Dynamic Vibration Absorber
For use in cantilever tooling structures

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Problem Statement

- In many high precision machining applications vibration is a concern

- Most boring bars for internal boring are not able to obtain a length to diameter ratio greater than 4-5 without noticeable vibration

- The project goal is to design an internal dynamic vibration absorber that will allow greater length to diameter ratios
State of the Art

- Many concepts have been designed and patented to reduce the vibration as the L/D ratio increases.

- Most researched designs consist of standard vibration absorbers as opposed to dynamic vibration absorbers.

- A summary of past concepts/designs is as follows:
Adjustable damping device in particular for boring bars and the like

On either end of the damping member there are two annular springs that can be adjusted in order to vary compression allowing different degrees damping for individual applications.
Enhancement of dynamic stability of cantilever tooling structures

- Machinable tungsten insert attached with rubber bushings inserts and tuning screws, as well as multiple components allows for improved L/D ratios
Problems with current designs

- The aforementioned designs do not allow larger L/D ratios due to low effective mass.

- Attempting to increase the effective mass of the system creates thin walls in the boring bar lowering the strength of the bar.

- Also, with the rubber o-rings the degrees of freedom the system greatly increase the complexity of the system allowing only translational motion in multiple directions.
Project Direction

- Create an internal system that translates linear motion to rotational motion in order to increase effective mass, thus reducing vibration

  - Relevant Methods:
    - Energy method for evaluating vibrating systems
    - Actual vs. Effective mass
When a system has more than one moving part, an effective mass can be determined if the motion of every point of the system is known. The effective mass of the system is the equivalent lumped mass at a specified point.

Transformation of linear motion to rotational motion allows the effective mass to be significantly larger than the actual mass.

A larger effective mass is important because it lowers the natural frequency of the system which can be seen in the following equation.

\[ \omega_n = \sqrt{\frac{k}{m_{\text{eff}}}} \]
Proposed design A

• A wire wrapped around a cylinder of radius, R, and mass, m, in order to induce rotational motion causing a rotation angle of $\alpha = \frac{x}{R}$

• A torsional spring element attached to the end of the mass gives us the required stiffness, $k$, given by $k = \frac{T}{\alpha}$
Proposed design A
Proposed design B

- A wire wrapped around a smaller machined radius of \( r \), at each end of the cylinder to increase rotational angle \( \alpha \), given by \( \alpha = \frac{x}{r} \)

- A torsional spring element attached to the end of the mass gives us the required stiffness, \( k \), given by \( k = \frac{T}{\alpha} \)
Proposed design B
Justification of Design B

- Angle of rotation is given by $\alpha = \frac{x}{r}$. The proposed design allows $r$ to decrease, which increases alpha, maintaining the mass of the damper while lowering the natural frequency of the system.
Prototype Design

- We decided to build the prototype out of PVC and metal piping (as can be seen)

Parts used:

1. PVC pipe (boring bar shaft)
2. Black steel pipe (damping mass)
3. Guitar string (support wire)
4. Rubber chord (torsional spring element)
Energy Method

• In order to find the effective mass of the system, the energy method must be applied to the forces acting on the system

• The energy method states that if the total energy of a system is constant, it’s rate of change is zero

• This can be seen by the equation:

\[
\frac{d}{dt} (T + U) = 0
\]
Energy Method

- If the system is undergoing harmonic motion, the equation becomes:

\[ T_{\text{max}} = U_{\text{max}} \]

- From the above equation, the natural frequency of the system is readily available.
Energy Method

• The maximum kinetic energy of Design B is given by:

\[
T_{\text{max}} = \left[ \frac{1}{2} J \dot{\theta} + \frac{1}{2} m (r \dot{\theta})^2 \right]_{\text{max}}
\]

• While the maximum kinetic energy of design B is given by:

\[
U_{\text{max}} = \frac{1}{2} k (r_2 \theta)^2_{\text{max}}
\]
Energy Method

• Equating the two energy equations to find the natural frequency of the system gives us:

\[ \omega_n = \sqrt{\frac{k_t}{\frac{J}{R^2} + m \frac{r^2}{R^2}}} \]

Where:

\[ \alpha_r = \frac{x}{r}, \quad \alpha_R = \frac{x}{R}, \quad \text{and} \quad k_t = \frac{T}{\alpha} \]
The increased angle of rotation and the decreased radius at the cylinder ends results in an increased effective mass.

The increased effective mass then directly results in a decreased natural frequency of the system.
Testing Methods

- Rigidly clamp one end of the “boring bar”
- Attach accelerometer to opposite end
- Strike prototype with hammer at end with accelerometer
- Record measurement frequency

* Testing will be done on final prototype assembly and similar design (without DVA) to prove prototype design allows increased L/D ratios while reducing chatter.
The next step in the improved Design B is to measure the deflection of the spring element insert so as to calculate the torsional stiffness of the material.

Upon completion of material measurements, calculations and testing must be done in order to find suitable values for the variables (m, r, R) that are present in the equation for natural frequency.

Determine whether offsetting the smaller ends of the cylinder from the larger radius center will increase the effective mass of the damper.
Timeline

- Oct. 18 - 28: Finalize calculations
- Oct. 28: Detailed drawings
- Oct. 28 – Nov. 12: Final prototype build
- Nov. 12 – Dec. 2: Testing & modification
Questions

- Anyone.......anyone........good