LASER INTERFEROMETER GRAVITATIONAL WAVE OBSERVATORY

- LIGO –

CALIFORNIA INSTITUTE OF TECHNOLOGY
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

Document Type	DCC Number	Date
Technical Note	**LIGO- T040197-00-R**	14^{th} November 2003

The Deep Fall Back Solution.
Passive External Pre Isolation and Stack Damping for LIGO

Status report November 2003

Alessandro Bertolini, Riccardo DeSalvo, Francesco Fidecaro, Maddalena Mantovani, Szabolcz Marka, Virginio Sannibale, Duccio Simonetti, Akiteru Takamori, Hareem Tariq

Distribution of this draft: TBD
This is an internal working note
of the LIGO Project.

California Institute of Technology
LIGO Laboratory - MS 18-34
Pasadena CA 91125
Phone (626) 395-212
Fax (626) 304-9834
E-mail: info@ligo.caltech.edu

Massachusetts Institute of Technology
LIGO Laboratory - MS 16NW-145
Cambridge, MA 01239
Phone (617) 253-4824
Fax (617) 253-7014
E-mail: info@ligo.mit.edu

www: http://www.ligo.caltech.edu/

1. Introduction

Due to intermittent low frequency seismic activity[1] the LIGO interferometers experience difficulties in achieving or maintaining lock for long periods. Occasional seismic bursts, with velocities up to 5 μm/sec in the 1-3 Hz frequency band, excite the resonant modes of the Seismic Isolation Stacks (SIS) to the point of overwhelming the LIGO mirror's control authority. In early 2002 it became clear that an urgent remedial action was needed. Large reduction of the mirror's low frequency residual motion are needed to reduce the mirror control authority necessary to maintain lock and to reduce the associated control noise. Careful assessment of the problem generated the requirement of at least 30 dB of additional attenuation in the 1-3 Hz frequency band[2]. Since LIGO is a running experiment, an external solution not requiring extensive intra-vacuum modifications was needed for the short term.

The ideal solution to eliminate the problem of the LIGO SIS's resonances being seismically excited is to suitably disconnect the Stack's Support Structure (SSS) from the seismic excitation; this step alone, if extended to sufficiently low frequency, would already exceed the requirements above. The residual SIS resonances would be detectable with external sensors on a floating SSS. The SIS resonances could be damped using external actuation (inertial damping); this step would further reduce the mirror actuators load and their associated actuation noise. Additionally active attenuation can be implemented to attenuate the seismic excitation well beyond the present remedial action requirements.

The geometry of the SIS, SSS, downpipe, and a BSC vacuum tank are illustrated in figure 1.

Three solutions were proposed and developed, all three introducing preattenuators at the top of the piers that support the SSS, and all three compatible with the in vacuum two stages of active attenuation foreseen in the Adv-LIGO baseline design.

Two solutions, the Magnetic External Pre-Isolator (MEPI) and the Hydraulic External Pre-Isolator (HEPI), are based on purely active attenuation, actuating on stiff suspension springs. The HEPI is the baseline outer stage of the Adv-LIGO active attenuation system. MEPI is a modification of HEPI in which the HEPI hydraulic actuators are replaced with electromagnetic ones. The third solution is derived from the previously downselected passive LIGO Seismic Attenuation System, a successful technique thoroughly validated by Virgo and the TAMA-SAS 3m interferometer tests. This solution relies on the passive disconnection of the SSS from the seismic excitation by means of low frequency suspensions. This report describes the development and the results obtained with prototypes testing this last solution.

The proposed plan was to produce, implement and test a LF passive isolation prototype in LASTI before the two stiff systems could be ready for tests, (this solution being simpler is much faster to implement) and, if successful, to implement them in the Livingston observatory at the end of the LIGO S2 data acquisition run while waiting for the baseline system to be developed. In March 2002, confidence in a speedy and successful implementation of the two stiff active solutions put this simpler idea in the backburner and it became the so called “deep fall back solution”. The Geometric Anti Spring Filter (GF) full size prototypes, one already in testing phase and one in production were finished and tested in a low priority mode. The latter GF prototype was a one sixth prototype (a full size, 2 blade version of the 12 blade GF necessary to equip each BSC pier) of a final design BSC SSS suspension system. Similarly, a one leg, full size, final design, IP prototype (one of the 3 IP legs supporting each GF) was produced and tested.

The development of the proposed solution is continuing in view of the predicted necessity to disconnect vacuum tanks from seismic excitation in future underground interferometers designed to operate beyond the present Newtonian Noise limitations.

Principle of operation.

The separation of a payload from the seismic disturbance can be effectively achieved with a sufficiently soft suspension. The force transmitted by a suspension and acting on the suspended masses is proportional to the suspension elastic constant multiplied by the seismic displacement. The softer the suspension, the smaller the transmitted forces and the smaller the acceleration applied on the suspended masses. The well-known 1/r2 roll off above the mechanical resonance frequency of the suspensions describes this behavior. To be effective, a soft suspension must float its payload with a suspension frequency that is substantially lower than the internal resonances of the payload itself (1-3 Hz for the SIS). A 100 mHz (or lower frequency) SSS effective suspension resonant frequency would easily satisfy the remedial isolation requirements.

There are several additional advantages provided by a soft suspension of the SSS.

- If the SSS is bolted to ground, or subject to an active attenuation mechanism it acquires an effectively infinite mass. The 580 Kg optics-supporting Down Tube (DT) resting on the attenuation stacks, resting on the SSS, behaves as a mass attached to ground through the stack's composite spring constant. The DT movements are then, in first approximation, not detectable by external sensors mounted on the SSS. If, on the contrary, the 2080 Kg SSS is softly supported, the DT oscillations react against the SSS mass. The recoil of the SSS (roughly 25% of the DT movement amplitude) can then be detected externally and counteracted effectively.
- The softly suspended SSS and DT connected by the SIS composite spring behave like a reduced mass, and the fundamental SIS resonances shift at higher frequency, where not only they are subject to less seismic noise, but they are also easier to damp.
- A soft suspension naturally offers an ideal platform for tidal corrections.
- A softly suspended SSS is the ideal platform to implement an active attenuation system, which would then require only tiny correction forces. Soft actuators can be made non-contacting and with constant force (vs. position), so they will not contribute a noise paths towards the stacks system.
- A soft connection between the piers and the SSS is also useful because it decouples the potentially dangerously close resonances of the pier (25 Hz) and of the SSS ones (23 Hz[3]).
- The large stroke (up to 10 mm) of a soft suspensions offers valuable earthquake protection.

The bellow's problem and its solution:

A soft suspension scheme was initially considered to be practically unfeasible given the topology of the LIGO SSS with its support points in air and the stacks support point inside UHV. Even if one was to implement a zero stiffness suspension of the SSS (a step equivalent to magically eliminating gravity and the necessity of external weight supports), the stiffness of the four vacuum bellows would still cause the SSS to oscillate at about 0.75 Hz[a] and short circuit a good fraction of the ground's seismic excitation to the SIS. The solution to this impasse and to passively satisfy the isolation requirements is to use the SAS know-how to neutralize the bellow stiffness.

The trick is to couple the bellow stiffness to a negative elastic constant suspension spring to induce a lower pre-attenuation resonant frequency. This can be achieved mounting negative stiffness an Inverted Pendulum (IP) and GAS Filter (GF) unit on each pier, in parallel to each of the four vacuum bellows. The scheme does not require modifications of the LIGO vacuum chambers. The negative stiffness IP and GF can be entirely made out of existing and well tested components. The simplicity of the mechanics makes the construction and installation cheap, fast and easy. The technical details are explained below; a complete executive design[4] and construction of prototypes using the same design were made in few weeks before March 2002.

Performance estimation:

In Virgo superattenuators and the TAMA SAS design, IP and top GF have been measured to produce 60 dB of passive attenuation factors even below the Hertz. Because of the differences between the topologies of the TAMA SAS suspension chain and the BSC/HAM chambers it is unlikely that the full TAMA performance will be achieved. Happily the LIGO remedial solutions, only requires attenuation of 30dB above1Hz which should be straightforward to achieve with the purely passive pre-attenuators, even taking into account the more complex LIGO composite topology.

Experience teaches that, once achieved a low frequency resonant frequency, a few factors can limit the system passive attenuation performance.

The bellow's stiffness can be easily neutralized, but in order to neutralize the seismic noise power that they inject on the SSS, one need to insure that each bellow and its compensation negative stiffness IP/GF are rigidly connected and are subject to the same

---

[a] The SSS mass is 2080 Kg. If we were to float it without gravity, only connected to ground through the bellows, the stiffness of the bellows would still cause the suspended structure to oscillate at a frequency F where:

$$F = \sqrt{(K/M)} / (2\,\pi) = \sqrt{(4.4\ 10^4 / 2080)} / (2\,\pi) = \sqrt{21.7} / (2\,\pi) = 4.66 / 2\,\pi = .74\ \text{Hz}$$

This resonant frequency would deliver only a factor of 2 attenuation at 1 Hz (20 at 3 Hz) and fall short of the required 30 dB of attenuation at 1 to 3 Hz.

seismic excitation. The bellows are mounted on the BSC vacuum vessel while the IP/GF are mounted on top of the piers. Both the vacuum vessel and piers are solidly bolted to the ground concrete slab. The tank is quite stiff and the piers present their first resonance above 25 Hz. Consequently one can consider the bellows and the IP/GF as acting rigidly at least up to 20 Hz. The IP/GF can then be expected to easily neutralize the seismic noise injected through the bellows in the required 1 to 3 Hz frequency band. For additional safety, and to further extend the attenuation bandwidth, each pier can be stiffened with respect to the vacuum chamber by adding a simple pair of straps between the pier and the close by vacuum chamber flanges.

A second limitation is due to the fact that the bellows are not massless springs and therefore can transfer seismic power through their resonances (at 33 Hz and above), a path that cannot be neutralized by the IP/GF negative stiffness mechanism[b]. This limitation though appears well outside the frequency range of the requirements. Additionally the bellow resonance problem can be mitigated by means of external eddy current damping mechanisms.

The next remaining seismic power injection mechanism is represented by the bellows acting as distributed masses, transferring seismic momentum to the SSS while recoiling against their own momentum of inertia. This mechanism is the equivalent of the seismic power transferred by the unbalanced leg on an IP, it results in a frequency independent saturation of the $1/r^2$ attenuation behavior at a level which is of the order of the mass ratio of the element in question (the bellow) divided by the mass of the payload it carries. The bellows cannot be balanced like an IP to null this effect. The mass ratio of the bellows and its payload is of the order of a fraction of a percent and therefore this path is expected not to introduce a passive attenuation saturation before the 40 to 60 dB level, well beyond the required 30 dB of isolation.

It can be safely concluded that neutralizing the bellow's stiffness would generate much more than the desired 30 dB remedial seismic attenuation. The excess attenuation

[b] The bellow's resonances recoil against the suspended mass and the mass ratio is very favorable so this problem is not expected to be a serious one unless the bellow resonances have very high Q factors. The bellows though are easily accessible; consequently their resonances Q factor can be easily spoiled with external Eddy current dampers. Alternatively, the bellow resonant problem, could be treated by means of an additional active attenuation system, much lighter, but topologically identical to that of the HEPI and MEPI.

is in any case useful and can result in upgraded interferometer performances. In particular, smaller residual mirror motion would strongly alleviate the mirror's control authority requirements and reduce the control noise, which is an important interferometer performance limiting factor.

Negative stiffness mechanism implementation

A 100 mHz suspension resonant frequency, sufficient to produce 40 dB attenuation at 1 Hz and exceed the requirements, is discussed in this chapter. Of course even lower frequency tuning are possible for expanded passive attenuation capabilities, but, as discussed later, they probably require electromagnetic springs in parallel for the fine tuning.

The LIGO bellows have been measured[5] to be reasonably good quality springs with transversal stiffness:

$K_x = K_z = 4.4\ 10^4$ N / m

and longitudinal stiffness:

$K_y = 0.6\ 10^4$ N / m

In order to neutralize their stiffness, the four IP/GF combinations suspending the 4 extremities of the BSC (or HAM) SSS must provide a negative stiffness of $4.4\ 10^4$ N/m each, as well as the lift necessary to float the payload. This is achieved with the mechanical design illustrated in figure 2. The SSS isolation system would consist of 4 mini-towers, each made by a GF and a tri-legged IP, mounted on top of each pier.

The IP has the same elastic constants in both horizontal directions. Due to the softer response of the bellow in the longitudinal direction, a corrective spring with

$K_y = 3.8\ 10^4$ N/m is mounted in parallel with each bellow's axis to equalize the bellow's stiffness in the two horizontal directions.

The BSC SSS weights 2080 Kg. To achieve a resonant frequency F = 100 mHz (± 10 mHz) we need to achieve a cumulative bellow and suspension effective elastic constant $K_{eff} = 0.8\ 10^3$ N/m (± $0.161\ 10^3$ N/m), which means reducing by a factor of 50 +/- 5 the original bellow stiffness. The required amount of negative stiffness and tolerance can be achieved tuning the GF radial compression for the vertical d.o.f. and the IP length and ballast load for the horizontal directions.

The generation of the required negative stiffness in the vertical and horizontal directions is described separately.

Horizontal negative stiffness:

Each IP minitower must support 1.3 tons of weight and a budget of 200 Kg of GF and ballast mass.

The anti-spring constant of a elastic-stiffness-free IP is

$K = - M g / h$

where M is the applied load [Kg] and h is the effective IP leg length [m].

To get the desired anti-stiffness $K_x = K_y = - 4.4\ 10^4$N/m we impose a leg length l_{IP} using the formula

$K_{as} = -4.4\ 10^4 = - 1500 * 9.8 / l_{IP}$ [Kg/m].

$l_{IP} = 0.334$ m

An IP equipped at both extremities with the tensional flex joints used at the top of the TAMA IPs is, to all practical purposes, an elastic-stiffness-free IP. The added advantage of tensional joints is that they are easier to use, buckling free, and allow for the relatively large angular movement required to maintain seismic isolation during earthquakes with such a short IP legs.

Fine tuning (and a reserve) of IP negative stiffness is obtained by adding or removing ballast mass on top of the GF body. The required stiffness accuracy (161 N/m for 100 +/- 10 mHz tuning) can be obtained with weight increments of 5 Kg. Of course the $K = 3.8\ 10^4$ N/m corrective spring mounted in parallel to the bellow's axis must be tuned to the same 161 N/m accuracy. The correction spring is made with a Virgo design cantilever blade attached to the SSS by means of a wire strung along the bellow axis. Small, commercial, helical springs mounted in parallel would provide stiffness fine tuning capability.

Vertical negative stiffness:

The vertical stiffness of the bellow is compensated by means of a negatively tuned GF.

The proposed GF is a simplified version of the LIGO-SAS GF prototype. It does not need the baricentral casing used in suspended filters, instead it would be equipped with the external frequency tuning mechanism used in the SAS prototype R&D. A negative stiffness GAS is simply a null stiffness GAS tuned with a larger amount of radial compression. Of course the different tune require a different relative angle between the blades and the wire linking the blades to the load. The geometrical comparison between a neutral and a negative stiffness GAS is shown in figure 3. The radial stress in a negative stiffness GAS is two or three time higher than in a neutral stiffness GAS, while the stress on the blade remains practically unchanged.

Each mm of radial compression of one GAS blade generates a incremental negative stiffness

$\Delta K_{blade} \sim 300$ (N / m) / mm

which means that the required stiffness accuracy (161 N/m) can be obtained with a radial compression tolerance of a fraction of millimeter per blade.

The GF is mounted on top of the IP and the four SSS beam ends would be bolted directly below the GF load disk.

The GAS filter geometry has been chosen, instead of the more advanced Monolithic Geometric Anti Spring geometry, because the GAS geometry uses separate suspension blades that can be replaced, even in situ, and therefore offers much more flexibility of use. Tests with the final prototypes showed the possibility of using bolted, monolithic-like geometries taking advantage of the simpler monolithic geometry while retaining the blade replacement capabilities.

Initial implementation

The ground tilt and temperature in the two LIGO sites is stable enough to allow passive tuning of both the horizontal and vertical frequencies down to 100 mHz, which is sufficient to produce the desired attenuation factor at 1 Hz and above. Hysteresis in the GAS springs may require the use of a parallel electromagnetic spring to achieve 100 mHz tuning. These electromagnetic springs, would then allow to compensate for thermal variations and allow much lower frequency tune.

Performance upgrades

Tuning the IP and GF to lower frequencies, say 10 mHz, would introduce very useful attenuation in the micro-seismic peak frequency region (at ~ 150 mHz). Purely mechanical tuning of the hardware to these low levels of elastic constants is practically impossible because of material hysteresis[6] and thermal stability issues. Introducing an electromagnetic spring in parallel with the IP or GF stiffness can solve the problem. The variable stiffness spring is obtained by driving a linear force actuator with a current proportional to the displacement read out by a co-located position sensor. Varying the gain of the control circuit changes the stiffness of the electromagnetic spring. With this fine correction capability an arbitrarily low effective resonance frequency can be achieved even when supercritical damping impedes the mechanical fine tuning used at higher frequencies[7,8]. At these low tuning frequencies the hysteresis in the materials, thermal drifts, and the daily tilt of the ground become relevant and would prevent setting and maintaining the GF and IP working points. It is calculated that a very low frequency working point correction algorithm, using the integrated error signal of the LVDT position sensors to generate a DC correction current for the constant force actuators would solve this problem as well as providing the necessary tidal corrections. A LVDT/constant-force-actuator configuration suitable to achieve these results is shown in figure 4.

It is useful to note that the entire fine tuning system, like the IP/GF mechanics, can be built using existing and extensively tested components[9],[10].

More advanced options: SIS inertial damping and active attenuation.

Advanced options will require some level of R&D prior to implementation.

Once the SSS is effectively freely floating in space, LVDTs, accelerometers and constant force actuators can be used to sense and damp the residual SIS excitation. We expect that SIS resonance damping can be performed from outside the vacuum tanks because the accelerometer sensors, operating on an already pre-attenuated platform, are expected to be well sensitive to the recoil of the stacks modes. These sensors will be in air, and therefore exposed to acoustic noise, this is expected to be of secondary importance because the acoustic noise generally starts being dominant only above 50 Hz and can be shielded with suitable an-acoustic cocoons surrounding each pier head.

The residual motion of the mirror payload can be depressed with a simple viscous damping scheme. A SISO systems should be sufficient for this task, as indicated by the successful damping tests that we performed, in air, to a prototype LIGO isolation stack [Virginio's AID tests].

Viscous damping can be upgraded to inertial damping, as in the Virgo damping scheme. It is worth noticing that the proposed scheme of the IP/GF pre-attenuators followed by the SIS passive attenuation chain is topologically analogous (although working at higher frequencies) to the successfully tested Virgo super-attenuator chains, i.e. a chain of passive oscillators preceded by a low frequency pre-attenuator equipped with inertial damping. The validation of the Virgo inertial damping scheme applies directly to the SIS scheme. The accelerometers for inertial damping are foreseen as an option in the design.

Further improvements would require a complete, wideband, Multiple Input-Multiple Output (MIMO) active attenuation feedback scheme. It is important to note that low frequency suspension produce a large frequency separation between the suspension resonances and the SSS internal frequencies. Because of this large separation, and because of the smaller and lower frequency forces needed to actuate on a body suspended at low frequency, we can safely consider the SSS as a rigid body and MIMO active attenuation becomes easier to implement than in the HEPI or MEPI scheme.

Inertial damping and active attenuation, being an optional not required to satisfy the remedial action requirements, could be implemented in a staged sequence to reduce the mirror control requirements and associated noise when and if necessary or desired.

Initial prototype tests.

We initially made three quick prototypes two of GAS springs and one with IPs, all using surplus vertical accelerometers and SAS components. In the mean time we produced a complete mechanical design of a BSC seismic isolation system and used it to build and test a full-size, two-blade GF pre-production prototype. The IP prototype leg was also a full size, full load compatible unit. The proposed BSC or HAM system consists of a 12 blade, 3 IP legs unit per pier.

To test feasibility of the negative stiffness filters we neutralized the stiffness of commercial springs (chosen to simulate the bellow stiffness) using the negative stiffness GAS prototypes in the vertical direction and IP in the horizontal plane. The tests on the preliminary prototypes were performed immediately, the tests on the pre-production prototype were performed later, after the proposed solution had already become the low-priority “deep fall back solution”.

First negative stiffness GF prototype and tests

The first negative stiffness GAS was performed in Pisa using a very small, vertical accelerometer Monolithic GAS spring coupled to a helical spring. The helical spring stiffness was the scaled equivalent of the LIGO bellow’s stiffness. The MGAS blade used was normally radially compressed 5.3% of its length to achieve neutral stiffness. The test found that the additional spring could be neutralized with a compression of 6%, well within the acceptable GAS compressional rate. This test, despite the very small scale, proved that the GAS concept could be used to generate negative stiffness.

Second negative stiffness GF prototype and tests

We then proceeded to a more representative test on a heavy payload, using spare parts from the Advanced LIGO SAS prototype.

A two-blade GAS configuration was assembled as shown in figure 5. The two 100.4 mm wide GAS blades were mounted on two sliding clamps that can be pushed towards each other by means of tuning screws. The two blades were connected to the load disk by means of two standard LIGO-SAS link-wires. A mass of 195 Kg was attached to the load disk to bring the GAS spring at its working point (a crown of twelve 110 mm wide blades would float the 1300Kg static load of each BSC pier). No effort was made to modify the blades or the links in view of different geometry, which limited the amount of achievable negative stiffness in this prototype. Although this test only produced about half of the negative stiffness needed to fully neutralize the bellows, it proved that the blades developed for the Advanced LIGO SAS prototype can be used to generate all the required lift and the negative stiffness without modification of the blade profile.

The blade's radial compression was initially adjusted to tune the GAS system to 250 mHz, corresponding to a net GAS stiffness of 480 N/m.

A single 4,700 N/m nominal stiffness helical spring was then attached above the structure, acting in parallel to the GAS spring. The vertical resonant frequency was measured and was found to be increased to 770 mHz, corresponding to an effective stiffness of 5,000 N/m, and in good agreement with the helical spring nominal stiffness.

The GAS radial compression was gradually increased to increase the negative stiffness, while monitoring the decreasing vertical resonant frequency. The blade's tip and wire inclinations were measured as well to estimate the stress applied to the links. The data is presented in figure 6.

The vertical resonant frequency of 520 mHz, corresponding to an effective stiffness of 2185 N/m, was achieved applying an additional radial compression of 4.5 mm per blade. Further radial compression would have neutralized the helical spring stiffness, but the experiment was stopped not to exceed the links stress safety limit[11]. The increase link stress derives from two facts, first negative stiffness requires stronger radial compressional forces than neutral stiffness, and more importantly, in a negative stiffness GAS spring the forces applied on the links have a different orientation, almost 20^{o} flatter than in a neutral stiffness GAS configuration. The angular mismatch between the blade's tips hooking point and the force direction generates an excessive stress on the link flex joint.

The data acquired illustrated the good behavior of the negative stiffness GAS scheme, predicting that a radial compression of 11-12 mm would fully neutralize the LIGO bellow stiffness. Both the load disk and the blade tip hooking points needed\\ to be suitably reoriented and the links strengthened to accommodate the negative stiffness requirements, but no redesign of the blade's profile proved necessary. The measurements allowed the finalization of the design of GAS filter optimized for seismic isolation on BSC or HAM chambers.

Negative stiffness, full scale IP prototype and tests

Although there was little doubt about the IP behavior, a test was performed to verify the negative stiffness IP concept. The actual IP would be composed of three legs in a triangular configuration. A single negative stiffness IP leg was built in a test to neutralize the stiffness of a star of four 1676 N/m helical springs.

The load was increased progressively to measure the decreasing IP resonant frequency.

The results are shown in figure 7. A horizontal resonant frequency of less than 100 mHz was easily obtained with a load of 102 Kg.

The IP negative stiffness K_- calculated for this weight neglecting the second order contributions of the spring star, and the corrections due to the effective bending point of the flex joint, was

$K_- = M g / l = 102 * 9.8 / 0.31 = 3354$ N/m

is in good agreement with the nominal stiffness K_+ of the springs used

$K_+ = 2 * 1676 = 3352$.

The flex joints used in this IP prototype can lift a 1300 Kg load. A triad of these legs loaded with 1400 Kg (the 1300 Kg BSC pier load and 100 Kg for the GAS filter weight and ballast) will produce a negative stiffness K_-

$K_- = 1400 * 9.8 / 0.31 = 44{,}250$ N/m which is exactly the amount required to neutralize the LIGO bellows transversal stiffness. Precision tuning of the negative stiffness is obtained by simply piling up ballast weight on the GF body.

The resonance Q factor of the IP was observed to decrease with frequency, in agreement with what expected taking into account the internal damping of the spring materials. Q factors between 1 and 2 at 100 mHz were observed. Similar values are expected when neutralizing the bellow stiffness. This means that no resonance damping may be necessary. Horizontal LVDT and voice coil actuators, as well as tuning parasitic springs attached to micrometric stages, are foreseen in the design for fine working point tuning and tidal corrections.

The 300 g aluminum IP leg is so light in comparison with the SSS mass, that it would not even require a counter-weight for batting center correction to generate the required attenuation level. Counterweighing can still be applied to minimize the amount of residual seismic power fed through by the legs. No performance improvement may be observed though, because the horizontal attenuation performance will likely be limited at the 10^{-3}-10^{-4} level by the bellow resonances.

No further information was needed to complete the design of a first article IP and GF tower for the LIGO seismic remedial system.

Third (BSC executive design) negative stiffness GF prototype and its tests

A two blade, full scale, pre-production, negative stiffness GS prototype, designed to carry one sixth of the LIGO BSC pier load and neutralize one sixth of the LIGO bellow's transversal stiffness (6,600 N/m) was built according to the final BSC suspension design (figure 8). Test on this prototype fully validated the BSC twelve blades GAS filter executive design.

The test GAS spring was mounted in parallel to two commercial, helical 4,700 N/m springs totaling 40% more of the equivalent transversal stiffness of the LIGO bellows. The spring worked as calculated and was tuned to fully neutralize the test springs. The prototype showed angular instability when tuned to fully neutralize the parallel springs. This instability (present only when providing antistiffness beyond the requirement), was neutralized in the prototype with the help of a set of four antirotation wires. The observed angular instability, still absent at the required 6,000N/m antistiffness, is any case totally irrelevant in a real BSC system due to the fact that, when the four GFs are mounted to the SSS, this degree of freedom simply does not exist. The additional antistiffness though may be useful in other systems like the suspension of vacuum tanks in underground detectors.

This prototype was tuned down to 220 mHz both by means of the radial correction screws and parasitic springs. The hooks of the commercial springs were observed to shift, emitting well audible clicks and vibrating, during measurements, when the system moved up and down. Because of these shifts and the low frequency, the equilibrium point of the prototype changed by millimeters whenever a click was heard, and occasionally even for events with no audible noise. These shifts effectively impeded any reduction of the system vertical resonant frequency below 200 mHz in this prototype. At this frequency the beginning of the onset of critical damping and hysteresis were observed as well, just as in all GAS springs[12]. The LIGO bellows are much better clamped than the test springs used in this experiment, consequently we expect that a GF in a LIGO BSC or HAM can be tuned and operated at lower frequency both with mechanical tuning and by means of an electromagnetic springs obtained driving the vertical linear actuator with the vertical LVDT position sensor and suitable gain. The LVDT and actuator foreseen in the executive design are adequate for this use.

The achieved 220 Hz tuning corresponds already to an expected attenuation of 20 at 1 Hz and 180 at 3 Hz. Once tuning the GF resonant frequency at or below 100 mHz, passive attenuation exceeding the requirements in the required 1 to 3 Hz bandwidth would be achieved.

The tests of this prototype also allowed an important design simplification. The expensive and hard to mount links between the blade tips and the load disk (figure 8-B) can be eliminated by simply mounting the individual blade tip against a slot in the load disk and bolting it in place in a fashion very similar to the monolithic GAS springs as shown in figure 10. This option preserves the flexibility of changing individual blades to match the applied payload while achieving the simpler and sturdier geometry of the monolithic design with no loss of performance. The third GAS prototype is being converted to the new design by simply replacing its two blades and load plate. Based on experience on monolithic GAS prototypes, we expect that this two blade prototype will be able to operate without centering or anti-rotation wires even at very high values of negative stiffness. The monolithic springs have actually shown less resonances and better performance on the high frequency side; this advantage is likely not to be available in the proposed system, due to the existing 33 Hz bellow resonances. The elimination of the expensive link is expected to reduce the preattenuator cost by about 10%.

After the test of the monolithic-like spring, no further significant advance is likely without installing and testing a full system on a BSC or HAM chamber.

Conclusions

A soft and passive IP/GF pre-isolator suspension of the LIGO SSS can satisfy and exceed the LIGO remedial seismic attenuation requirements of 30 dB in the 1 to 3 Hz frequency region, to allow stable operation of the LIGO interferometer and introduce earthquake immunity up to excursions of the order of the centimeter. More aggressive, probably staged, implementation can be expected to provide additional attenuation, some of the options are listed below.

This system, intended as a cheap and temporary remedy of the present seismic attenuation deficit, was designed and tested in time to be installed just after the S2 run without opening the LIGO vacuum chambers. It was supposed to be replaced as soon as the Advanced LIGO active seismic attenuation system would become available. However this low frequency pre-isolation scheme is fully compatible with the in vacuum Ad-LIGO active seismic attenuation scheme, and could replace the other pre-isolation options.

A semi-passive upgrade with electromagnetic springs obtained coupling horizontal LVDT signals to the corresponding constant force actuators could be used to reduce the SSS suspension resonant frequencies and attenuation pole to below the micro-seismic peak. This would already reduce the mirror LF residual motion and reduce the force requirements of the mirror actuators.

The excellent low residual mirror motion achieved by Virgo at the bottom of a 6 m tall attenuation chain is due to the low frequency tuning and inertial damping applied at the top of the Virgo IP-top filter pre-attenuator. This slow mirror motion results in better beam stability and strongly reduce the mirror control system strength requirements. Building an analogous IP-GF pre-attenuator for the LIGO seismic attenuation stacks will produce the same beneficial effects. Inertial tack resonance sensing and damping using external sensors and actuators would become possible with the proposed soft suspensions and passive preattenuation of the SSS.

Active attenuation, a step above inertial damping, could be implemented as well. We expect that active attenuation implementation would be easier to implement than on a stiffer system, and could further improve over the passive attenuation performance.

A mechanical design of the first article is available in ref. [ ].

The cost of the mechanical components of the IP-GF minitowers for a LIGO BSC or HAM chamber was priced to $50,000[c]. The accelerometer sensors designed for inertial damping and readout and control electronics would cost an additional $40,000.

[c] Price based on the GAS non-monolithic design.

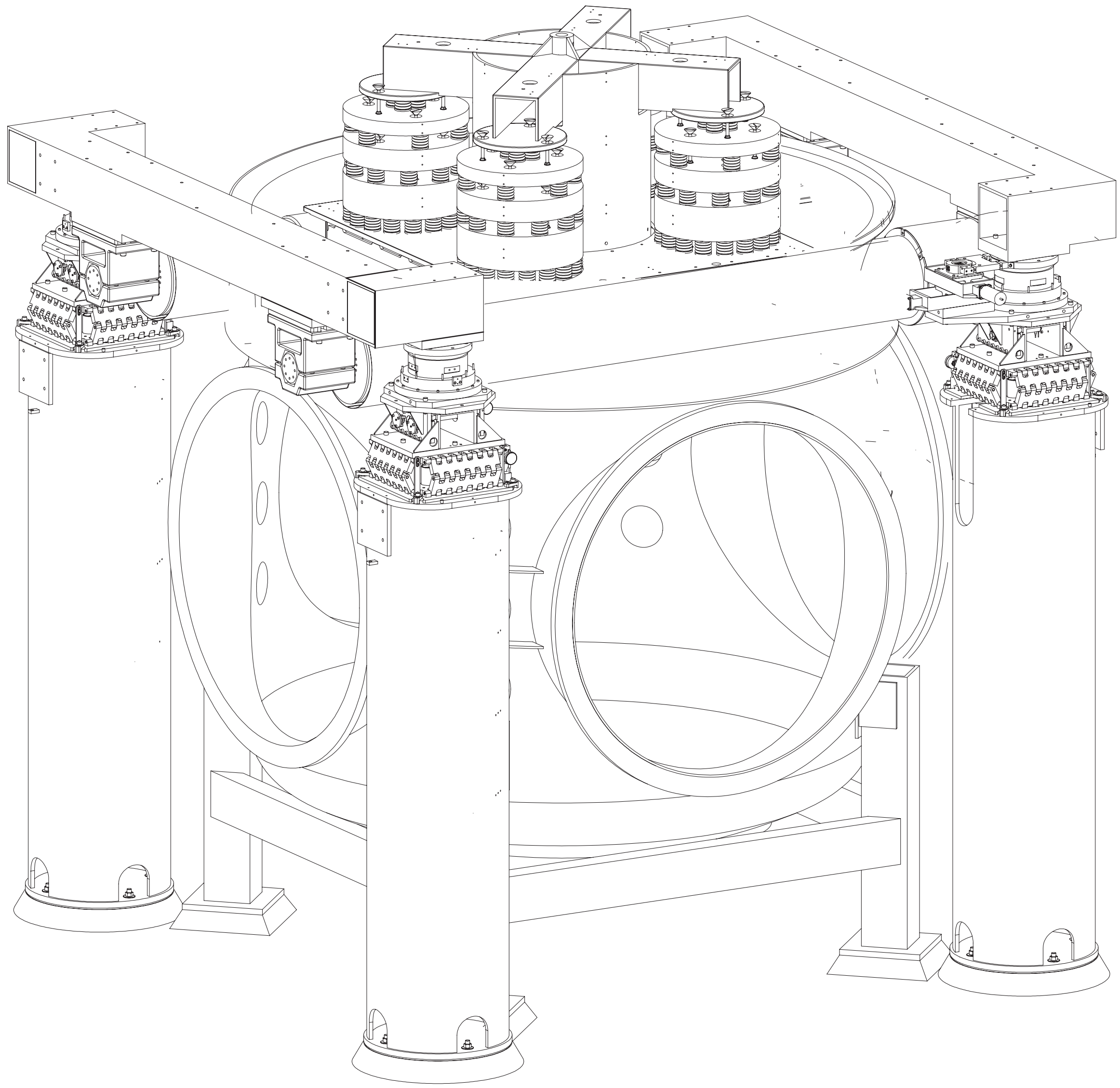

Figure 1: Topology of the LIGO seismic attenuation stacks.

Four Seismic Attenuation Stacks, SIS (1) sit on the Stack Support Structure, SSS (2,3 and 4). The SSS is a quadrilateral part composed of two pipes (2) linked together by the rectangular table (3) supporting the SIS and by two rectangular-cross-section cross beams (4) outside the vacuum. The SSS is supported on top of four piers (5) at the extremities of the two cross beams. The two pipes penetrate inside the vacuum tank (6) through four bellows (7). The SIS supports the Down Pipe, DP (8) that carries the LIGO optics. The SSS is connected to the seismic excitation through the piers and the bellows. The SSS supports (9) must be replaced with a mechanism capable to isolate the SSS from the seismic noise entering through the piers and neutralize the seismic noise entering through the bellows.

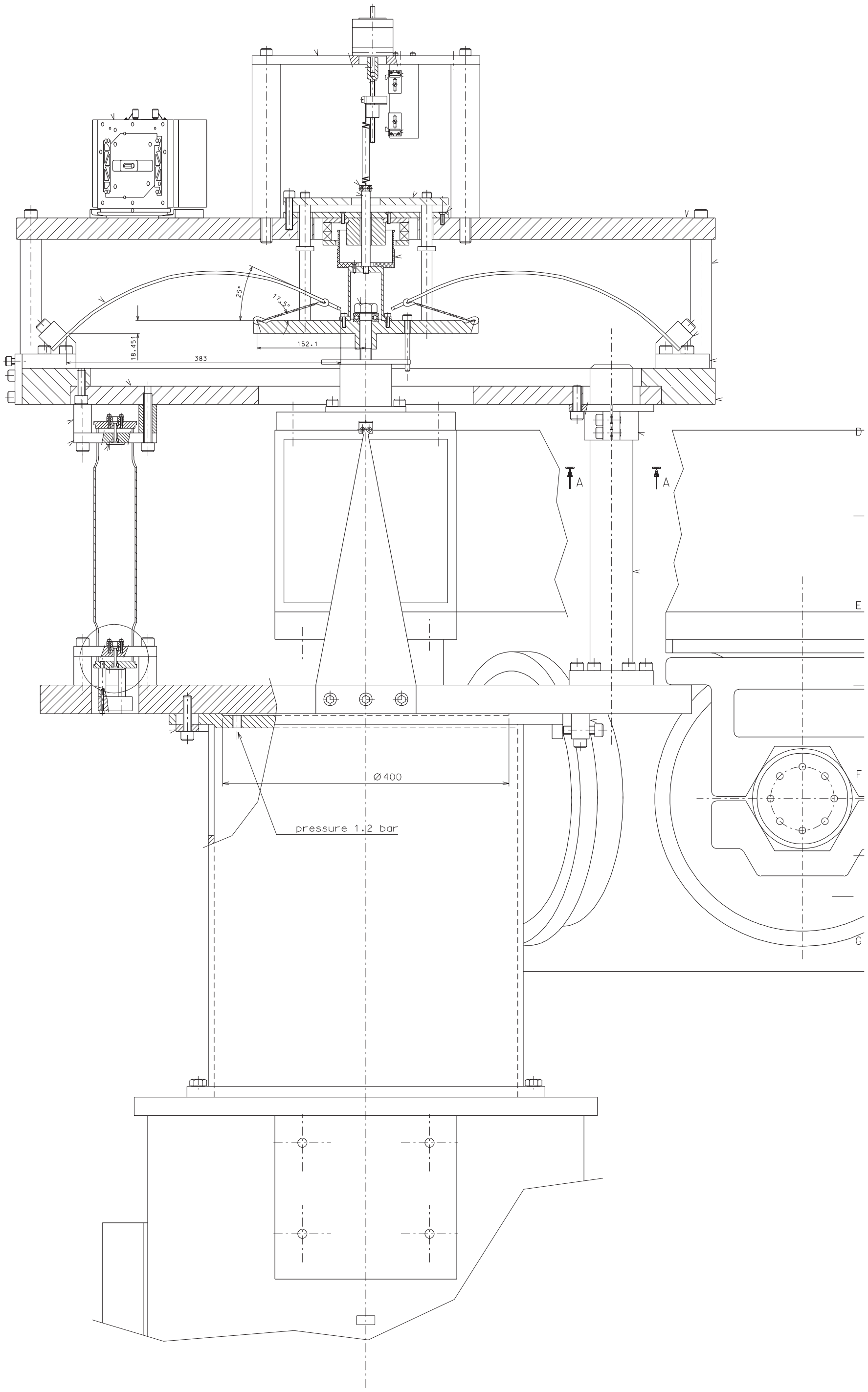

Figure 2: Negative stiffness seismic suspension system for the SSS.

It is composed of three short IP legs (1), supporting a GAS filter (2), in its turn supporting the SSS (3). Each mini-tower sits on top of one of the existing piers (4). The weight of the SSS is carried by twelve GAS blades (5) and wire links (6).

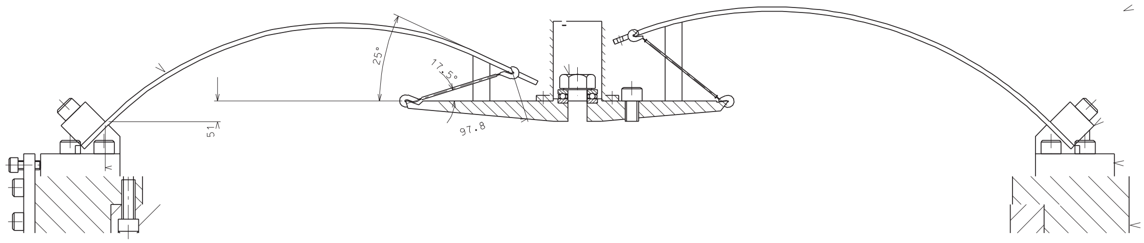

Figure 3: Comparison between a neutral stiffness GAS blade (right) and a negative stiffness one (left). The link wire in a negative stiffness works at a much lower angle to transfer the increased radial compression force while carrying the same load. The blade has almost the same curvature, indicating quite similar stress levels in both configurations.

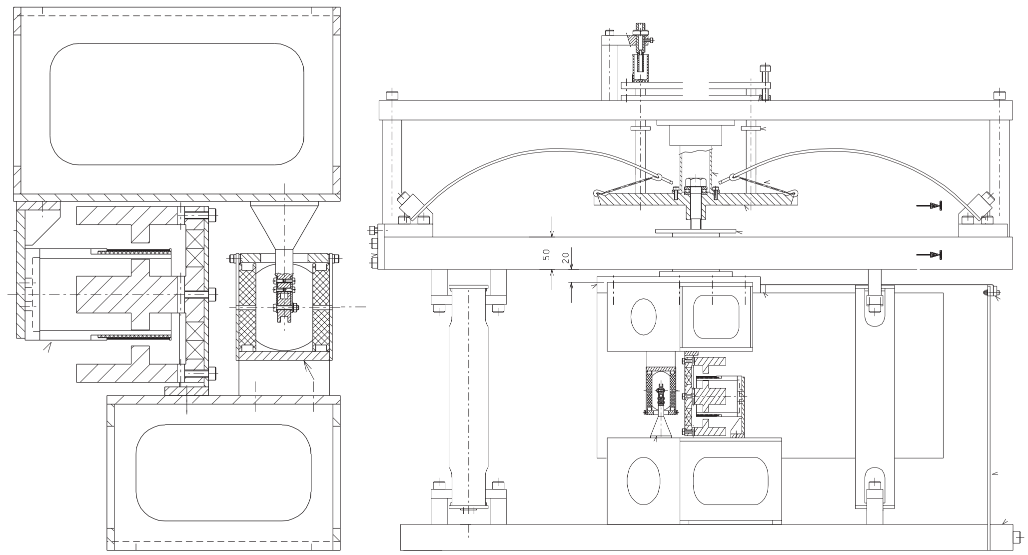

Figure 4: Co-located pair composed by a LVDT position sensor and a constant force actuator (left) and their positioning inside a mini-tower (right). The sensing/actuation axis of the pair is tangential with respect of the vacuum tank center.

For the vertical direction a smaller LVDT, visible on top of the minitower, provide the drive signal for a voice coil actuator placed coaxially with the GAS filter load disk.

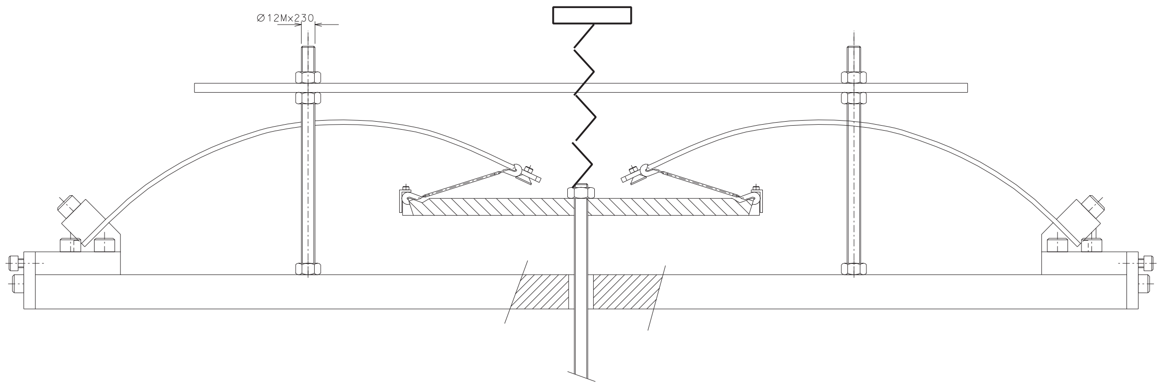

Figure 5: Test setup of negative stiffness GAS. It was built with two surplus blades and link wires from the Advanced LIGO SAS suspension prototype. A spring is attached to the load disk to provide a stiffness comparable to the stiffness of the LIGO vacuum bellows. A second prototype, built according to full scale design, was built and is shown in figure 8.

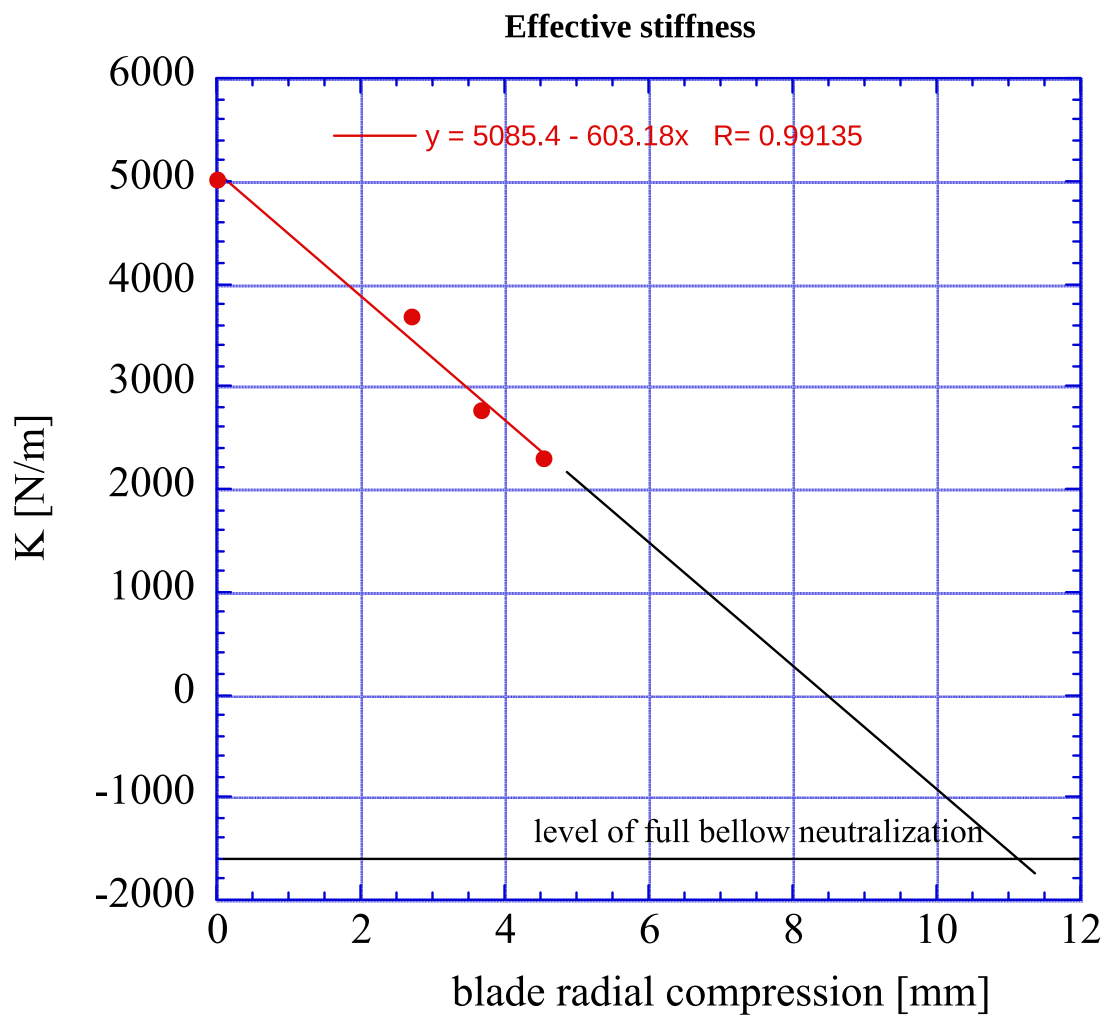

Figure 6: Effective stiffness of the GAS and test helical spring system as a function of the radial compression (in mm) of the GAS blades.

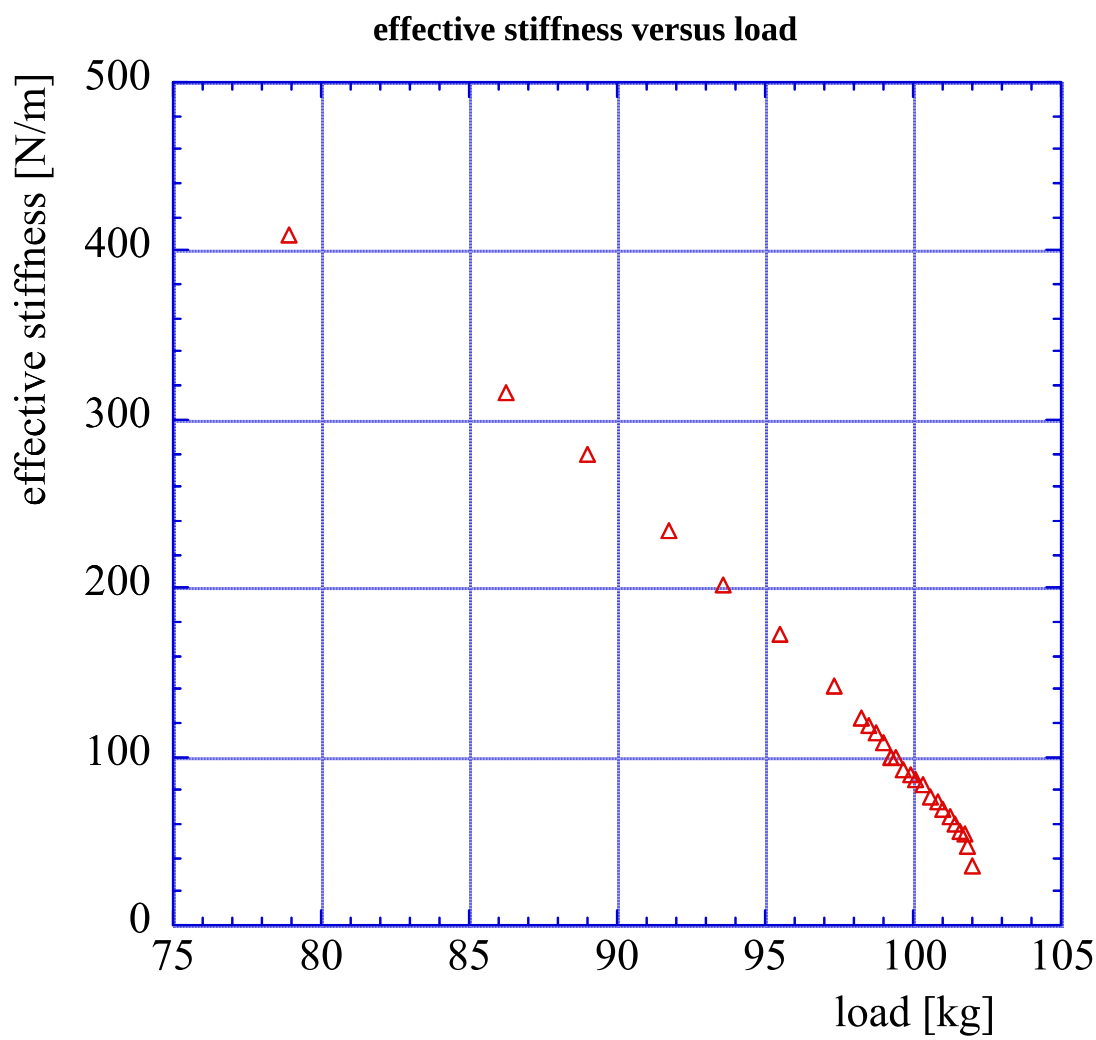

Figure 7: Effective stiffness of the IP and test helical spring as a function of load.

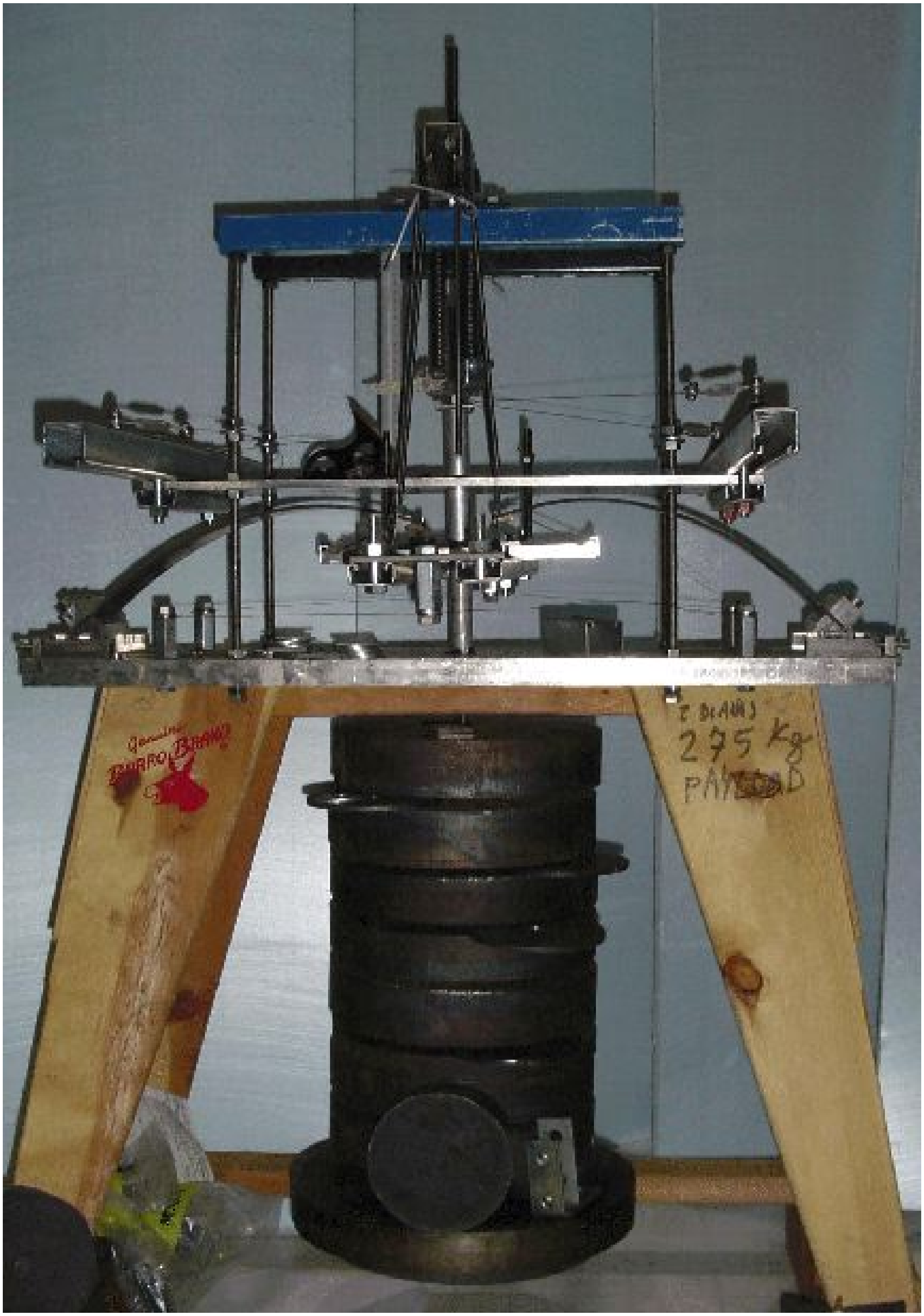

Figure 8-A: Pre-production Negative stiffness GAS prototype.

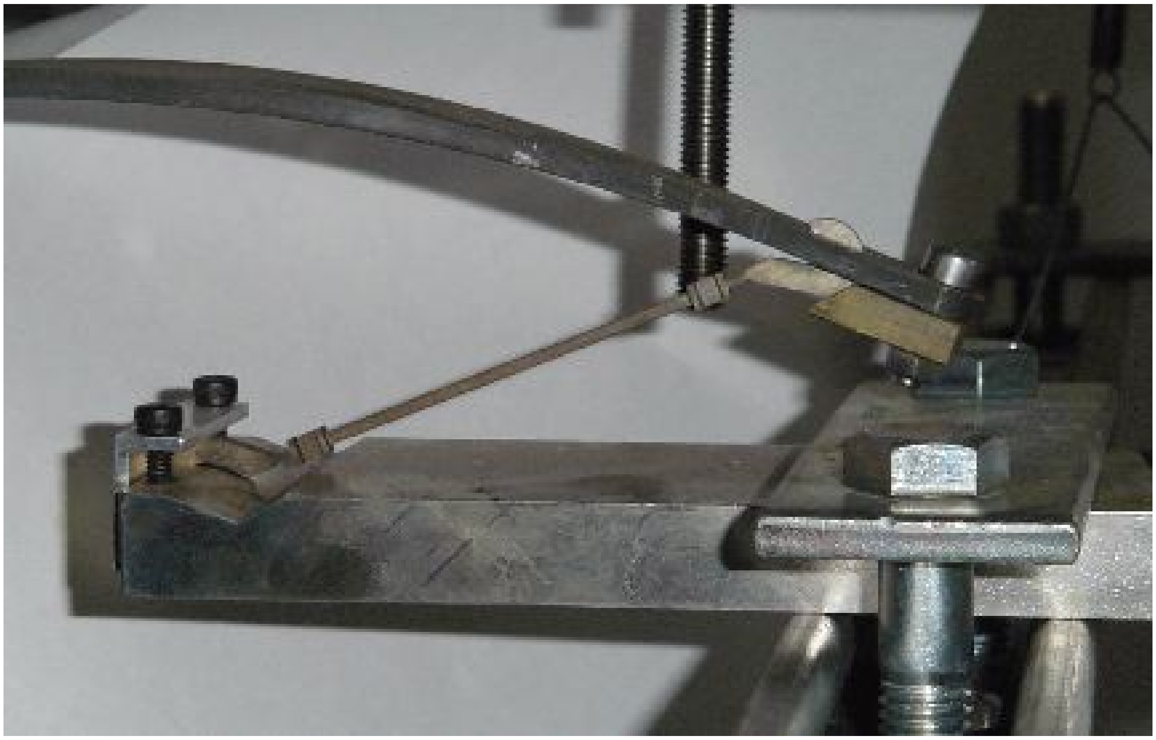

Figure 8-B: Detail of the flex joint in the negative stiffness GAS. It was found that this part, the most expensive of the assembly, can be suppressed in favor of a bolted, monolithic-like geometry.

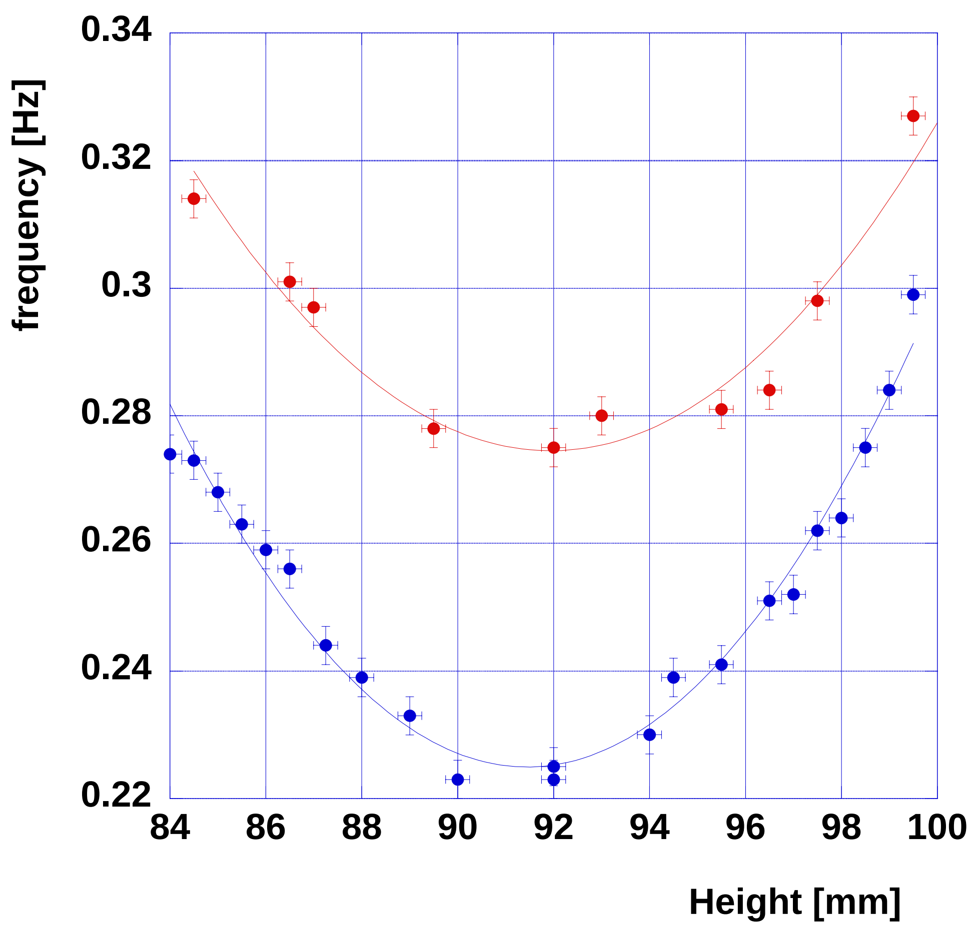

Figure 9: Tuning curves of the GF prototype. The two curves differ by 1 mm of radial compression. The error bars in length are 0.5 mm while the frequency error bar correspond to a 0.3 second error in counting oscillations with a stopwatch. The measured change is of 295 N/m, well in agreement with what 300 N/m calculated from the blade design.

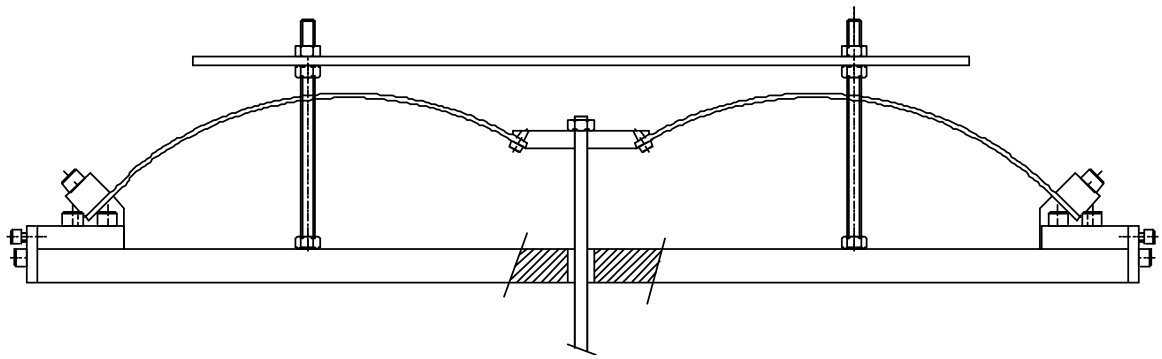

Figure 10: Simplified Monolithic-like design for the negative stiffness GAS springs to be compared with the design of figure 5.

References

1 Rai Weiss reports G-010325-00M and 1011236204

2 LIGO-T-020033-00-D

3 http://www.ligo.caltech.edu/docs/T/T960214-B.pdf
and LIGO-C970257-00-D T. Thompson et al. "PDR/CDR design review document 1997
Pier rigidity
Joe Giaime, remedial group meeting 3,8,2002.

4 LIGO- D200001-00-R

5 http://www.ligo.caltech.edu/docs/T/T980123-A.pdf

6 Hareem Tariq, LIGO-P040002, accepted for publication in NIM

7 Maddalena Mantovani, Thesis, LIGO-P040025

8 Maddalena Mantovani, LIGO-P040039, submitted for publication in NIM.

9 C. Wang, NIM A 489 (2002) 563-569

10 H. Tariq, NIM A 489 (2002) 570-576

11 LIGO-T020052

12 LIGO-P040002