

REDUCED BORING BAR VIBRATIONS USING DAMPED COMPOSITE STRUCTURES

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ABSTRACT

A new technology, called Stress Coupling Activated Damping (SCAD©), was applied successfully to a lathe boring bar. It reduced high frequency vibrations by up to 20 db. It can be applied to a wide range of structural designs. The geometries of the damped structures are not limited to thin plates but can be applied to tubes, I-beams, and complex structures. This allows SCAD© technology to be applied to several industry design problems, including the metrology, medical, aerospace, automotive and machine tool industries. SCAD© will also allow boring bars to: 1) be optimized for stiffness, frequency and loss factor, 2) be 'tuned' to a specific resonant frequency, 3) have improved damping regardless of boring bar diameter, 4) lengthen the usable tool length without a significant increase in cost, and 5) decrease cutting surface vibration induced pattern.

1.0 INTRODUCTION

Vibrations caused by engine noise, rotating parts, and air turbulence affect airplanes, machinery, and other structures. With the advent of lightweight materials, uncontrolled vibrations have become a common, yet significant, problem. For example, the Lockheed C-141B transport airplane had maximum cargo weight conditions lowered because of cracks in the wings. These cracks occurred because of vibrations caused by the turbulent boundary layer. Uncontrolled vibrations cause fatigue damage and structural failure at lower stresses. It is of critical importance to control vibrations in sensitive components. Vibrations also cause noise in sensitive electronic equipment found in space structures and airplanes.

Reducing vibrations can be accomplished by increasing the stiffness of the structural component or by reducing the source of

the vibrations. Unfortunately, stiffening structures tends to shift the damaging resonant frequencies to other structural components, and reducing vibrations at the source is usually not a viable alternative.

There are two possible damping methods for controlling vibrations: passive and active damping. Passive damping can be accomplished relatively inexpensively. Part geometry and material changes are used to reduce vibrations. This is done by converting kinetic energy (movement) to thermal energy. Examples of passive damping methods range from gas shock absorbers to padding on aluminum baseball bats. Active damping is expensive. It requires an outside power source and, in many cases, a feedback control system. Piezoelectric actuators and feedback sensors are often used in active damping applications.

Currently, passive and active vibrational damping methods have limited usefulness. Viscoelastic materials used in passive damping methods dissipate less than ideal amounts of thermal energy. The need for a power source, force transducers, and feedback controls makes active damping methods difficult and expensive to implement.

Fiber reinforced composites have an advantage over conventional materials in many applications because of their favorable strength to weight ratios, corrosion resistance, and unique stress coupling properties. Composites have 10 to 100 times the damping of metals, but damping is still low, less than 1% usually. Viscoelastic materials, on the other hand, exhibit low strength to weight ratios, but can provide high levels of damping when significant shear forces are generated in the material. With an appropriate combination of both materials, it is possible to create a structure that is lightweight, stiff, and highly damped.

2.0 RELATED RESEARCH

One of the more common passive damping technologies is called Constrained Layer Damping (CLD) [Kerwin, 1959]. CLD is achieved by bonding a thin constrained layer of a metal sheet, usually aluminum, to an existing structure with a viscoelastic adhesive. Internal friction in the form of hysteresis losses in the viscoelastic material generates thermal energy, thus reducing the vibrational energy through the dissipation of heat. There are three major disadvantages of CLD:

- it adds weight and bulk
- it may only be applied to the surface of the structure
- it is only effective for out-of-plane vibrations

Barrett used the inherent shear coupling properties of composite materials to design a damped composite tubular component [Barrett 1989]. Shear displacements occur naturally when fiber reinforced composites are loaded in tension or compression along any axis not parallel or perpendicular to the fibers. Constructing a plate with a layer of positive fiber angle orientation, a viscoelastic material layer, and a layer of negative fiber angle orientation will generate large shear strains, but only at the ends of the plate, when an axial load is applied. These strains are caused by shear coupling, which comes from the coupling of shear loads and normal loads.

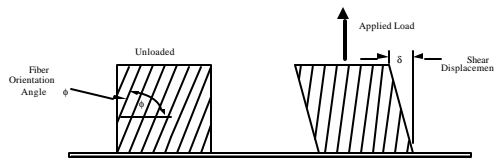


FIGURE 1. SHEAR DISPLACEMENTS OF FIBER-REINFORCED COMPOSITES.

Axial loads applied to tubular elements will cause the off-axis angles of a composite layer to rotate (see Figure 1). Barrett consequently designed a tube and combined the concepts of constrained layer damping with shear coupling to create a stiff, highly damped tube [Barrett, 1989]. The tube consisted of an inner positive angle composite layer, a viscoelastic damping layer, and an outer negative angle composite layer. Barrett's tube maintained the high stiffness because the axial loads traveled through the composite material and high damping because shear strains were produced across the viscoelastic layer. But, the tubes experienced shearing only at the ends of the tubes. Clamping the tube at the ends eliminated the damping effect, reducing this design to a conventional composite tube.

In 1992, Olcott developed and reported the findings of a new damping concept called Stress Coupling Activated Damping (SCAD©) at Brigham Young University [Olcott, 1992]. SCAD© uses the stress coupling effect of anisotropic materials, such as fiber reinforced composites, to distribute damping more uniformly

through the volume of embedded viscoelastic layers. The fiber orientation angle in each stiffness layer is alternated several times along the length of the component. Each time the fiber orientation angle is altered, a region of high shear is generated across the damping layer (see figure 2). By controlling the orientation angle, thickness, segment lengths and moduli, significant shearing occurs throughout the viscoelastic layer. Also, since the primary load path through the part is in the composite stiffness layers, the part retains high stiffness. Even clamping the ends of the structure will only eliminate shearing at the ends, thus resulting in insignificant reductions of the overall damping effect. SCAD© is also applicable to several geometries, rather than just tubes like Barrett's design, since the edges of the component may also be rigidly clamped without reductions in damping [Olcott et al., 1992].

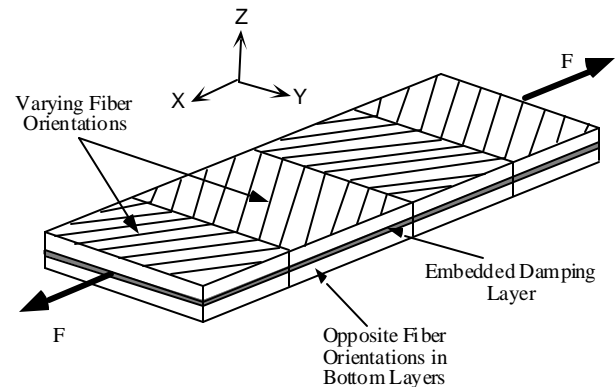


FIGURE 2. OLCOTT'S DAMPING CONCEPT.

For this research Olcott's model was chosen as a basis. Following will be a brief outline of his analytic model.

3.0 ANALYTIC MODELING

An analytical model was developed to analyze these passive damping composite structures. The model, developed by Olcott, determines the loss factor and effective modulus of components.

3.1 SCAD© Analytic Model

Olcott's model, which determines the loss factor and the effective modulus of components and overall structures, uses several assumptions for analysis [Olcott, 1992], summarized as follows:

- The stiffness layers are assumed to be thin.
- The membrane may only be loaded axially and in transverse shear.
- Displacements are not a function of y , therefore $du/dy = dv/dy = 0$.

- The damping layer moduli are much lower than the stiffness layer moduli.
- The only out-of-plane stresses applied to the stiffness layers by the damping layers are the shear stresses τ_{xz} and τ_{yz} , acting throughout the volume of the viscoelastic layer.
- There is no slip at the interface between a stiffness layer and a damping layer.
- Displacements in the z-direction are negligible.

The general orthotropic stress-strain relationships applicable to composite stiffness layers represented in matrix notation are:

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & Q_{16} \\ Q_{21} & Q_{22} & Q_{26} \\ Q_{61} & Q_{62} & Q_{66} \end{bmatrix} \begin{bmatrix} \epsilon_{xx} \\ \epsilon_{yy} \\ \gamma_{xy} \end{bmatrix}$$

The constants Q_{11} through Q_{66} are functions of the complex axial modulus, transverse modulus, poisson ratio, shear modulus, and the fiber orientation angle. From the analysis assumptions, only σ_{xx} and τ_{xy} stresses may be applied to the stiffness layers, implying that $\sigma_{yy} = 0$. The strain in the y-direction, however, is not zero due to poisson effects. Under these conditions, the above equation may be reduced to a 2x2 matrix:

$$\begin{bmatrix} \sigma_{xx} \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} K_{11} & K_{16} \\ K_{61} & K_{66} \end{bmatrix} \begin{bmatrix} \epsilon_{xx} \\ \gamma_{xy} \end{bmatrix}$$

Olcott then develops equations for a single stiffness layer with stress components that affect equilibrium; for example, the effects of adjacent viscoelastic layers as coupled shear stresses on the faces of the stiffness layers. The governing equations for each stiffness layer are then combined into a matrix form and non dimensionalized:

$$\begin{bmatrix} \hat{K}_{11} & \hat{K}_{16} \\ \hat{K}_{16} & \hat{K}_{66} \end{bmatrix} \begin{bmatrix} \frac{\hat{z}^2 \hat{u}}{\hat{z}^2} \\ \frac{\hat{z}^2 \hat{v}}{\hat{z}^2} \end{bmatrix} + D_o \begin{bmatrix} 1 & 0 & -2 & 0 & 1 & 0 \\ 0 & 1 & 0 & -2 & 0 & 1 \end{bmatrix} \begin{bmatrix} \hat{u}_{j-1} \\ \hat{v}_{j-1} \\ \hat{u}_j \\ \hat{v}_j \\ \hat{u}_{j+1} \\ \hat{v}_{j+1} \end{bmatrix} = 0$$

where:

$$\hat{K}_{pq} = \frac{K_{pq}}{E_{11}} \quad \text{a function of the complex modulus } E_{11}, \text{ fiber angle orientation and poisson ratio for the stiffness layer}$$

$$\hat{u} = \frac{u}{L_s} \quad \text{displacements in the axial direction divided by the segment length}$$

$$\hat{v} = \frac{v}{L_s} \quad \text{displacements in the transverse direction divided by the segment length}$$

$$D_o = \frac{G_d L_s^2}{E_{11} t_s t_d} \quad \text{a function of the damping layer and its thickness; and the adjacent stiffness layer and its thickness}$$

The solution to the equation is of the form:

$$\begin{bmatrix} \hat{u}_1 \\ \hat{v}_1 \\ \dots \\ \hat{u}_n \\ \hat{v}_n \end{bmatrix} = \sum_{j=1}^n \begin{bmatrix} \hat{u}_1 \\ \hat{v}_1 \\ \dots \\ \hat{u}_n \\ \hat{v}_n \end{bmatrix} \left[\hat{c}_{+j} e^{(\lambda_j) \hat{x}} + \hat{c}_{-j} e^{(-\lambda_j) \hat{x}} \right]$$

The constants \hat{c}_{+j} and \hat{c}_{-j} are found by applying the appropriate load and displacement boundary conditions of the damped component being modeled. Particular analysis elements may be found which allow a small section of the whole component to be efficiently modeled, which will then yield the solution to the entire original component.

4.0 APPLICATION OF TECHNOLOGY

SCAD© technology can be applied to a wide range of designs. The geometry of the structure is not limited to just thin plates but rather to several different geometries, including, but not limited to: tubes, plates, and I-beams. This allows SCAD© technology to be applied to several industry design problems, including: metrology, medical, aerospace, automotive, machine tool industry, and others.

Machine tools (i.e. laths, mills, grinders, robots, etc.) have traditionally suffered from vibrations introduced by control parameters (i.e., feed rates, spindle speeds, cutter or end-effector, etc.), motors, gears, shafts, etc. SCAD© technology can be applied to many of the structures and components of a machine tool. For example, the drive shafts in these machine tools could be constructed completely from damped composites. In the event that strength is a problem, this technology can also be applied to the shafts as an over wrap. It is anticipated that SCAD© technology will revolutionize the construction of such machine tool components as bearing races, ways, column, carriage, etc.

The most logical system to isolate on existing machine tools is the cutting tool itself. When chatter (vibration) is introduced to the cutter, there is an immediate and visible effect on the surface being machined. If unmonitored and uncorrected, the vibrations of a turning tool will pass from the tool to the tool post, through the

carriage and ways, potentially causing the head-stock and work piece to resonate, characterized by violent vibration. If, however, SCAD© were applied to the tool and it were tuned so that its first mode were significantly shifted and any induced vibrations were dissipated within its structure, this machine tool resonance would be avoided.

It is also expected that this technology will be applied successfully to the automotive, aerospace, and ship building industries. In the auto industry, frames and body panels could be constructed from damped composites. In airplanes, the stringers and air-frames components could likewise be constructed using these methods. SCAD© structures could also be used in ship building to isolate the vibrations induced by the motors and power-train systems from the hull and other structures of the ship.

Not only is SCAD© technology applicable to these industries but also to the sound and metrology industries. In the sound industry, SCAD© technology will help in the reduction of sounds from vibrating structures, panels, etc. Reducing vibrations on high precision measurement machines means that the accuracy of measurements may be significantly increased as well.

4.1 Experimentation

As an example of the technology applied to the machine tool industry, a boring bar was used to test vibration reduction. A one inch boring bar turned down to one-half inch diameter was over wrapped with the SCAD© technology composite and tested against an unmodified boring bar. The experimental variables are shown in Table 1.

TABLE 1. EXPERIMENTAL VARIABLES USED

Boring Bar	1" dia carbon steel
Diameter of steel core	0.5"
Segment Length	1.1"
Number of segments	6
Outer Diameter of wrap	1"
Stiffness Material	graphite/epoxy prepreg
E_1	17.3 Msi
E_2	1.25 Msi
G_{12}	0.85 Msi
ρ	0.058 lbm/in ³
t_s	0.0055"
Fiber Orientation	$\pm 26^\circ$
Viscoelastic Material	3M ISD 112 at 0.005"
η	90%
Joint Overlap	12.5%

The experimental variables were chosen using Olcott's analysis model for single damping layer wraps. The frequency of the original boring bar was found and an assumption was made that

the stiffness of a boring bar with a composite wrap would not significantly change. In order to see general trends, it was determined that multiple viscoelastic layers should be incorporated into the design. Olcott's model was still used even though slight differences would be found between the designed boring bar's resonant frequency and the predicted value. As will be shown, this assumption was correct.

4.2 Lay-Up

The over wrap consisted of the following several steps:

1. A zigzag, or segmented layer prepreg, was hand laid up (see Figure 3). Each layer consisted of six 1.1" x 20" segments with $\pm 26^\circ$ fiber plies. The joints were designed and analyzed to determine the best type of joint [Lee, 1993]. A 12.5% overlap was determined to be optimum.

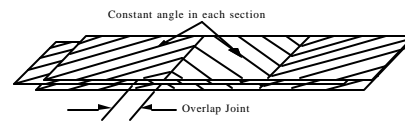


FIGURE 3. ZIGZAG PREPREG HAND LAY-UP.

2. The viscoelastic material was applied to the first prepreg layer. Rotz developed a method of co-curing embedded layers of damping material in composite structures [Rotz, 1991]. The co-curing process theoretically improves the bond between the damping material and the composite layers, as long as during curing the damping material properties are not degraded by high curing temperatures.

3. The zigzag prepreg of the opposite orientation was hand laid up.

4. The three layers were then rolled onto the turned down boring bar using two heated and padded plates to consolidate the plies as the tube was rolled (see Figure 4).

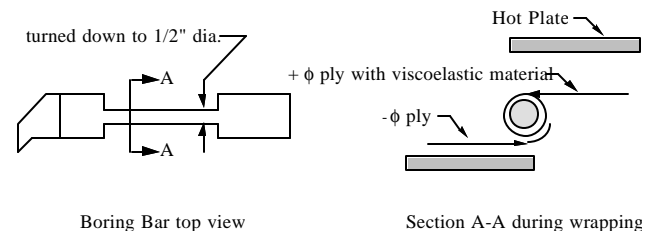


FIGURE 4. CROSS-SECTION OF BORING BAR AND WRAPPING SCHEMATIC.

5. The boring bar was over wrapped for consolidation and then suspended vertically in an oven and cured for four hours at 250°F.

4.3 Data Acquisition Setup

The boring bars were tested in transverse vibration at Brigham Young University. Equipment used included a structural dynamics analyzer, an accelerometer, a force hammer, and power/signal conditioners.

The tubes were suspended at each end by soft rubber bands to approximate free-free boundary conditions. Transverse motion of the tube was monitored by attaching the accelerometer to one end, then striking the opposite end with a force hammer (see Figure 5). The force and acceleration signals were fed into the structural analyzer, which calculated the transfer function. The tube was struck ten times and the data averaged by the analyzer.

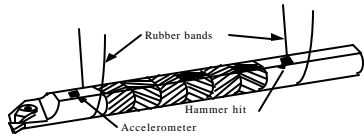


FIGURE 5. DATA ACQUISITION SETUP.

5.0 RESULTS

During testing, the volume of the tool was essentially the same, whether the bar was steel, or over wrapped. With this design, the dynamic stiffness was also kept relatively constant between the two bars, since the resonant frequency is nearly the same for both specimen (unmodified boring bar $\omega_n = 1.2$ MHz over wrapped boring bar $\omega_n = 1.0$ MHz (see Figure 6)). The difference in resonant frequencies results from designing for a single damping layer but using multiple damping layers.

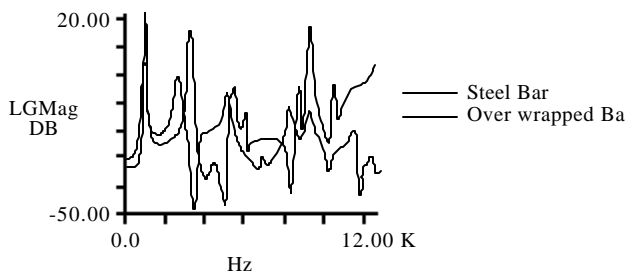


FIGURE 6. FREQUENCY RESPONSE DIAGRAM OF CARBON STEEL AND OVER WRAPPED BORING BAR.

The frequency response diagram (Figure 6) shows that the over wrapped bar exhibits a 5 decibel drop at low frequencies and a drop of 20 decibels at higher frequencies. Thus, vibrations, especially at higher frequencies, are reduced while maintaining the

structure's stiffness and diameter. As a result, better surface finish will be attained when using the over wrapped boring bar.

The boring bar may also be designed to resonate at a different frequency making it a tunable structure. The boring bar may possibly be designed to either resonate far below operating limits, or higher than the operating limits to reduce vibrations of the tool. This is done by selection of viscoelastic material properties, segment length, thickness of the damping and or stiffness layers and segment orientation.

It is expected that an optimally designed boring bar (for both damping and resonant frequency) will exhibit the same performance trends but with significantly higher damping. The expected benefits include:

- Primary mode resonance at higher frequencies with damping greater than 15%
- Customizable frequency response for special applications
- Improved length to diameter ratios
- Increased resistance to chatter

This technique may also be applied to boring bars of any diameter. This is a major advantage over weighted boring bars which have a limited minimum diameter to reduce tool vibrations.

6.0 CONCLUSIONS & RECOMMENDATIONS

It has been shown that a boring bar with a SCAD® composite over wrap has several advantages over a steel boring bar. The advantages are as follows:

- Stiffness, frequency, and loss factor are optimized
- The boring bar resonant and damping frequency are tunable
- This technique is not limited by the boring bar's diameter
- An increase in usable tool length for little cost
- There is decreased machine tool vibration

These advantages can also be applied to other tooling and other industries to reduce vibrations. Models were proven correct and can provide predictive analysis for designers.

Currently, Olcott's analysis equations are being developed for a more general case. This case would include the possibility of several alternate damping and stiffness layers and the application of different types of viscoelastic material, either symmetrically or non symmetrically to the structure. In this manner, a designer may better tune the structure in order to maximize the damping and stiffness in a specific frequency range by varying the structure variables. Multiple damping layers will give a higher potential for energy absorption; thus more damping may be attained while still maintaining a stiffness design limit.

Research at Brigham Young University focuses on improving the performance, manufacturability, and extending the uses of SCAD® related technology and concepts. The following areas have been identified as follows:

- Refine models through statistically designed experiments
- Improve the optimization of multiple damping layers

- Improve bending equations for the prediction models
- Improve manufacturing methods for:
 - zig zag prepreg with specified angle orientation
 - wrapping procedure for existing boring bars
- Manufacturing and design of:
 - machine tool components, active control integration, aerospace structures, panels, marine structures, automotive components, metrology components, and noise control.

This research was initiated by Brigham Young University to determine the practicality of highly damped fiber composite technology to solve vibration problems in several industries. An initial experiment with a machine tool boring bar showed significant results. Improvements and further research on the SCAD© technology are currently being pursued.

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