the boring bar using the receptance coupling approach [3-8]. The FRF of the combined holder and boring bar is shown in Fig. 2.

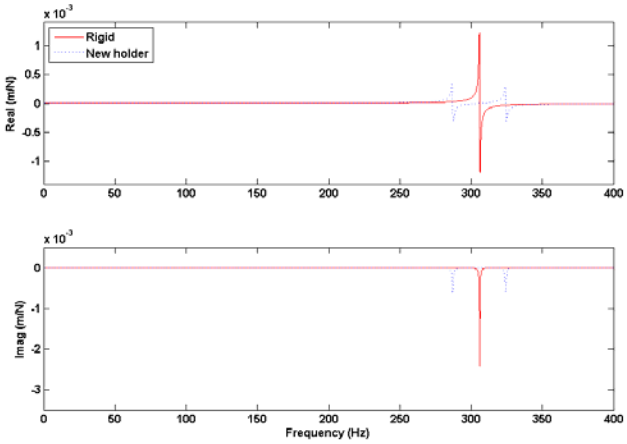

Figure 2: FRFs for the boring bar rigidly coupled to the holder (dotted line) and cantilever boring bar alone (solid line).

Figure 2 shows that the first bending mode of the boring bar is shifted down from 306 Hz to 287 Hz with a 74% reduction in amplitude. A second mode is also observed, but shifted up by the same amount to 324 Hz with the same amplitude. In the bandwidth shown, the system transforms from one bending mode (SDOF) to two bending modes (2DOF), split about the cantilever boring bar natural frequency, due to the addition of the holder. The increased dynamic stiffness of the coupled system is counter-intuitive (i.e., adding a flexible element increases the assembly stiffness), but is similar in nature to the well-known dynamic absorber effect.

FINITE ELEMENT MODEL

Finite element (FE) modeling was used to aid in the design and analysis of the holder. This enabled the convenient analysis of complicated geometries, which is more challenging when applying the analytical beam models that assume constant cross-sections [1]. All models were drawn using Autodesk Inventor 10 and imported into ANSYS Workbench 10 for analysis. Workbench subroutines were used to auto-generate part meshes; mesh refinement was introduced around critical areas. Solid three-dimensional (3D) elements were used and no cross section approximation or beam simplification was applied. Boundary conditions were implemented as appropriate. Each model was then checked for convergence and a final simulation was completed.

The ISO A10-SCLPR2 NE4 boring bar was modeled using cantilever boundary conditions and a harmonic analysis was carried out to identify its dynamic behavior. Figure 3 shows the results of the ANSYS model.

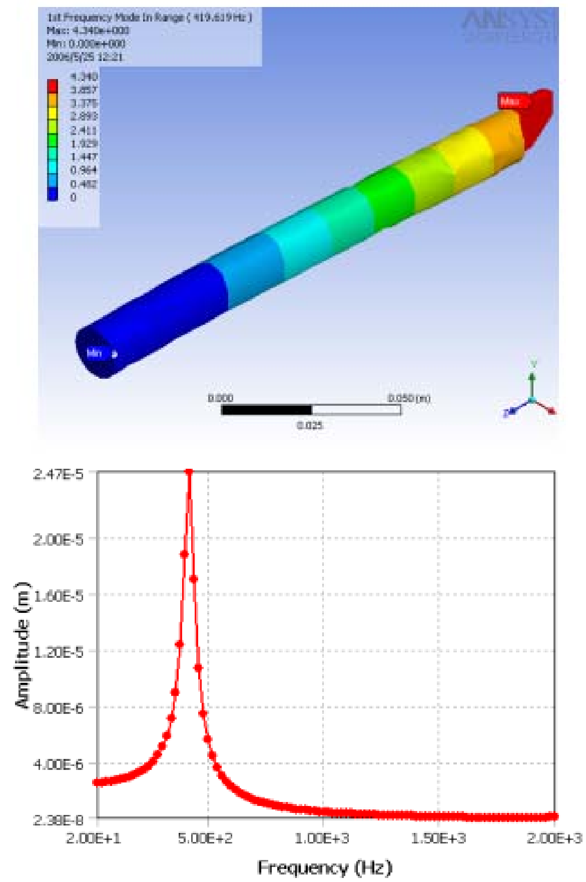

Figure 3: FE model of cantilever boring bar.

A notch-type flexure holder was selected to approximate a SDOF response. The notch and holder geometry were chosen to match the holder fundamental natural frequency to the FE boring bar results. Because high stiffness was required for the holder, the corresponding mass had to be large to realize the low natural frequency (419 Hz). A brass sleeve was clamped around the aluminum holder to obtain the desired response. Next, the boring bar was rigidly connected to the holder and the cantilever assembly FRF was computed; see Fig. 4. A 74% amplitude reduction relative to the boring bar alone (Fig. 3) is observed.

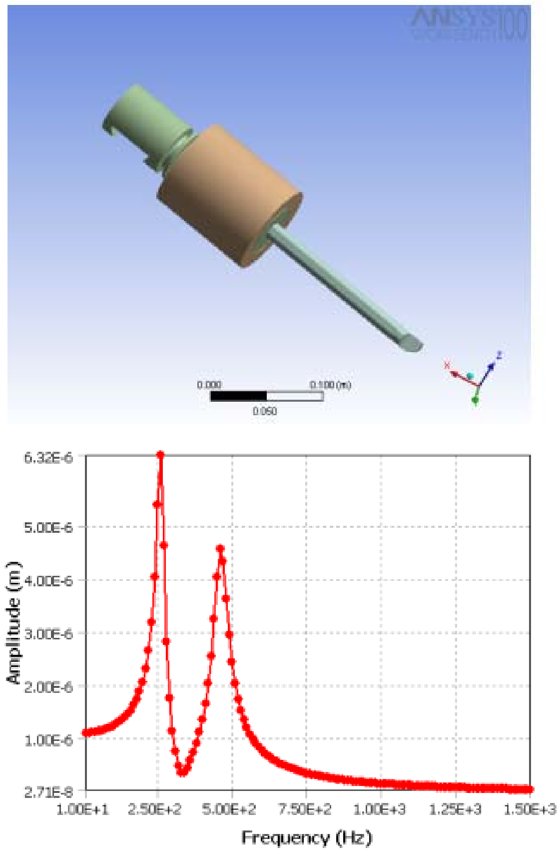

Figure 4: FE model of cantilever holder-boring bar assembly.

EXPERIMENTAL RESULTS

The 6061-T6 aluminum holder and brass sleeve were machined and two experimental setups were compared. In the first setup, a vise mounted to a machining center table was used to approximate a "grounded" boundary condition. In the second setup, the holder was clamped to a traditional dovetail tool post, again attached to the same machine table. The final holder and brass sleeve geometries are shown in Fig. 5. The outer diameter is the holder is 53 mm.

<u>Vise setup</u>

Before the boring bar was inserted into the holder, its cantilever FRF was recorded by clamping the boring bar directly in the vise at the 6:1 L:D ratio. The boring bar was then inserted in the holder using a thermal shrink fit connection. The assembly was also clamped in the vise and the FRF was measured at the free end of the boring bar. Figure 6 shows a comparison of the cantilever boring bar and assembly FRFs. A 53% reduction is amplitude is observed for the assembly.

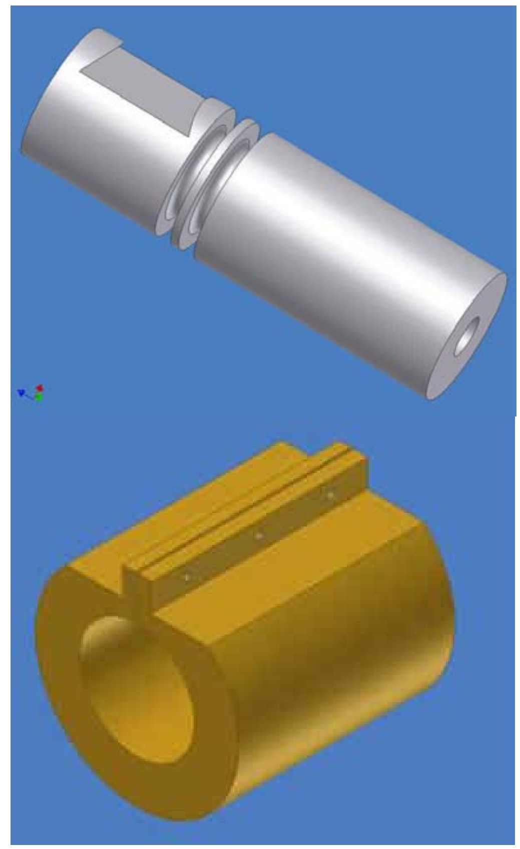

Figure 5: Geometry of holder and brass sleeve. The holder outer diameter is 53 mm.

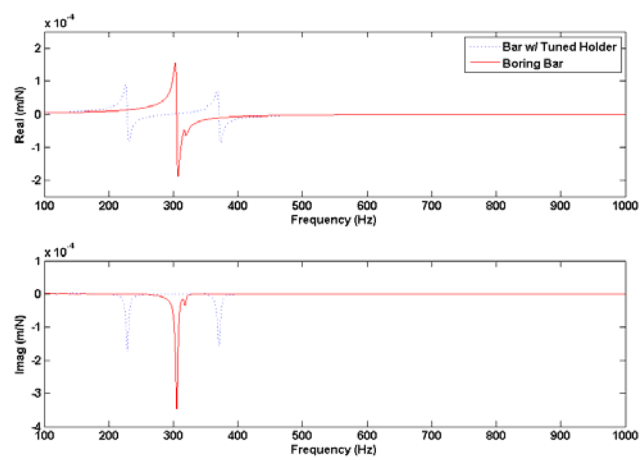

Figure 6: Comparison of cantilever boring bar (solid line) and holder-boring bar assembly (dotted line) FRFs. The assembly is 53% stiffer.

Dovetail setup

Finally, the assembly was attached to a standard dovetail tool post (see Fig. 7). The direct FRF at the free end of the boring bar was again measured. The results are shown in Fig. 8. Note that the scale for Fig. 8 is the same as for Fig. 6. It is seen that the response is slightly stiffer (68% amplitude reduction over the cantilever boring bar) and additional modes are present. This is presumed to be due to additional damping in the non-rigid dovetail connection.

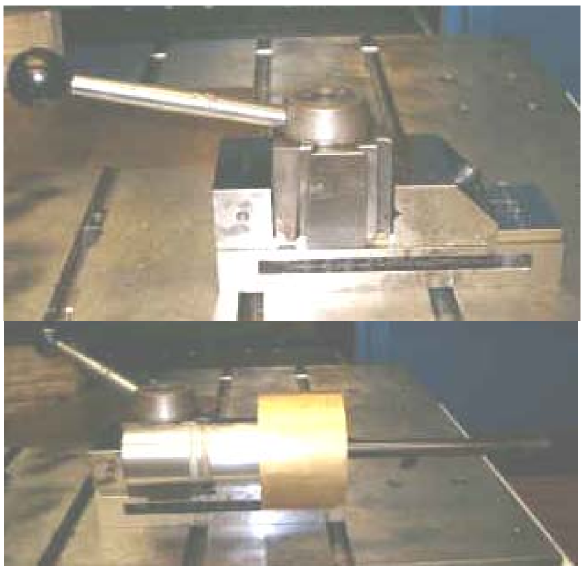

Figure 7: (Top) Photograph of dovetail tool post. (Bottom) Photograph of holder-boring bar assembly clamped to dovetail tool post.

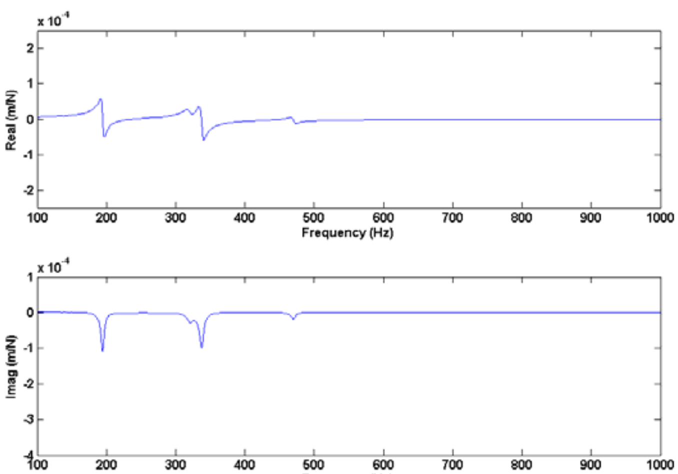

Figure 8: FRF of holder-boring bar assembly clamped to dovetail tool post.

CONCLUSIONS

This paper described a flexible tool holder which acts as a dynamic absorber for a boring bar. By introducing flexibility into the holder (using a 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). Finite element models were used to refine the design. Components were machined and frequency response function measurements were completed. 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.

ACKNOWLEGEMENTS

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