LIGO II Suspension: Reference Designs

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1 Aims

- To describe the baseline design for the LIGO II multiple pendulum suspensions for sensitive optics in BSC and HAM chambers.
- To indicate areas of flexibility as well as areas of rigidity of the design.
- To evaluate key performance measures.
- To identify and enumerate key interface parameters.

2 Outline

Design for thermal noise performance is paramount. This constrains the design of the lowest 3 stages of the suspension in most respects (therefore most aspects of the HAM suspension are constrained by this).
Second level design considerations are isolation and local damping.
A third level of design considerations is the provision of suitable actuators for global control.
The above have been arranged in order of increasing design flexibility (there is more possibility for change in the later items).
Note that the reference designs presented here replace those presented in the LSC White Papers.

3 Tools

The design work employs a number of tools.

- Thermal noise design (Maple code) [1]
- Mechanical design and performance simulation (MATLAB code) [2]

All non-commercial programs are made available to LSC members.

4 Design principles

In this section we describe the design principles for the BSC suspensions, the design process for the HAM suspensions is similar.

4.1 Thermal noise

The key aspect is reduction of the thermal noise originating in the suspension. Two main contributions are from dissipation in the silica fibers used to suspend the mirror, giving a direct horizontal noise components, and from flexing of the lowest set of blade-springs, giving a vertical noise component which couples into horizontal.

The noise from the fibres is minimized by suitable choice of the ribbon design.

The noise from the blades is reduced by choosing their stiffness, the ratios of the mirror and intermediate masses and the stiffness of the silica fibers in the lowest stage.

The dimensions of the lower masses, the density of the intermediate mass and the dimensions of the lowest blades are all constrained by the need to limit and filter thermal noise.

4.2 Suspension thermal noise estimation

Thermal noise calculations are based on Fluctuation Dissipation Theorem applied to coupled harmonic oscillators.

For each stage two degrees of freedom are considered: the horizontal and the vertical. Imposing a periodic force on the bottom stage, the maple code [1] solves the dynamic equations giving the displacement at the bottom mass and then the corresponding transfer function can be calculated. The coupling between the horizontal and vertical transfer function is assumed to be $\sim 10^{-3}$.

The bottom stage is monolithic whereas the upper stages are made using clamped steel wires. Considering that the thermal noise mostly comes from the bottom stage, at least above 10 Hz, the modeling of the silica stage is more detailed then the other stages. In particular, for the stages n and 1 (BSC suspension) an effective loss angle of 3×10^{-4} and an effective dilution factor of 30 has been assumed (these stages do not need to be state-of-the-art). For the stage 2 (BSC suspension) the respective numbers are 1×10^{-4} and 100. For the ribbons of the silica stage, a measured loss angle of 1.4×10^{-7} is used and the dilution factor is calculated from the theory. The thermoelastic effect has been included.

In the code the following modes are not present: internal modes of the blades, violin modes of the ribbons, internal modes of the test mass and all the rotational modes. The same techniques were used to model the HAM suspension.

4.3 Isolation and local damping

Optimum horizontal isolation would be obtained by having approximately equal lengths and masses for each stage of the suspension. Thermal noise requirements over-ride this, however, and the only action we take is to avoid any stages shorter than 300 mm (BSC). The limited height available for the HAM suspensions essentially determines the maximum attenuation, and we have tried to divide the space as equally as possible among the 3 stages (given other constraints).

Vertical isolation is provided by the blade springs in 3 layers (BSC) or 2 layers (HAM). We choose the softest springs that fit within the available space [1], while operating with a maximum stress of 850 MPa.

Damping of 22 to 24 modes of each (BSC) pendulum is required to a suitable Q (normally the target is about 5). This must be achieved without adding to the mirror displacement noise in an unacceptable way. Both the sensor noise and the design of the pendulum to isolate against this are important. The many remaining parameters n, s, d are chosen to give acceptable damping of all modes within the above constraints (see the accompanying sketches for definitions of all the pendulum parameters).

The two possible approaches are active damping and eddy-current damping. These are compared in section 6.5.1.

4.4 Actuation for global control

TBD The actuation requirements depend on the SEI solution adopted. We await selection of the SEI solution before proceeding too far in the design of the global control actuators. The designs will be based on those developed for GEO 600.

5 Possibilities for radically different designs

The HAM suspension design is essentially determined unless even more radical changes are made, e.g. to replace HAMs.

The BSC suspension design has been done taking into account the requirements that were presented to us by SYS. We have tried to minimise technical risk, where possible.

One of the fundamental assumptions that leads to the design we present is the common isolation platform for multiple suspension chains. This requires a clean break between isolation and suspension, and while this fits well with the nature of the 'stiff' suspensions, it has been pointed out that it does not necessarily give an 'optimum' solution with the 'soft' suspensions.

We are confident that we would be able to design suspension chains that can interface with somewhat different isolation system designs: consider an integrated isolation/suspension chain supporting one mirror. Here eddy current damping could be applied high up in the chain, and can provide suitable damping of all modes without adding too much noise. This might allow the elimination of one (at very most) stage of the combined system. It would not be excessively difficult to optimise the suspensions for this application. We have not, however, had time to fully evaluate this type of design. (We feel that a secure and stable design can

[1] We assume that the suspension can occupy a volume defined by a downward projection of the rectangular top mounting surface.

best be achieved by using two separate independently damped suspension chains for mirror and reaction mass.)

6 The baseline designs

As agreed, we have a working assumption of $10^{-11}\,\mathrm{m}/\sqrt{\mathrm{Hz}}$ for the local control sensor noise (vertical sensors being critical since longitudinal, pitch and yaw are turned off during sensitive modes). Most rigid body modes of the suspension are to be damped to $Q \sim 10$ or less. This necessitates a quadruple pendulum suspension [2].

6.1 Outline drawings and parameters

The sketches and schematic diagrams should be consulted along with the parameter list given in the Annex. Drawings of the suspensions are not available at present – further engineering design is needed over the coming year.

6.1.1 HAM suspension module

A sketch of the proposed HAM suspension module is shown in simplified form in figure 1. Note that this is a sketch intended to show the approximate layout of the parts, much more work is needed before we can provide a final drawing. We need to look at such fundamentals as precise beam height and mirror diameter before the details can be fixed. The design is very similar to the GEO 600 main suspensions (drawings provided previously), but with some modifications to make it as short as possible. We believe this is a very low risk design.

6.2 Key components

We provide conceptual designs of some of the key suspension components.

6.2.1 Attachment of the pendulum to the support structure

For the HAM suspension module the attachment consists of sitting the module in place on the optical table. There are some adjustable clamps to be designed. The optics are fixed in place for transport, the restraints are released and act as end stops for the motion. Much more detail to follow (adaptations of the GEO 600 design).

For the BSC suspension system there is a more complex arrangement. The suspension has to be assembled in a 'cage' which can be transported into the BSC and then suspended from the optical table. It is expected that most of the cage structure will remain *in situ* but sitting on the base of the chamber, not hanging from the optical table. This design needs some engineering input.

[2]Triple pendulum option: Given $10^{-13}\,\mathrm{m}/\sqrt{\mathrm{Hz}}$ sensors a triple pendulum design can be made to work at a saving of about 0.3 m in the overall length of the suspension.

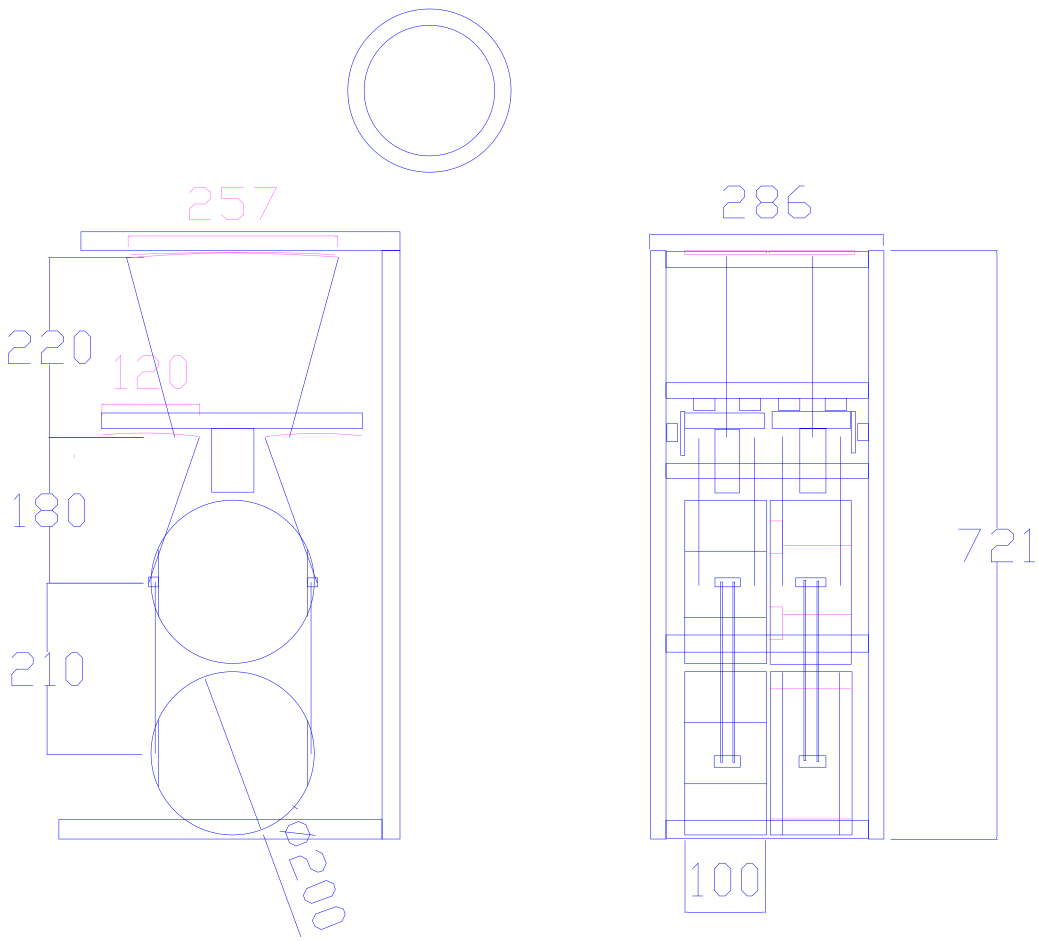

Figure 1: Sketch of the HAM suspension module with mirror and reaction chains in a support structure. The reaction chain is drawn with the thinnest (annulus) possible annular reaction mass, sketched in face view above the face and side views of the whole structure. Note that most detail is missing from the face view, and some from the side view. Grooves would be machined in the intermediate mass to avoid interference with the suspension wires from the top mass.

6.2.2 Designs of all blade springs, and their attachments

The basic blade dimensions are given in the parameter tables in the Annex. The blades are made from Marval maraging steel, processed in the manner described [3].

Clamps for the blades and wires are TBD (based on available techniques, low perceived risk, but acoustic emission must be avoided).

No dampers are proposed for the blades of the HAM suspension given the high isolation at the resonant frequencies, and since $Q <\sim 10000$ for these modes.

Dampers following the VIRGO/GEO designs may be proposed for one or more stages of the BSC suspensions. TBD.

6.2.3 Designs of the various upper metal masses

HAM suspension: see sketch in overall HAM suspension figure (fig. 1), detail to follow – but see also GEO 600 design.

BSC suspension: outline designs have been studied, but no final designs are available. Questions of material choice may influence the final design and require alteration of some suspension parameters. These are relatively complex mechanical elements and their design should not be rushed – TBD.

We aim to minimise the number of variants of these parts. This requires careful design of reaction chains to have the same mass as the mirror chains. It may not be practical in every instance.

6.2.4 Designs of the connections of the silica ribbons and break-offs at mass 2 (BSC suspension)

The baseline here is to follow the methods developed for GEO 600, but adapting them for ribbon technology. Significant research is needed.

The HAM suspensions use the techniques developed for the GEO 600 main suspension (the design is almost the same in many respects). Drawings will be provided later. Here the silica fibres are of circular section.

6.2.5 Design of the local control actuators, their holder and the support structure for the suspension

It is intended to use common active sensor/actuator units for the HAM and BSC suspensions.

The HAM holder/support structure is shown in the sketch (fig. 1).

The sensors and actuators remain to be designed (simple occlusion sensors and coil-magnet actuators, based on earlier designs).

This would vary if eddy current damping was employed, see section 6.5.1.

The BSC support structure has to satisfy a number of functions and has not yet been designed.

6.3 Global control, actuation and reaction chain designs

The reaction suspension chain must be customised for each type of suspension. For this reason the issues relating to global control, actuation and reaction suspension chain have to be dealt with several times over.

6.3.1 ETM suspension

A complete reaction chain is provided. The upper 2 stages would be very similar to those in the mirror chain. The lower stages would have similar masses and moments of inertia to those in the mirror chain, but be of low cost materials (metals, some silica). The precise design is TBD.

6.3.2 ITM suspension

It is not clear what the requirements are for the reaction chain here. Either 3 or 4 stages could be fitted depending on the feedback bandwidth that is requested. In the latter case an annular reaction mass could be used – as described below for the recycling mirror suspensions. If 3 stages were fitted we would attempt to design the lowest stage to have its mass equal to the mirror plus its intermediate mass – allowing the same designs to be used for the masses further up the chain.

6.3.3 BS suspension

We do not have sufficient detail of the requirements to design the BS suspension – TBD.

6.3.4 RM suspension

We assume the recycling mirrors need fast feedback ($>\sim 10\,\text{Hz}$). In this case we must use an annular reaction mass. The reaction chain needs to be re-designed, but we attempt to keep as many as possible of the geometrical parameters the same to minimise the number of different parts needed.

Our suggested option is to make a composite reaction mass from three annuli, two of silica and one of steel. One silica piece carries the electrostatic pattern while the steel provides almost the same mass and moments of inertia as the solid silica mirror. With a steel annulus of 200 mm O.D. 70 mm thick and 160 mm I.D., as the basis, we add radially similar 15 mm thick silica plates at each side. This provides 6.9kg mass (as per the mirror) within the same volume. The moments of inertia differ inconsequentially. This is the base-line design, subject to further detail.

6.3.5 MC:end suspension

Here we assume fast feedback is needed, therefore we fit an electrostatic drive behind the mirror. The reaction suspension chain can be of similar masses and moments of inertia to the main suspension chain. Detail to follow.

6.3.6 MC:transmitting suspension

We propose the same solution as per the RM (with any minor changes needed to fit the distribution of light beams), where wideband feedback is needed. Feedback up to 0.5 Hz is available even without a reaction suspension chain. If feedback with a few Hz bandwidth is needed we recommend fitting a double pendulum reaction chain (just top 2 stages). The top stage would be the same as for the main chain, with a metal lower mass of 13.8 kg replacing the two lower stages (lightened steel cylinder with same outline dimensions as the mirror). This gives the minimum number of new parts (especially blades, which need to be made in matched sets and are best made in large batches).

6.4 Interfacing

6.4.1 Wiring

The wiring follows the same pattern for all suspension chain pairs. If there is no reaction chain then there is no global wiring. Otherwise the same number of local control channels are needed for every chain, plus the global control channels for the pair. Local control is the same (in this respect) for BSC and HAM suspensions. Global control has fewer actuators in the HAM suspensions (however, we have not yet made the full estimates for HAM suspensions so in the mean time the BSC figures stand as 'conservative' estimates for the HAMS).

Wiring for local control. In the baseline design each local control sensor needs 2 signal channels (4 wires) one for the light source and one for the returned signal. An open question here is whether additional complexity (such as a monitor diode for the emitter or split detectors) will be needed to achieve the required sensor performance at 10 Hz. This could conceivably lead to a 50% increase in the sensor channel count. The local control actuators need one signal (2 wires) per degree of freedom. The total for a single pendulum chain is therefore 36 to 48 wires per chain for local control.

Wiring for global control. Here we assume the worst case of 4 actuators per stage (even through there are only 3 degrees of freedom to be controlled). This gives 3 stages of 4 actuators or 12 channels so 24 wires. An option is to allow tapped coil actuators (3 wires per unit) on one (probably not more) stages giving a total of 28 wires. The feedback on the lower stages would be electrostatic.

Ratings: sensors and electromagnetic actuators 100 mA and 50 Vdc with a higher rating for the electrostatic drive channels of 200 Vdc (100 V maximum expected bias voltage). The resistance from the electrostatic drive to the electrical feedthrough should be kept to $< 5\,\Omega$ per wire (to avoid damping of the pendulum). For the coil based actuators the resistance of each wire (from feedthrough to suspension point) should be $< 2\,\Omega$. (Figures to be rechecked.)

It is not yet known whether short or open circuiting of the electrostatic drive will provide the most efficient means of disconnecting it and removing any associated damping during sensitive running. Shorting it could conceivably require a lower impedance than is specified above, or another channel to operate some kind of (solid state) switch.

6.4.2 Power

The power dissipated within the vacuum system consists of $< 1\,\mathrm{W}$ per local control sensor and actuator, and $< 1\,\mathrm{W}$ (maximum) for a coil based global control actuator. This dissipation occurs in the coil itself. Additional dissipation will occur in the wiring. We suggest that there will be about 15 W dissipation per suspension chain pair (typical). It is not yet possible to provide a precise estimate for each actuator.

6.4.3 Mechanical attachment

The reference design assumes connection is to a flat optical bench type plate. SUS requires a defined area of free space on that plate. The dimensions of the required area are $\sim 0.35\,\mathrm{m}$ in the longitudinal direction by ~ 0.6 m in the transverse direction. No attachments are needed outside that area. The attachments will probably involve a single attachment of a rotational (yaw) stage on which the suspension chain(s) and associated parts hang (TBD).

6.4.4 Load

The suspension mass consists of the mass of the quadruple pendulum(s) and the mass of the support frame. We expect that the support frame will be partly demounted during installation, to reduce the load on the suspension, and to simplify the mechanical design (FE design to control resonances, especially). The mass of a standard quadruple pendulum is 151 kg (including blades). We could expect $20\,\mathrm{kg} < \mathrm{load} < 50\,\mathrm{kg}$ of additional load per suspension. The mass of a 2 chain system would therefore be 340 – 400 kg. (Precise figure TBD).

6.5 Additional notes

6.5.1 Local control options

As part of the overall design study we have carried out a review of the local damping method. This was needed as the situation is somewhat different from the one which applied in GEO 600 where the damping can be rolled off very strongly by the observation band. Given that the pendulum modes extend up to at least 5 Hz, and the observation band begins at 10 Hz, there is no reasonable method of filtering the noise at the lower end of the observation band by any significant amount.

The two options that we have considered are:

- **eddy current damping**, passive damping of the first (top) stage
- **active damping**, of the first stage using optical sensors and coil-magnet actuators for feedback

each in a 3 or 4 stage pendulum suspension.

Advantages and disadvantages-

Eddy current damping:

- can damp 2 degrees of freedom with each damper (if suitably designed)
- considerable reduction in control channels (wires, controllers)
- hard to vary the damping constant
- care needed to ensure stray fields are at an acceptable level (actually may not be much harder than with the active system's magnetic actuators)

active damping

- damping of each degree of freedom is easily controlled
- scope for 'tweaking' of the transfer function away from pure velocity damping to get a better settling time (relatively modest effect)
- 6 loops needed per suspension chain damped
- complexity of wiring and need for controllers is a disadvantage
- low noise sensors are needed

6.5.2 Performance comparison

There are 3 measures of the performance of a damping system

- damping: how well all modes are reduced in Q
- short-circuiting of the vibration isolation[3]
- noise: what noise is introduced into the damped mass

These performance measures are closely linked. With passive damping it is obvious that the isolation will be compromised for any frequency above $\omega_o Q$ when the magnitude of the 'stiffness' of the damper exceeds the magnitude of the stiffness of the suspension. A simple 1 Hz stage could only be damped to a $Q \sim 10$ if maximum attenuation at 10 Hz is sought. In fact these values are approximately appropriate to our case (with the greatest care one could get a $Q \sim 5$ for the lowest mode of the reference pendulum with suitable gain margin and active control). That a single stage is compromised above 10 Hz may not be very significant, and this can easily be evaluated.

Short circuiting occurs in both active and passive solutions (to a similar extent if the chosen Q is similar). It is not a problem for active systems when the gain is in any case turned down in order to remove sensor noise (but see below for the case of vertical isolation which may be reduced by this effect depending on the target Q chosen).

[3] This applies when a stage is damped with respect to a higher stage of the suspension.

Sensor noise in the active solution is replaced by a thermal noise force in the passive. The motion resulting from passive damping of the top stage of the quadruple pendulum to $Q \approx 10$ is $\sim 8 \times 10^{-21}\,\mathrm{m}/\sqrt{\mathrm{Hz}}$ at 10 Hz in the longitudinal direction [4]. This would be obtained with an active system if the sensor noise was $\sim 5 \times 10^{-15}\,\mathrm{m}/\sqrt{\mathrm{Hz}}$. Damping a triple pendulum passively requires a system that can be turned down during observation (to $Q > 100$) to avoid excessive thermal noise above the goal for the BSC suspensions. The performance requirements for the HAM suspensions can probably be met with eddy-current damping of a triple suspension.

An experimental test of a prototype pendulum with eddy-current damping suggests that achieving the necessary damping is feasible within the other design constraints. The dampers can be reasonably compact (a few units of about 50 mm by 50 mm by 20 mm perhaps).

7 Performance measures

A selection of output from the simulation codes, with the parameters given in the Annex as input. The information on the mechanical impedance of the system is to follow.

7.1 BSC Suspension

Note that the performance of the main chain is shown. The performance of the reaction chain is essentially identical except with respect to thermal noise.

7.1.1 Thermal noise

From the Maple code [1]. The result for the BSC suspension is shown (fig. 2). Since the silica technology is still being developed, curves have been shown which correspond to two different ribbon designs. The baseline design for the BSC suspensions is to use 1 mm by 0.1 mm ribbons. The first violin modes for the BSC suspension are then about 480 Hz.

7.1.2 Isolation

From the MATLAB code. To obtain the curves shown run qp_ref.m with the appropriate settings. Graphs with local damping on (fig 3) and off (fig 4) are shown for longitudinal motion (lc_onq = 1, 0 respectively) to show the difference between acquisition and running modes here. Vertical is always as shown in fig 3. The two longitudinal, undamped, transfer functions correspond to the shortest (poorer isolating) and longest (best isolating) suspensions which we recommend.

[4] This calculation was done accurately by introducing the appropriate viscous damping force on to the top stage of the suspension in the MATLAB model

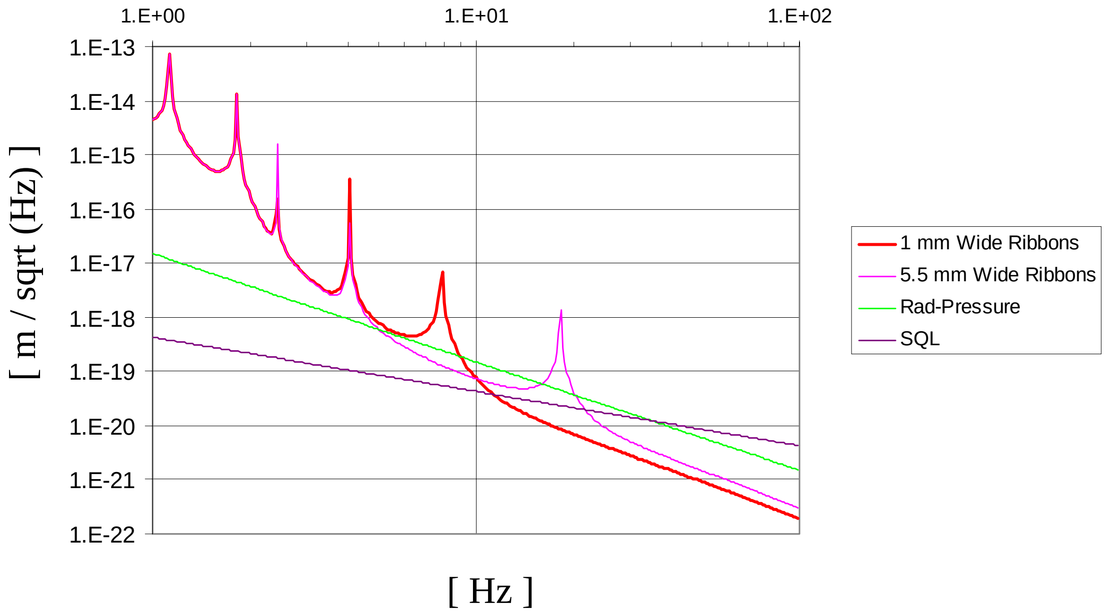

Figure 2: Thermal noise in the proposed BSC suspensions. The baseline proposal has ribbons of 1 mm by 0.1 mm cross section – giving excellent thermal noise. Ribbons 5.5 mm broad could also be used, with increased safety, increased thermal noise near 20 Hz and lower violin modes.

Figure 3: Longitudinal (red) and vertical (blue) isolation curves with local controls on. A cross-coupling factor of 0.001 has been included in the vertical isolation curve.

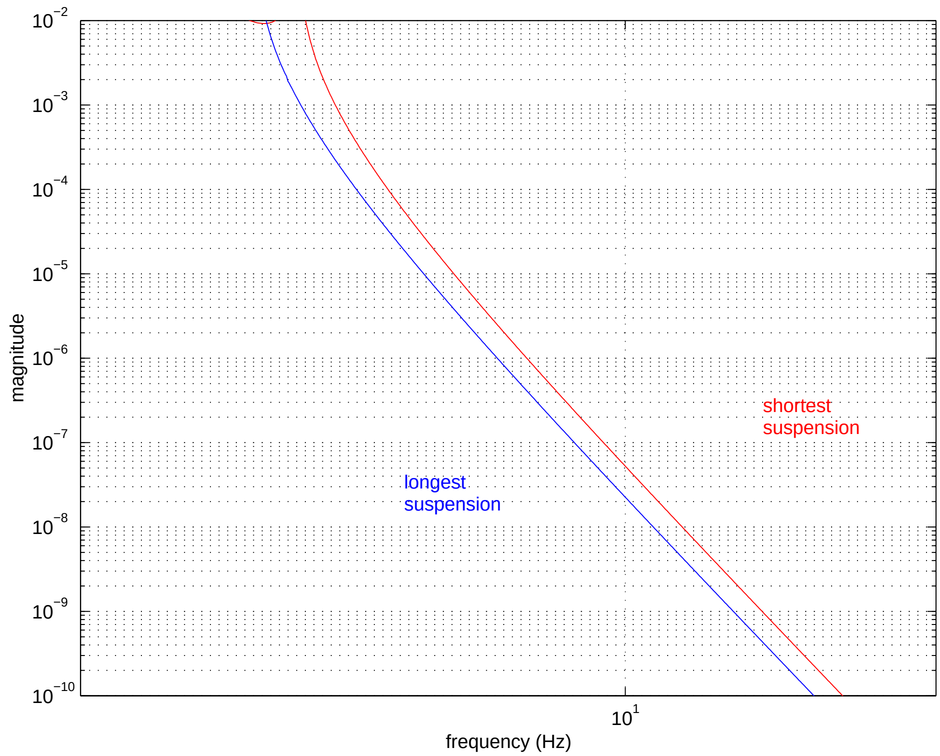

Figure 4: Longitudinal isolation curves with local controls off shown for both the shortest (1.67 m in red) and longest (2.00 m in blue) suspensions.

7.1.3 Local damping

From the MATLAB code. Obtain these results by running LTIVIEW after running qp_reference. Select the inputs/outputs marked 'LC' and plot the responses as needed – a typical example is shown here (fig 5). Note that the local control method and algorithm are to be finalised.

7.1.4 Global control

We include here instructions how to use the MATLAB code provided to generate the necessary performance measures of the complete SEI/SUS system. At this stage we make the approximation of an infinitely stiff suspension point for the quadruple pendulum. The SEI simply provides a displacement (PSD or sine wave) at that point.

Dummy curves are included here to allow a reality check for any curves subsequently generated by SEI. To obtain real curves insert into the function qp_reference (v1.01) at line 174++ the appropriate suspension point displacement power spectral density in $m/\sqrt{Hz}$. These curves should take into account the expected ground motion, any noise from sensors and the isolation provided, to produce the final suspension point motion.

The program will then calculate the residual *rms* motion with the suspension under local and global control (fig 6). An impulse response will illustrate system stability (fig 7). The actuation force needed at each length control actuator will

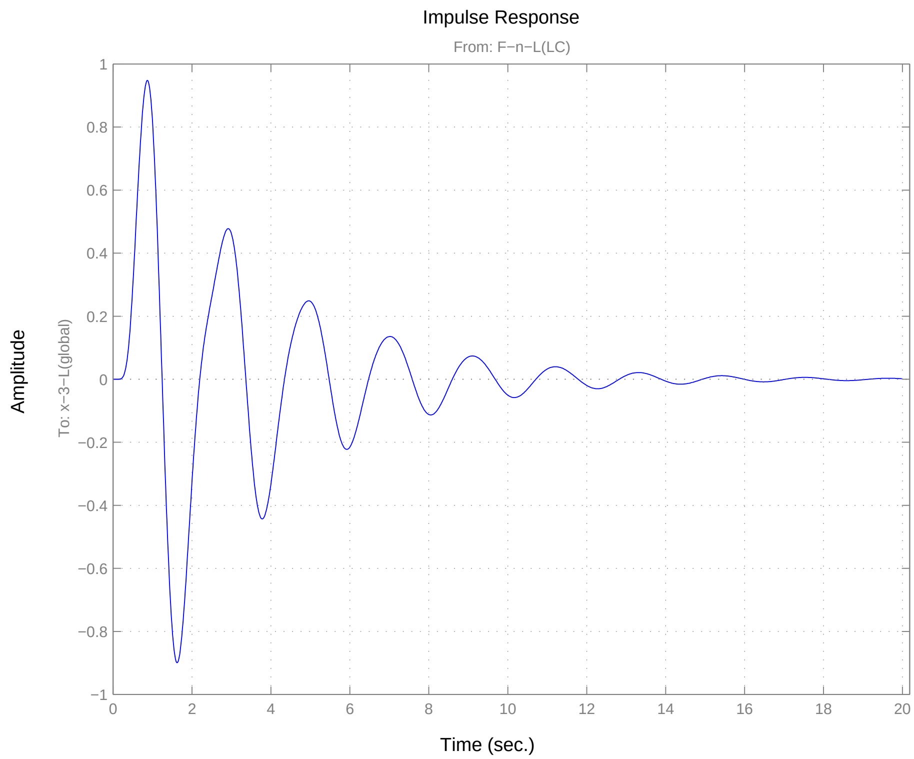

Figure 5: Mirror motion resulting from a longitudinal impulse applied to the suspension point, with the local damping on.

be plotted (fig 8). (Note that these curves require some interpretation as they show incorrect results above crossovers for non-dominant actuators.)

7.2 HAM (modecleaner or recycling mirror) Suspension

Note that the performance of the main chain is shown. The performance of the reaction chain is essentially identical except with respect to thermal noise.

7.2.1 Thermal noise

The results for the HAM (fig. 9) suspension is shown. Since the silica technology is still under development curves have been shown for two designs. The baseline design for the HAM suspensions uses 250 micron radius fibres giving first violin modes at about 470 Hz.

7.2.2 Isolation

From the MATLAB code. To obtain these curves run mc_example.m with the appropriate settings. Graphs with local damping on (fig 11) and off (fig 10) are shown for longitudinal (lc_onq = 1, 0 respectively) to show the difference between acquisition and running modes here. Vertical is always as shown (fig 11).

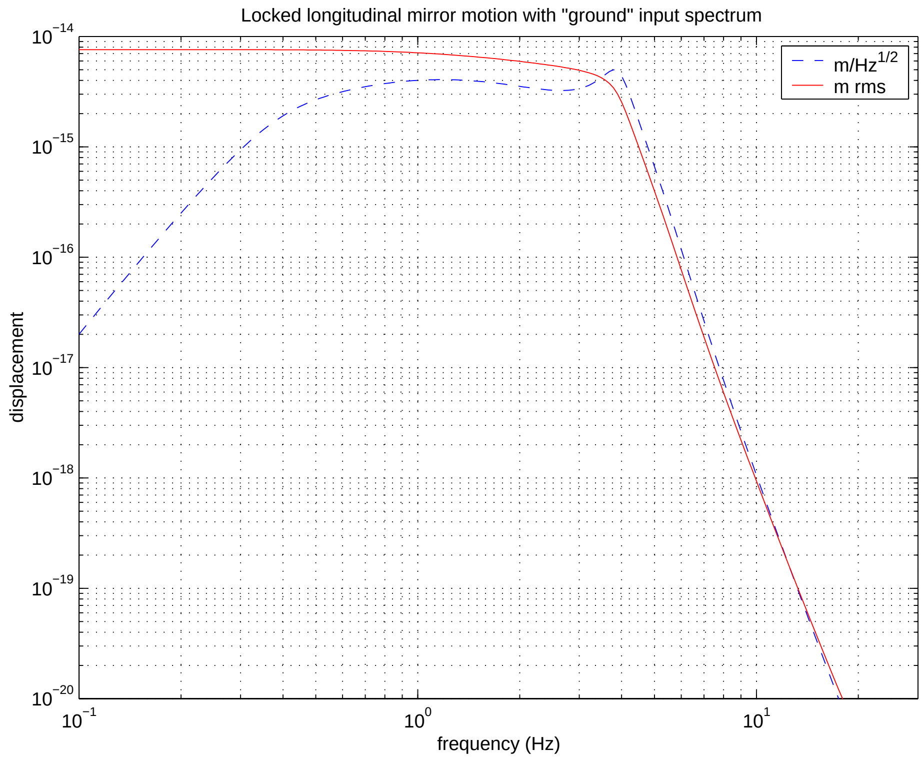

Figure 6: Residual longitudinal mirror motion with an example suspension point motion (roughly a normal Hanford ground spectrum) with global control applied.

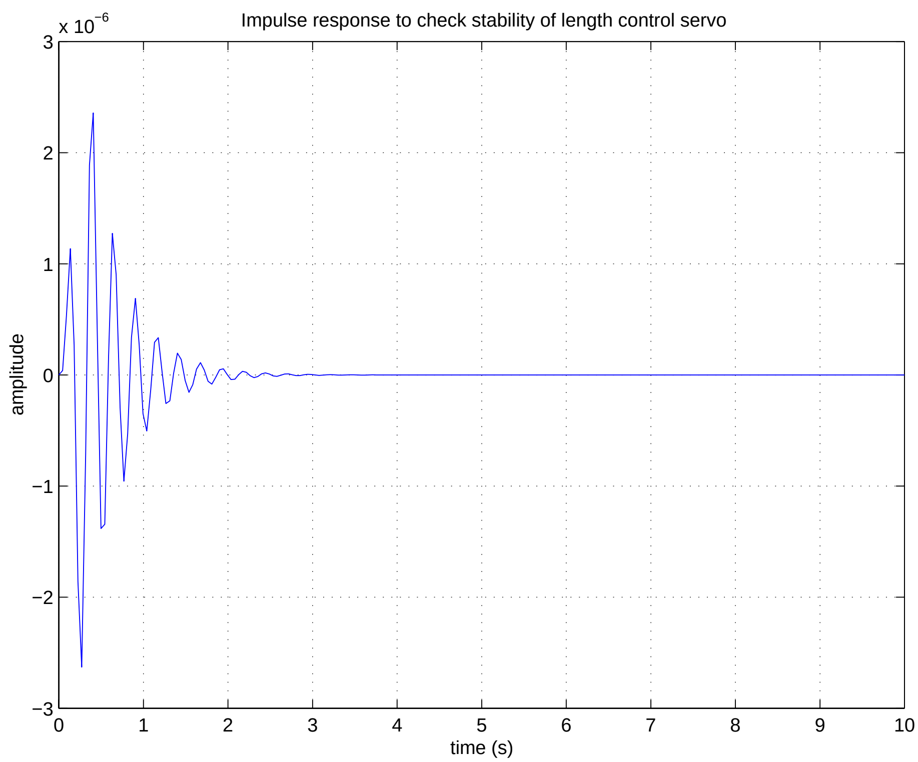

Figure 7: Impulse response with an example suspension point motion (roughly a normal Hanford ground spectrum), with global control applied.

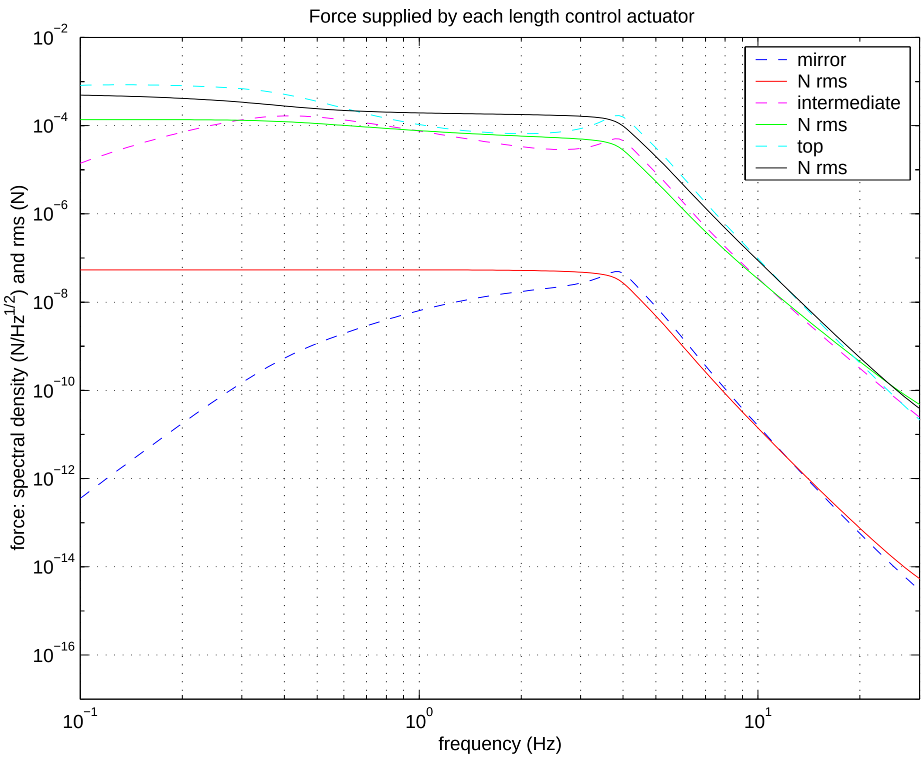

Figure 8: Actuator force with an example suspension point motion (roughly a normal Hanford ground spectrum). Note that one should only use information from this graph at frequencies below the unity gain frequency for a given actuator.

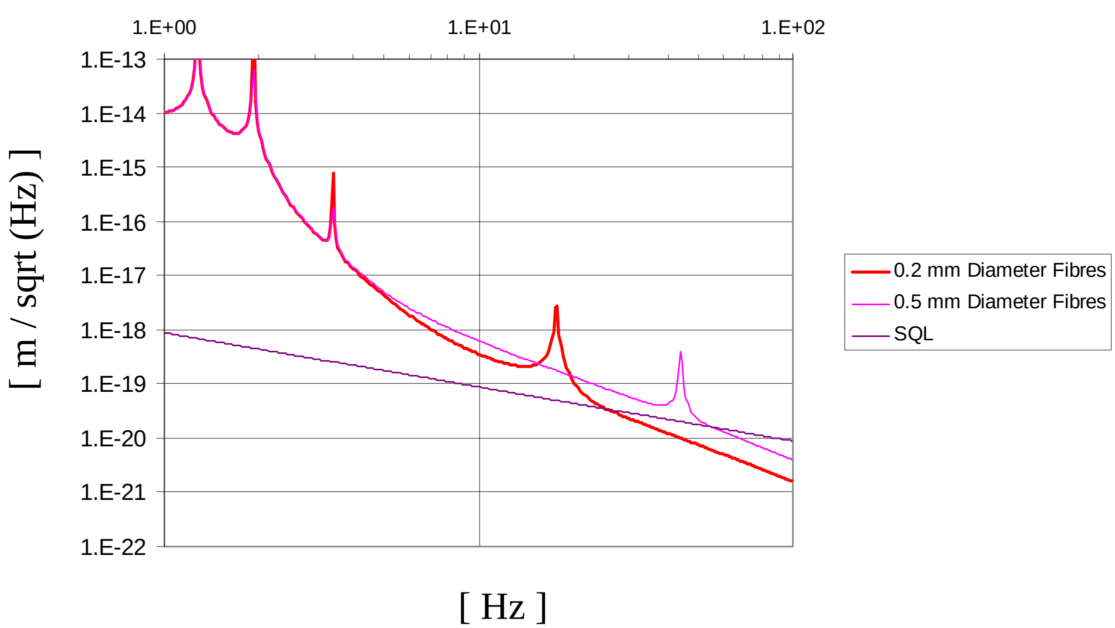

Figure 9: Thermal noise in the proposed HAM suspensions. This meets the goal with safety, except in a narrow band near the highest vertical mode.

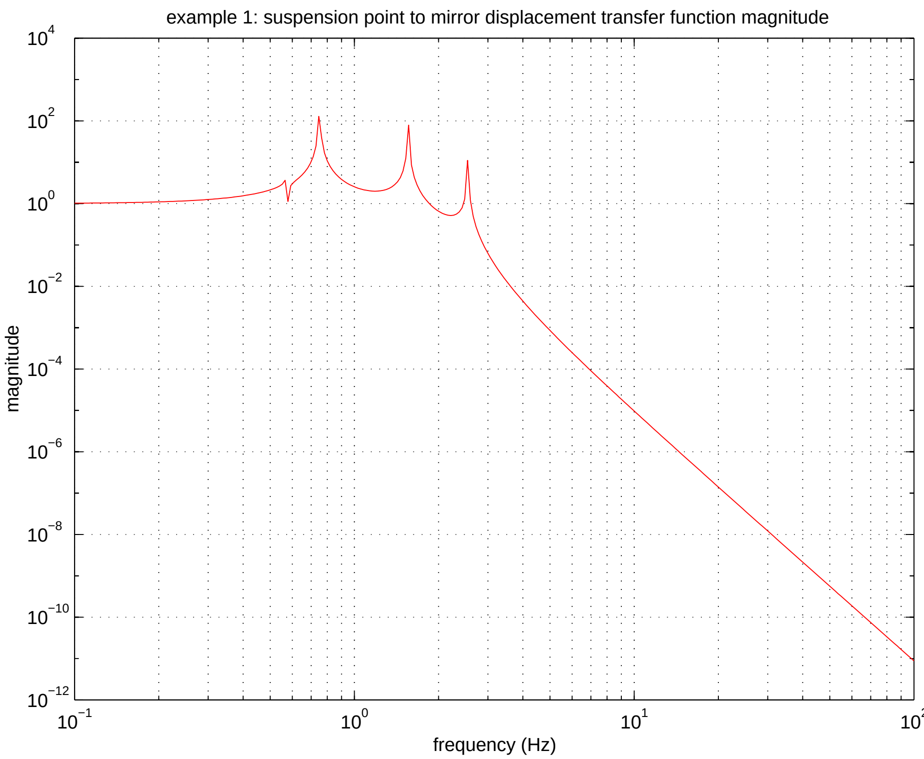

Figure 10: Isolation provided by the short triple pendulum when un-damped in the longitudinal. Note the peaks of the resonances are not resolved.

7.2.3 Local damping

From the MATLAB code. Obtain these results by running LTIVIEW after running mc_example. Select the inputs/outputs marked 'LC' and plot the responses as needed – a typical example is shown here (fig 12).

7.2.4 Global control

A global control example is not included for these suspensions. Actuation at any stage can be simulated by modifying the code as needed. Please contact us at Glasgow if you require advice or help. Note that typical performance will be similar to that obtained with the BSC suspension. TBD.

8 Conclusion

The BSC and HAM suspension chains have been presented in about as much detail as is possible at this stage in the design process.

References

[1] G. Cagnoli distributed with this note 01/2000.

Figure 11: Isolation provided by the short triple pendulum when damped in the longitudinal (appropriate for acquisition) and in the vertical (appropriate for all operating modes). Includes the short-circuiting effect of the local damping at nominal gain. No cross-coupling factor has been included in the vertical curve, which shows poorer isolation at high frequencies. These models should be reasonably accurate up to $> 50\,\mathrm{Hz}$ even though ideal springs have been used to model the blades.

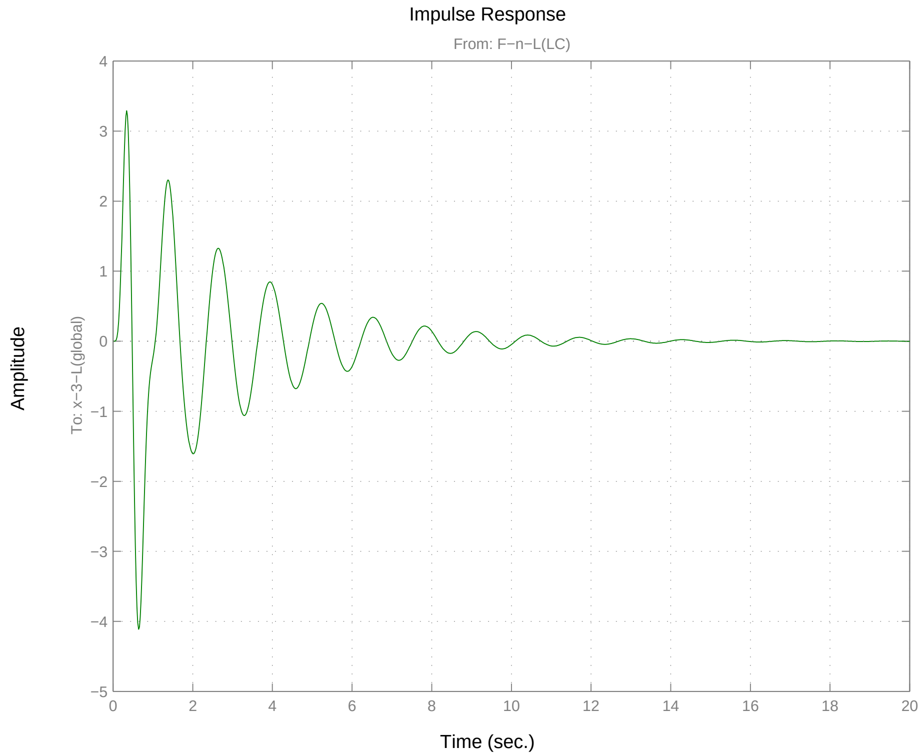

Figure 12: Ring-down of the longitudinal motion of the mirror after a unit impulse is applied to the local control actuator.

[2] MATLAB code as distributed to LSC members. The basis for this code is the state space model described in the PhD. thesis of C.I.Torrie, Glasgow 1999.

[3] Our design is descended from the VIRGO one. The methods are described in C.I. Torrie PhD. thesis, Glasgow 1999, and references therein.

Annex: Parameter list

8.1 Parameters for BSC quadruple pendulum

Masses of each stage, with some dimensions and densities where known.

```
mn: 29.4840                                                   %top stage
m1: 29.4840                                                   %stage 1
ix: 0.1200     ir: 0.1400     den2: 7800     m2: 57.6344  %stage 2
tx: 0.1200     tr: 0.1400     den3: 4000     m3: 29.5561  %stage 3
```

Further detail of the top two stages is not yet avialable, the design process is at too early a stage to give final dimensions.

Wire lengths, numbers, radii and elastic modulii

ln: 0.5400	nwn: 2	rn: 7.0000e-004	Yn: 1.6500e+011
l1: 0.3040	nw1: 4	r1: 6.0500e-004	Y1: 1.6500e+011
l2: 0.3020	nw2: 4	r2: 4.0200e-004	Y2: 1.6500e+011
l3: 0.6000	nw3: 4	r3: 1.8000e-004	Y3: 7.0000e+010

Blade lengths, bases, thicknesses, frequencies and stresses

lnb: 0.5500	l1b: 0.4800	l2b: 0.4000
anb: 0.1100	a1b: 0.0656	a2b: 0.0656
hnb: 0.0050	h1b: 0.0055	h2b: 0.0045
ufcn: 2.1874	ufc1: 2.3895	ufc2: 1.6627
stn: 8.6029e+008	st1: 8.3114e+008	st2: 7.7319e+008
intmode_n: 62.2273	intmode_1: 89.8703	intmode_2: 105.8836

The blade lengths, widths, thicknesses, uncoupled mode frequencies [5] per stage, maximum stress and first internal mode frequencies are listed (descending). The precise blade shape is TBD.

Pendulum break-off details

dm: 0.0010	dn: 0.0010	d0: 0.0010	d1: 0.0010
d2: 0.0010	d3: 0.0010	d4: 0.0010	
sn: 0	su: 0.0020	si: 0.0020	sl: 0.0070
nn0: 0.2500	nn1: 0.0500	n0: 0.2000	n1: 0.0700
n2: 0.1200	n3: 0.1465	n4: 0.1415	n5: 0.1415

See the schematic diagrams for definitions of these quantities. The ribbon design introduces new variables to account for the different effective lengths in different dimensions:

```
twistlength: 0.0060   %length of the twisted section of ribbon
d3tr: -0.0050         %effective flex-point for roll model
d4tr: -0.0050         %effective flex-point for roll model
```

The small difference in the effective ribbon length in different directions has been neglected at present.

Pendulum vertical lengths, and total length

tln: 0.5016	tl1: 0.2748	tl2: 0.3008	tl3: 0.6000
l_total: 1.6772			

These are the vertical suspension-stage lengths, given for the shortest recommended suspension. The longer design can be obtained by adding 0.1 m to the length of each of the top 3 stages. No other change is needed. (The revised parameters are in the MATLAB program quadopt.m, but commented out.)

[5] The frequency which would be observed if the blade were supporting just its dynamic load, not the total static load that it normally bears.

Local control parameters and actuator positions

gain: 6	gainvpr: 6	gainlty: 6
lever_pitch: 0.0600	lever_roll: 0.0900	lever_yaw: 0.0900
gain_pitch: 6	gain_roll: 2.8000	gain_yaw: 6
gain_longitudinal: 6	gain_vertical: 4	gain_transverse: 12

8.1.1 Parameters for HAM triple suspension

Mass outline dimensions, masses and moments

m1: 12.6000	material1: 'steel'	
m2: 6.9178	material2: 'silica'	
ix: 0.1000	ir: 0.1000	
m3: 6.9178	tx: 0.1000	tr: 0.1000

Wire lengths, numbers, radii and elastic modulii

l1: 0.2200	nw1: 2	r1: 6.0500e-004	Y1: 1.6500e+011
l2: 0.1800	nw2: 4	r2: 3.0200e-004	Y2: 1.6500e+011
l3: 0.2100	nw3: 4	r3: 2.5000e-004	Y3: 7.0000e+010

Blade lengths, bases, thicknesses, frequencies and stresses

l1b: 0.2800	a1b: 0.0450	h1b: 0.0025
ufc1: 2.0831	st1: 7.7454e+008	intmode_1: 120.0494
l2b: 0.1200	a2b: 0.0311	h2b: 0.001
ufc2: 2.9808	st2: 7.8528e+008	intmode_2: 261.4410

Pendulum break-off details

di: 'all 0.001'		
si: 0.0400	sl: 0.0150	
n0: 0.1300	n1: 0.0600	n2: 0.0400
n3: 0.1065	n4: 0.1015	n5: 0.1015

Here round fibres are used so there is no twist to take account of.

Local control parameters and actuator positions

lever_pitch: 0.0400	lever_roll: 0.0900	lever_yaw: 0.1000
gain_pitch: 0.5000	gain_roll: 0.3333	gain_yaw: 1
gain_longitudinal: 1	gain_vertical: 0.3333	gain_transverse: 2

Note that the stage spacing in this suspension is given by the wire lengths to within ~ 2 mm. The final dimensions are subject to variation by a few mm.

Note that the mass and outline dimensions for mass 1 are subject to modification during the final design process, for this reason the outline dimensions have been suppressed. This is a relatively complicated object and the values given above should be regarded as guide or target values. The overall pendulum sketch (and full drawings of the GEO 600 main suspension) give a good idea of what this mass looks like.

There is some scope for variation without loss of performance. Yaw damping performance, while adequate, is subject to checking when the full design is available.

Note that 4 grooves must be taken out of the intermediate mass to clear the 4 steel wires which support the mass. These wires need to taper. The need for grooves can be avoided if the suspension is made about 80 mm longer.