LASER INTERFEROMETER GRAVITATIONAL WAVE OBSERVATORY

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Concept for a bounce & roll mode damper for the quad suspensions

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1 Introduction

The feasibility of using tuned mass dampers to passively damp the bounce and roll modes of the quad suspension is explored in Brett Shapiro's note LIGO-T1500271. That document contains calculations of the damping and thermal noise performance of a tuned mass damper applied to the tips of the UIM blade springs, as a function of several parameters of the damper. This document contains a concept for a specific design of a damper, based on the calculations of T1500271.

2 Target damped Q

We first need to decide how much we want to reduce the bounce and roll mode Qs. The current (undamped) Qs have been measured to be about 500,000. Typically the modes can get excited to levels that are 3 or 4 orders of magnitude above thermal excitation. Let's say we want the modes to damp down to thermal level in 100 seconds or less. Taking into account the fact that with a lower Q, the modes won't build up as much in the first place, the *e*-folding time tau should satisfy:

$$\log\left(10^4 \cdot \frac{Q_d}{500{,}000}\right)\tau < 100\text{ sec},$$

where Q_d is the damped Q. We also have $Q_d = \pi\tau f \approx 10 \cdot \pi\tau$, so the condition on Q_d is:

$$\log(Q_d/50) \cdot Q_d < \pi\times 10^3.$$

This is satisfied for $Q_d < 1000$. This assumes the worst case of 10^4 for the current factor above thermal excitation. For an excitation 1000x thermal, the condition would be $Q_d < 2000$. I conclude that the target damped Q should be in the range of one to several thousand.

3 Damper parameters

The main parameter that must be chosen is the damper mass. For a given damper mass, there is an ideal damping factor that yields the smallest bounce/roll mode Q_d, as calculated in T1500271. Larger damper mass will give a lower Q_d, but it will also increase thermal noise. A mass ratio of $\mu = 5\times 10^{-5}$, as defined in T1500271, is a good compromise between these two effects.

The ideal damping factor for this mass ratio is 0.01 (structural damping). With this damping factor, this damper mass will yield bounce and roll mode Qs of $Q_d = 200$ if the damper is perfectly tuned, $Q_d = 350$ if mistuned by 0.1%, and $Q_d = 2000$ if mistuned by 1%. It is not yet clear how closely tuned the dampers can be made in practice, but a 1% or smaller mistuning sounds feasible.

The longitudinal thermal noise from the damper (assuming cross-coupling as given in T1500271), at 20 Hz, is 5.5e-20 m/rtHz for structural damping, and 7e-20 m/rtHz if the damper is viscously damped. This is to be compared with the thermal noise from the silica suspension fiber, which is 1.6e-20 m/rtHz at 20 Hz. The thermal noise from the BRD is thus a factor of 2-3x smaller than the intrinsic suspension thermal noise at 20 Hz (and a greater factor at higher frequencies); this is an acceptable level of additional noise (suspension thermal noise is already a factor of several below quantum noise at 20 Hz).

The mass ratio of $\mu = 5\times 10^{-5}$ corresponds to bounce dampers of 1 gram on each blade spring, and roll dampers of 0.45 gram. For roll I'll use a slightly higher mass of 0.5 gram so that the damper spring constant can be the same for both dampers (see table below).

	Bounce	*Roll*
Frequency	9.8 Hz	13.9 Hz
Damper mass	1.0 gram	0.5 gram
Spring constant	3.8 N/m	3.8 N/m
Static sag (g/w_0^2)	2.5 mm	1.25 mm
Ideal damping factor -- structure	0.01	0.01
viscous	6e-4 N/m/s	4e-4 N/m/s

4 Damper design

The damper concept is a simple mass at the end of a cantilever design. The resonant frequency is given by:

$$2\pi \cdot f = \left[\frac{3EI}{L^3(m + 0.25m_b}\right]^{\frac{1}{2}}$$

where E is the Young's modulus of the cantilever blade, L is the length and I is the area moment of inertia of the cantilever blade, m_b is the mass of the blade, and m is the mass mounted at the end of the cantilever. The blade mass is a small correction (<1%), so I'll neglect it. Then we can just deal with the spring constant of the blade:

$$k_b = \frac{3EI}{L^3} = \frac{Ewt^3}{4L^3}$$

where t and w are the blade thickness and width, respectively.

Damper material. I will assume the entire damper is made of copper. Copper has a relatively high loss factor for metals, possibly as high as 0.007. It is also readily available, easy to machine, and easy to incorporate additional eddy-current damping. Some manganese-copper alloys may have higher damping (and could be looked into), but without some of these features.

Blade design. The Young's modulus of copper is 120 GPa. I'll use a blade width of w = 4 mm. This gives the ratio:

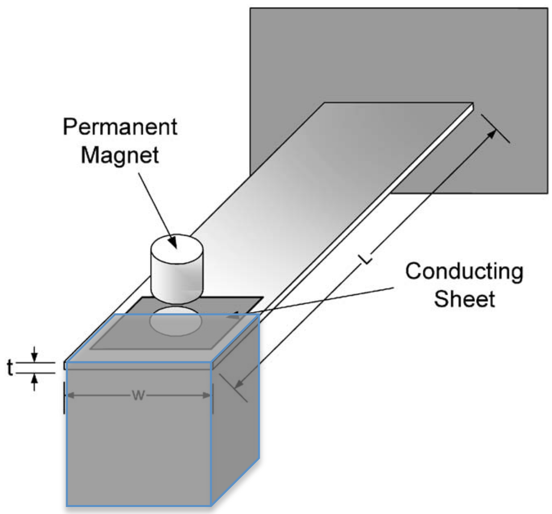
$$\frac{t}{L} = \left[\frac{4k_b}{Ew}\right]^{1/3} = \frac{1}{316}$$

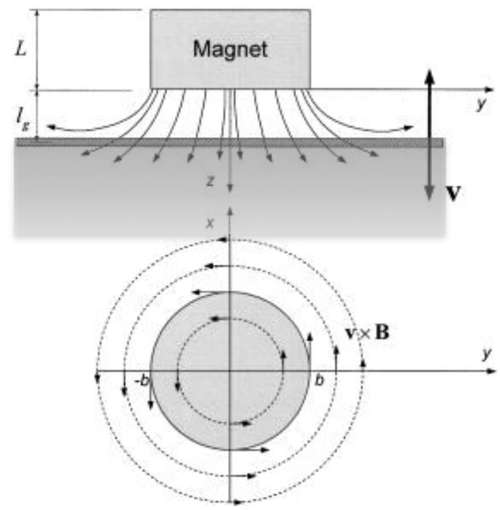
Using 2-mil 'shim stock' (51 micron thickness), the blade length would be L = 1.6 cm.

Masses. The masses can be cubic, or approximately cubic. Using copper (8.95 gm/cm^3), the bounce mode mass (1 gm) would be 4.8 mm on a side, and the roll mode mass (0.5 gm) would be 3.8 mm on a side.

Frequency tuning. To fine tune the resonant frequency of the damper, there are a few options: the mass can incorporate a tapped hole for a tuning screw; material can be removed (filed/sanded/machined) from the mass; material could be added to the mass (soldered on).

Damping factor tuning. If the damping factor of copper is not high enough, we can add a little eddy-current damping (ECD) with a small, nearby magnet. This is depicted below,.





We can make a rough calculation of the damping afforded by this geometry. The induced current, and thus the damping term, comes from the radial component of the B-field. The damping coefficient is

$$\frac{F}{v} = 2\pi\sigma \iint_{0,0}^{h,r} y B_y^2(y, l_g + z)\, dy\, dz = \sigma A_e h_e B_y^2$$

where σ is the conductivity of the mass (6e7 S/m), h is the thickness of the mass (z direction), and r_e is an effective radius of the copper mass (in the x-y plane, if it were a cylinder). In the second equality, h_e is an effective depth to represent the integration over z, and A_e is an effective area, such that the product $A_e B_y^2$ represents the integration in the x-y plane. Note that B_y is zero at x,y = (0,0), and is a maximum around the magnet radius or further, depending on the separation l_g.

Assuming the magnet is not very close to the damper mass, then to within a factor of 2, I expect that h_e = 1 mm, and $A_e = \pi(2\text{ mm})^2$. If half the ideal damping factor is to come from eddy current damping, for the bounce damper should have an ECD factor of 3e-4 N/m/s. This would require an average B_y = 200 gauss. This could be achieved with a 2mm diameter x 2mm thick NdFeB magnet, and a separation l_g of 3-4 mm.

5 Construction & mounting

The design concept is further fleshed out in the drawing shown below.

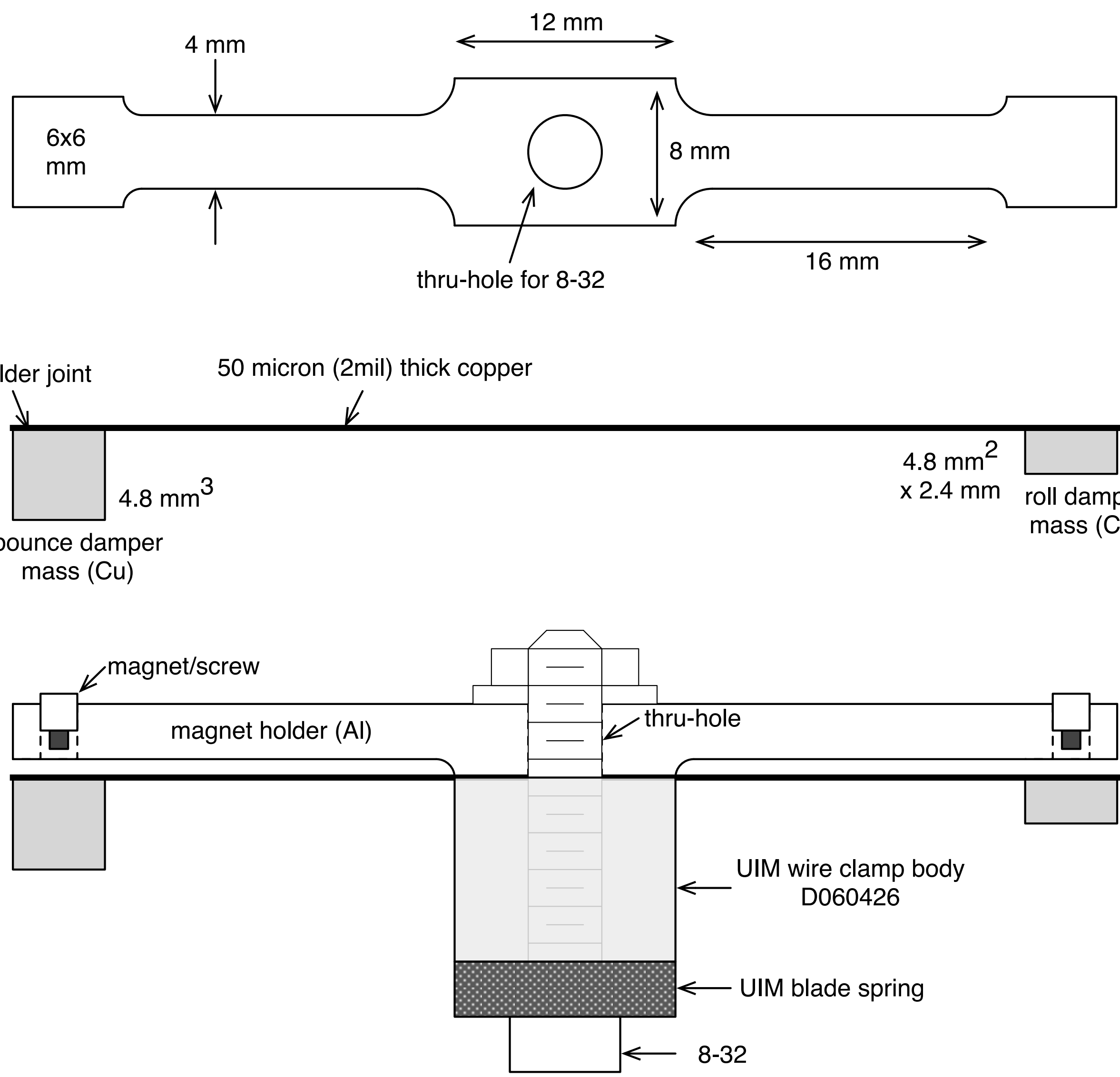


Figure 1. Top: Copper cantilever blade piece, used for both the bounce and roll mode damper. **Middle:** Side view of the cantilever blade and damper masses; the copper masses can be soldered to the copper blade. Though not shown, tapped holes could be included in the damper masses for frequency tuning screws. **Bottom:** Cantilever/mass assembly shown with the eddy-current dampers; an aluminum magnet holder mounts on top of the cantilever, and holds a magnet at each end. The magnet is bonded to a grub screw for adjustment of the damping coefficient. The cantilever/mass/ECD assembly is mounted on top of the existing UIM wire clamp body (D060426) with a single screw. This is the screw closest to the base of the UIM blade. The existing screw would need to be removed and replaced with a longer screw. The cantilever/mass/ECD assembly then would be slipped over this longer screw, and hold down with a nut, as shown. Note that this side view does not indicate the sag that would occur on each damper cantilever.

Note that there will be some coupling of longitudinal motion of the test mass (left-right in the above figure) to BRD motion, given that the damper blade will not be flat and the damper mass is not symmetric about the blade. Some estimate of this effect should be made to make sure that the longitudinal thermal noise is not compromised. This coupling would be expected to be smaller for a BRD assembly rotated by 90 degrees, such that the BRD blades were perpendicular to the suspension longitudinal degree of freedom. This wouldn’t fit so conveniently to the end of the UIM blade as drawn above, but it is an option that may need to be explored.

Since the damper blade will be curved under load, it is not clear if making the damper mass symmetric on the blade would reduce the coupling to longitudinal. But the option of symmetric damper masses should be considered. In that case the masses could be attached to the blade simply with a screw: