European Coordination for Accelerator Research and Development

PUBLICATION

CLIC quadrupole stabilization and nano-positioning

Collette, C (CERN) *et al*

06 October 2010

The research leading to these results has received funding from the European Commission under the FP7 Research Infrastructures project EuCARD, grant agreement no. 227579.

This work is part of EuCARD Work Package **9: Technology for normal conducting higher energy linear accelerators**.

CLIC quadrupole stabilization and nano-positioning

C. Collette [1], **S. Janssens** [1], **K. Artoos** [1], **P. Carmona** [1], **M. Guinchard** [1], **C. Hauviller** [1]
[1] CERN, Engineering Department,
23 Geneva 1211, Switzerland
e-mail: **christophe.collette@cern.ch**

Abstract

In the Compact LInear Collider (CLIC) currently under study, electrons and positrons will be accelerated in two linear accelerators to collide at the interaction point with an energy of 0.5- 3 TeV. This machine is constituted of a succession of accelerating structures, used to accelerate the beams of particles, and electromagnets (quadrupoles) used to focus the beams. In order to ensure good performances, the quadrupoles have to be extremely stable. Additionally, they should also have the capability to move by steps of some tens of nanometers every 20 ms with a precision of +/- 1nm. This paper proposes a holistic approach to fulfill alternatively both requirements using the same device. The concept is based on piezoelectric hard mounts to isolate the quadrupoles from the ground vibrations in the sensitive range between 1 and 20 Hz, and to provide nano-positioning capabilities. It is also shown that this strategy ensures robustness to external forces (acoustic noise, water flow for the cooling, air flow for the ventilation) directly acting on the quadrupoles. In the first part, the strategy adopted is presented, and its advantages compared to other stabilization strategies are discussed. In the second part, experimental validations are presented on two test benches. Both set-ups consist of a compact mass, supported by piezoelectric hard mounts. Isolation and nano-positioning performances are presented, and specific issues of sensor noise and resolution are addressed.

1 Introduction

The stabilization of structures at the nanometer scale is a concern in various fields of precision engineering, like interferometers [1], microscopes [2] or manufacturing [3]. In the Compact LInear Collider (CLIC) currently under study [4], electrons and positrons will be accelerated in two linear accelerators to collide at the interaction point with an energy of $0.5 - 3\ TeV$. To acquire such a high energy, the total length of the machine should to be up to $48\ km$. This linear accelerator will consist of a succession of accelerating structures and heavy electromagnets (quadrupoles). The former are used to accelerate the particles to increase their energy; the latter are used to maintain the beam inside the vacuum chamber (alternating gradient) and to reach the required luminosity at the collision point. However, any oscillation of one quadrupole deflects the beam, and reduces the luminosity. More precisely, if $\Phi_x(f)$ is the power spectral density of the vertical displacement of the quadrupole, it has been estimated that the integrated Root Mean Square (RMS) $\sigma_x(f)$, defined as

$$\sigma_x(f) = \sqrt{\int_f^{\infty} \Phi_x(\nu) d\nu} \tag{1}$$

must stay below $1\ nm$ [5] above $1\ Hz$ to ensure sufficient performances. Similarly, it must stay below $5\ nm$ in the lateral direction. This concerns about 2000 quadrupoles per beam line. Figure 1(a) shows the typical power spectral density in the LHC tunnel, $\Phi_w(f)$, and the power spectral density of the measurement noise, $\Phi_n(f)$. Figure 1(b) shows the corresponding RMS integrated curves. One sees on this figure that, in order to reach the objective of stability, one needs to design a support that reduces the ampltitude of the ground motion by roughly a factor four in the frequency range between 1 and 20 Hz. Below 1 Hz, the ground motion corresponds essentially to coherent (micro-seismic) waves propagating on the surface of the terrestrial crest; above 20 Hz, the ground motion does not provide any significant contribution to the integrated RMS at 1 Hz.

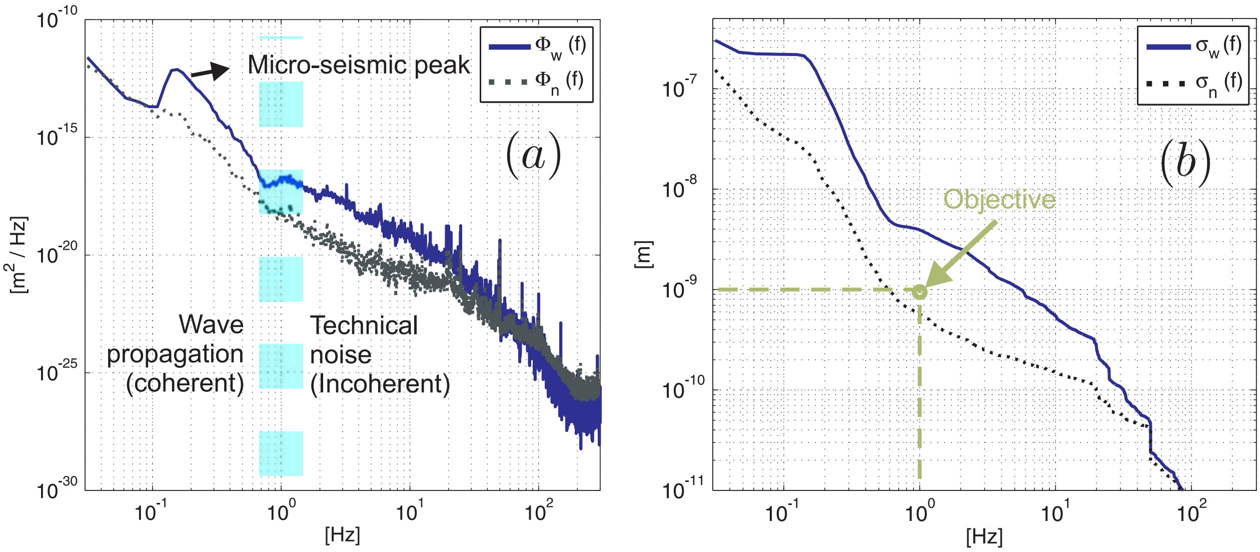

Figure 1: Typical power spectral density of the ground vibration in the LHC tunnel, and power spectral density of the measurement noise; (b) Integrated RMS vertical displacement in the LHC tunnel calculated using Equ.(1); Integrated RMS of the sensor noise, and objective of the stabilisation.

In addition to this requirement of stability, the support strategy has to fulfil some other requirements:

- About 80 of these quadrupoles should have the capability to move by steps of some tens of nanometers every 20 ms [6], with a precision of $+/-\ 1nm$.
- The size of the tunnel is very restricted, and the space available for the mounts should not exceed a height of 15 cm.
- The direct environment of the future CLIC collider is subjected to radiations and stray magnetic fields. In order to ensure a full compatibility, this requirement excludes the use of electromagnetic equipments (electromagnetic actuators and sensors using coils likes commercial seismometers).
- In operating conditions, the quadrupoles are also subjected to several types of disturbances, commonly referred to as technical noise: acoustic noise, cooling system, ventilation. The supports should accordingly ensure a sufficient robustness to the external forces generated by these disturbances.
- The support should allow a temperature change of about 25 K during transients. Features should be implemented to protect the actuating system during transport and handling.

To demonstrate the feasibility to fulfil such stringent requirements, it is planned to build a mock-up of the longest CLIC main beam quadrupole on its support. This paper shows experimental results obtained on two simplified test benches, constituting intermediate steps toward the feasibility demonstration. The paper is organized as follows. Section two presents the control strategy adopted to stabilize the quadrupole. Section three presents an experimental validation of the stabilization strategy on a scaled test bench, and on a full scale test bench. Section four draws the conclusions.

2 Control strategy

2.1 Literature review

In a passive suspension constituted by a simple sping k in parallel with a dashpot c, an overshoot appears on the transmissibility between the ground excitation and the payload response at a frequency corresponding to the resonance of the payload on the suspension stiffness (see e.g. the black dashed curve in Fig.2a, tuned to have a resonance at 20 Hz). Then, for higher frequencies, the curve is decreasing, and the isolation (transmissibility below 1) starts at $\sqrt{2}f_n$ (where $2\pi f_n = \sqrt{k/m}$).

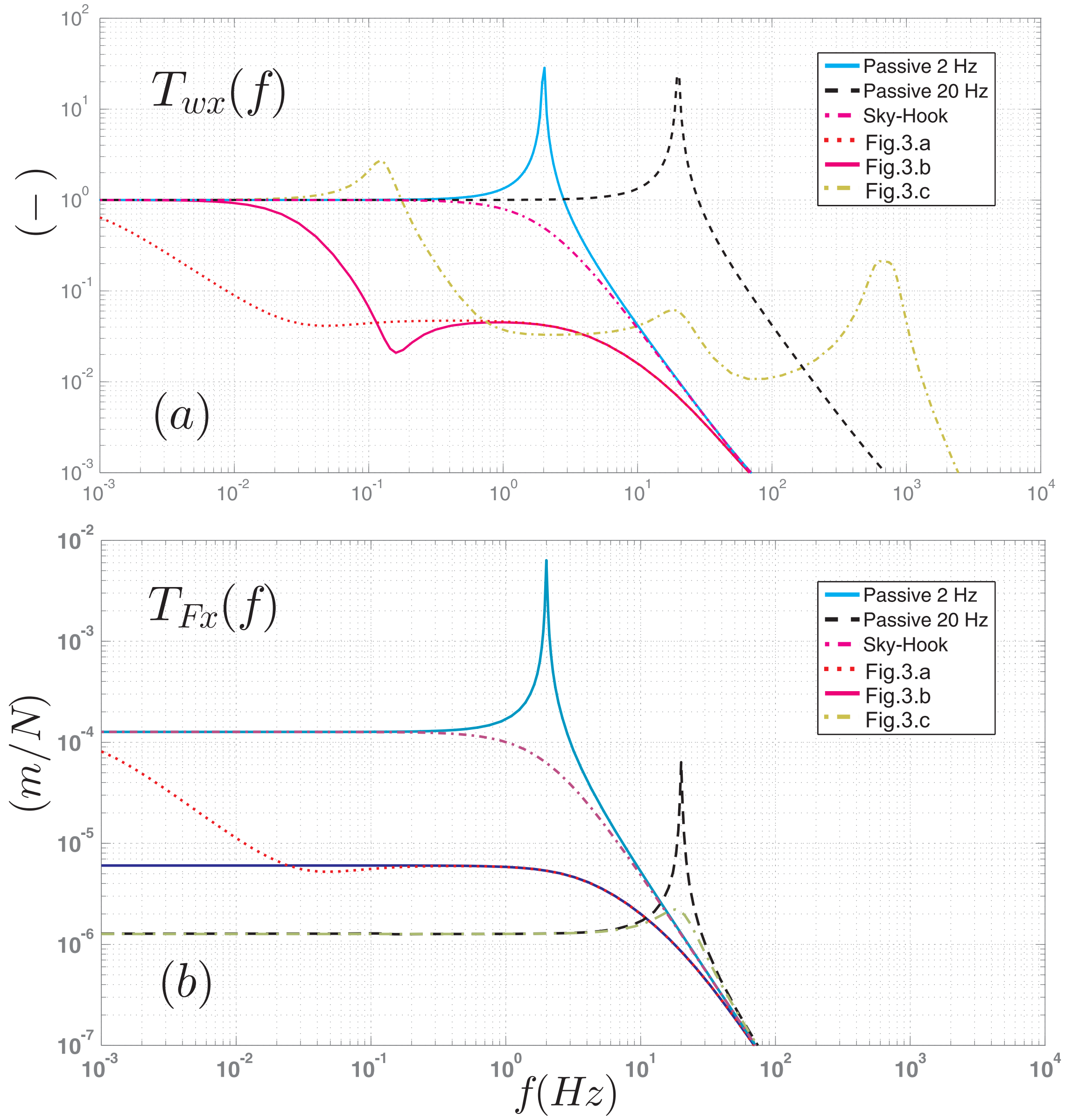

Figure 2: Various passive and active control strategies. (a) Transmissibilities $T_{wx}(f)$ between the ground and the quadrupole; (b) Compliance $T_{Fx}(f)$ between a force applied on the quadrupole and the resulting displacement.

In order to increase the passive isolation, the first idea is to reduce the value of f_n as much as possible [7, 8]. However, as the resonance frequency of the system decreases (e.g. resonance tuned at 2 Hz in Fig.2a), it also becomes rapidly unacceptably sensitive to any external force directly applied on the quadrupole at very low frequency, and especially at the resonance frequency. This is illustrated in Fig.2(b), showing the transmissibility (compliance) between a force applied on the quadrupole and its vertical displacement x. This is a reason why we are restricted to active isolation. A velocity feedback (so called *sky-hook damper* [9]) can remove the overshoot at the resonance (Fig.2), but does not reduce the compliance at very low frequency (Fig.2b). Figure 3 shows three other configurations of active isolation strategies. The first two ones are based on the use of a reference mass, mounted either on the payload (Fig.3a [10, 11]) or directly on the ground (Fig.3b [12, 13, 14, 15]). The feedback law is based on the measurement of the relative displacement between the reference mass and the payload. The transmissibilities are also shown in Fig.2a. At very low frequency, and for comparable plant, the strategy (b) has the advantage over (a) to be more robust to external forces (see Fig.2b).

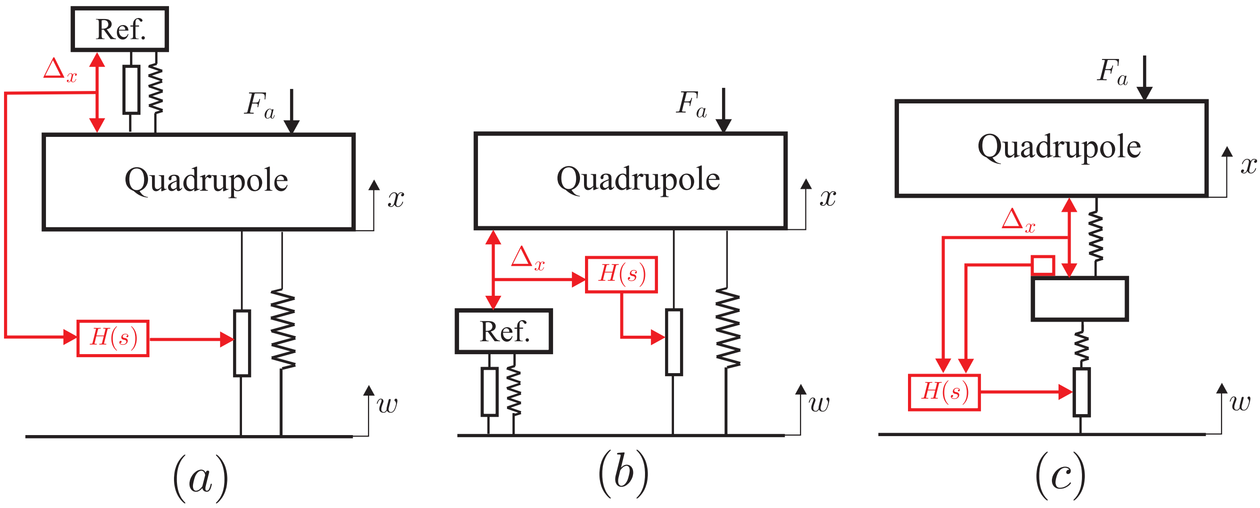

Figure 3: Three strategies used for active vibration isolation.

The third strategy (Fig.3c [16, 17, 18]) is an example of two-stages active mount. It is constituted of an intermediate mass mounted on a stiff piezoelectric stack, and in series with an elastomeric layer. In this case, the feedback is based on the combination of a measurement of the relative displacement between the two masses and the absolute velocity of the intermediate mass. The typical transmissibility and compliance are also shown in Fig.2. This strategy has already been used in previous studies on quadrupole stabilization [19, 20]. While efficient for the stabilization of quadrupoles, these systems are still much too soft to fulfil the positioning requirements. For this reason, it has been decided to use the strategy (a), using a geophone for the reference mass, stiff piezoelectric actuators, like in [21, 22], and reduce actively the transmissibility $T_{wx}(f)$ at low frequencies (mainly the range between 1 Hz and 20 Hz) [23]. Many details of the strategy can be found in [24].

2.2 Quadrupole dynamics

The strategy adopted to support the quadrupole is inspired from the concept of a Stewart platform [25, 26, 27]. It is a well known concept that has been applied for both vibration isolation and precise positioning of ground and space structures. To tackle with the large length of the quadrupole, the six legs are mounted as depicted in Fig.4. The orientations of the legs result from a tradeoff between the following requirements: provide a good stability in the longitudinal direction, manoeuvrability in both vertical and lateral directions, allow a sufficient resolution in the vertical direction, and ensure a static equilibrium when no control is applied. Assuming that the quadrupole is rigid, the dynamic equations of the system are

$$M\ddot{\mathbf{x}} = \mathbf{F} \tag{2}$$

where M =diag$(m, m, m, I_\theta, I_\phi, I_\psi)$ is the mass matrix, $\mathbf{x} = (x, y, z, \theta, \phi, \psi)$ is the vector describing small displacements of the quadrupole, and $\mathbf{F}$ is the vector of resulting forces and torques applied by the legs on the quadrupole. $\mathbf{F}$ is related to the axial forces in each leg by

$$\mathbf{F} = B\mathbf{f} \tag{3}$$

where $\mathbf{f} = (f_1, f_2, ..., f_6)^T$ is the vector of active control forces in the six legs and B the force jacobian matrix. Assuming that there is no damping in the legs, f_i is given by

$$f_i = k_a(-q_i + \Delta_i + w_i^l) \tag{4}$$

where k_a is the axial stiffness of each leg, q_i and w_i^l are respectively the displacement of the quadrupole and the ground in the direction of the leg. Δ_i is the elongation of the leg due to a voltage V_i applied to the piezoelectric stack

$$\Delta_i = nd_{33}V_i \tag{5}$$

where nd_{33} is a characteristic of the actuator.

Replacing (3) and (4) in (2) gives

$$M\ddot{\mathbf{x}} + K\mathbf{x} = k_aB\boldsymbol{\Delta} + k_aB\mathbf{w}^l \tag{6}$$

or again

$$M\ddot{\mathbf{x}} + K\mathbf{x} = k_aB\boldsymbol{\Delta} + k_aBE\mathbf{w} \tag{7}$$

where $K = k_aBB^T$ is the stiffness matrix, $\mathbf{w}$ and $\mathbf{w}^l$ are the ground excitation vector and the ground excitation vector in the legs, and E is the excitation matrix projecting $\mathbf{w}$ in the directions of the legs.

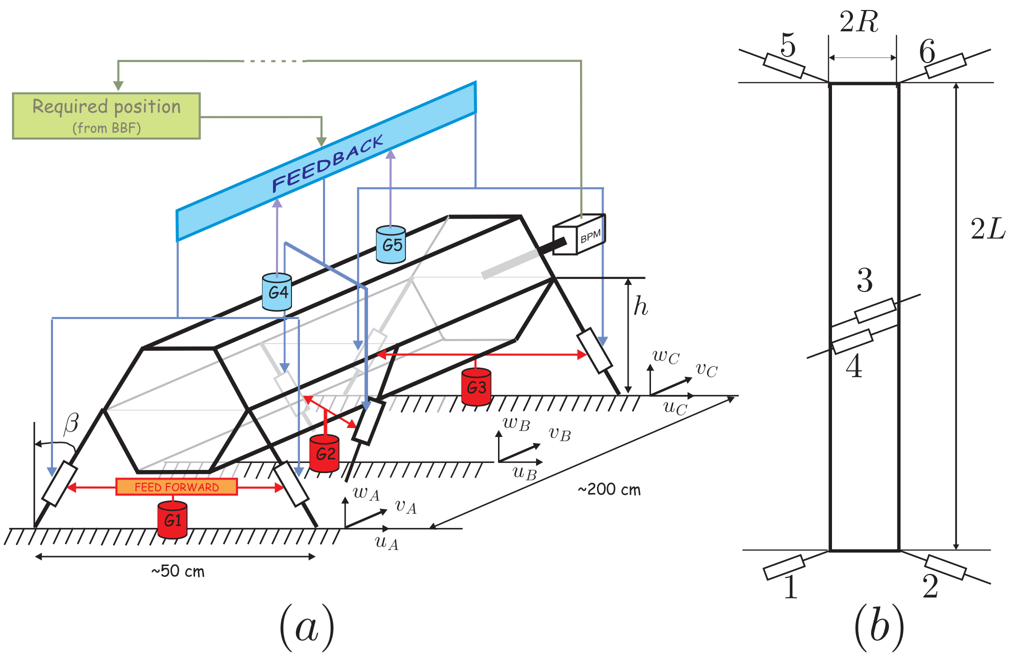

Figure 4: Simplified drawing of the quadrupole: (a) Perspective view; (b) Top view.

Let J be the Jacobian matrix relating the elongations velocities of the legs $\dot{\mathbf{q}}$ and the velocity vector $\dot{\mathbf{x}}$ as $\dot{\mathbf{q}} = J\dot{\mathbf{x}}$. According to the virtual work principle, we have

$$\mathbf{F}^T\delta\mathbf{x} = \mathbf{f}^T\delta\mathbf{q} = \mathbf{f}^TJ\delta\mathbf{x} \tag{8}$$

After identification, we have $\mathbf{F} = J^T\mathbf{f}$. Comparing with (3) leads to $B = J^T$.

The analytical expression of J is found as follows. First, let us split the velocity vector $\dot{\mathbf{x}}$ into translational and rotational components such as $\dot{\mathbf{x}}^T = (\mathbf{v}^T, \omega^T)$ where $\mathbf{v}^T = (\dot{x}, \dot{y}, \dot{z})$ and $\omega^T = (\dot{\theta}, \dot{\phi}, \dot{\psi})$. Then, the velocity of the fixation point of leg i is

$$\mathbf{v}_i = \mathbf{v} + \omega \times \mathbf{p}_i \tag{9}$$

where $\mathbf{p}_i$ is the coordinate of the extremity of leg i in the reference frame fixed on the quadrupole. If $\mathbf{1}_i$ is a unit vector in the direction of leg i, the velocity of the extension of the leg is obtained by projecting $\mathbf{v}_i$ along $\mathbf{1}_i$

$$\dot{q}_i = \mathbf{1}_i^T \mathbf{v}_i = \mathbf{1}_i^T(\mathbf{v} + \omega \times \mathbf{p}_i) \tag{10}$$

or

$$\dot{q}_i = \mathbf{1}_i^T \mathbf{v}_i = \mathbf{1}_i^T(\mathbf{v} - \mathbf{p}_i \times \omega) \tag{11}$$

Proceeding the same way for each leg, we have finally

$$J = \begin{pmatrix} \dots & \dots \\ \mathbf{1}_i^T & -\mathbf{1}_i^T \tilde{\mathbf{p}}_i \\ \dots & \dots \end{pmatrix} \tag{12}$$

where $\tilde{\mathbf{p}}_i$ is the antisymmetric matrix calculated from $\mathbf{p}_i$ to express the cross product.

2.3 Quadrupole stabilization

After integrating the lateral and vertical ground velocities measured by the geophones (G4 and G5 in Fig.4), there is a linear relationship between the sensor output and the coordinates of the quadrupole

$$\mathbf{y} = C\mathbf{x} = CJ^{-1}\mathbf{q} \tag{13}$$

where $\mathbf{y}$ is the measurement vector, C a matrix containing the sensor dynamics. Then, the forces exerted by the actuators on the quadrupoles are

$$\mathbf{F} = k_a B \mathbf{\Delta} = k_a B H(f) C^{-1} J \mathbf{y} \tag{14}$$

where $H(f)$ is the controller.

3 Experimental validations

3.1 Scaled test bench

In order to test experimentally the stabilization strategy described above, a scaled test bench has been designed to represent one sixth of the quadrupole mounted on one active leg. The experimental setup consists of a guided piezoelectric stack, clamped in a double membrane like structure to allow only a vertical motion. Two geophones are used to measure the vibrations at both ends of the actuator. The aim of the experiment is to stabilize a small mass laying on the top of the membrane, i.e. the geophone itself (Fig.5a).

It is located in a tunnel where the amplitude of the ground motion is similar to the values measured in the LHC tunnel. The nano-positioning capability of the leg has been tested with a square command, with an amplitude of 10 nm and a frequency of 50 Hz. On sees in Fig.5b that the actuator follows the request very

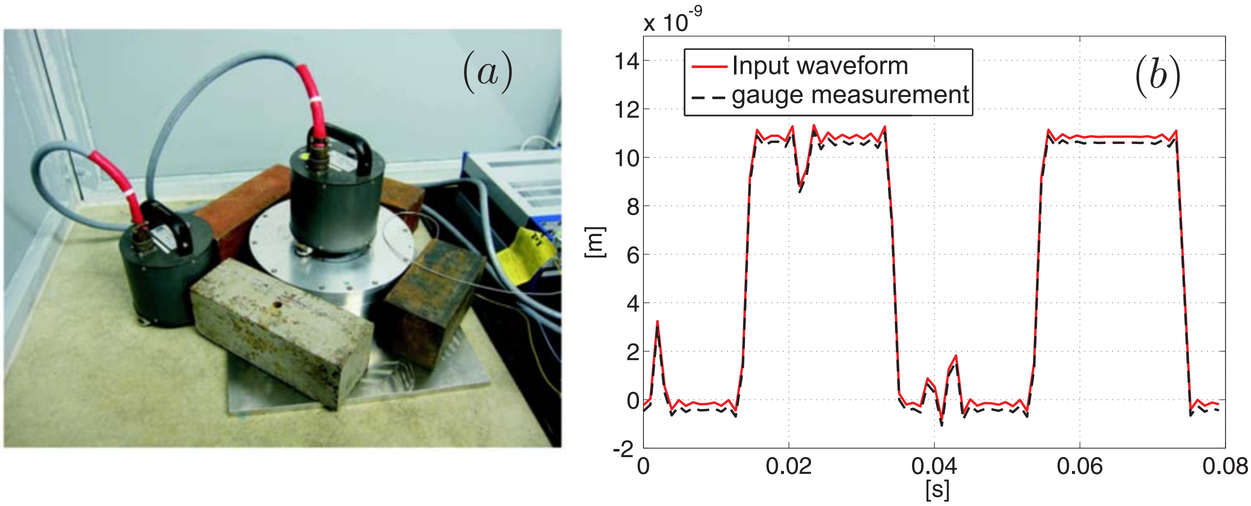

Figure 5: (a) Picture of the scaled test bench; (b) Nano-positioning experiment.

well. For the stabilization of the top geophone, the controller is based on the measurement of the absolute velocity and is designed as follows: a proportional gain, a Butterworth high pass filter at $0.5\ Hz$, introduced in the controller to remove the drift in the signals, and a lag at $30\ Hz$ to improve the stability. Figure 6 shows the RMS integrated displacement of the top geophone for two experiments, one performed during the day and one performed during the night. During the day, at $1\ Hz$, one sees that the feedback control has reduced $\sigma_x(f)$ from $4.7\ nm$ to $1.4\ nm$, i.e. a reduction by a factor 3.5. During the night, when the ground motion is even lower, σ_x is reduced from $1.8\ nm$ to $1\ nm$ at $1\ Hz$.

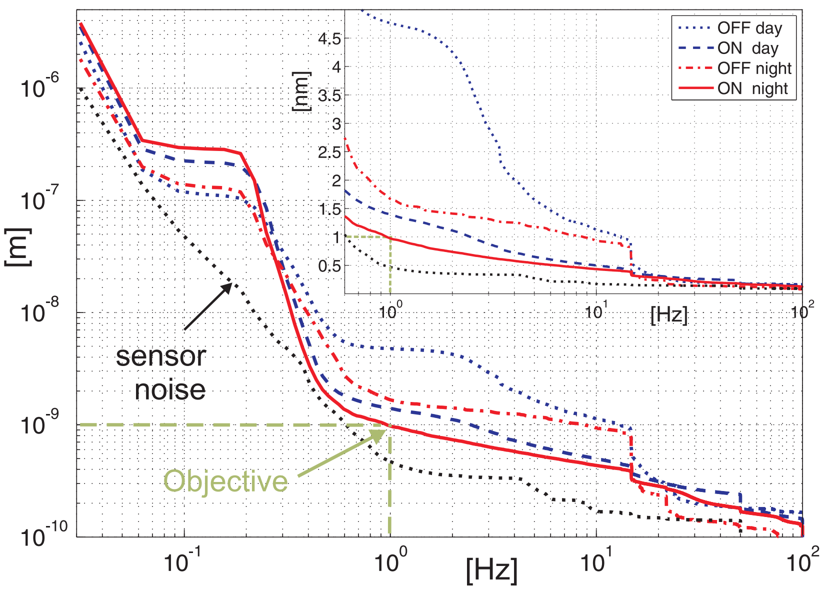

Figure 6: Comparison of RMS integrated of the top displacement $\sigma_x(f)$ when the controller is ON and OFF, during the day and during the night.

Although this value corresponds to the requirements, the controller still needs to be improved to reach this required value during the day, i.e. in conditions similar to a realistic, active accelerator environment. Better results are anyway expected from an optimized combination of the feedback and feed forward, and a more adapted hardware. As an intermediate step between this extremely simple, scaled test bench, and the full scale quadrupole (400 Kg weight), it has been decided to construct a full scale setup, in order to address specific issues. This bench is briefly presented in the next section, along with preliminary results.

3.2 Full scale test bench

The test bench presented in this section is a tripod, inspired from [21, 22], but designed to be modular, in order to be able to address a certain number of difficulties including: the stabilization in both vertical and lateral direction, the nano-positioning in both vertical and lateral direction, mounting, jointure and guide design. Figure 7a shows a picture of the test bench. It is constituted of a compact mass of 100 Kg, supported by three legs, two passive mounts and one piezoelectric stack actuator.

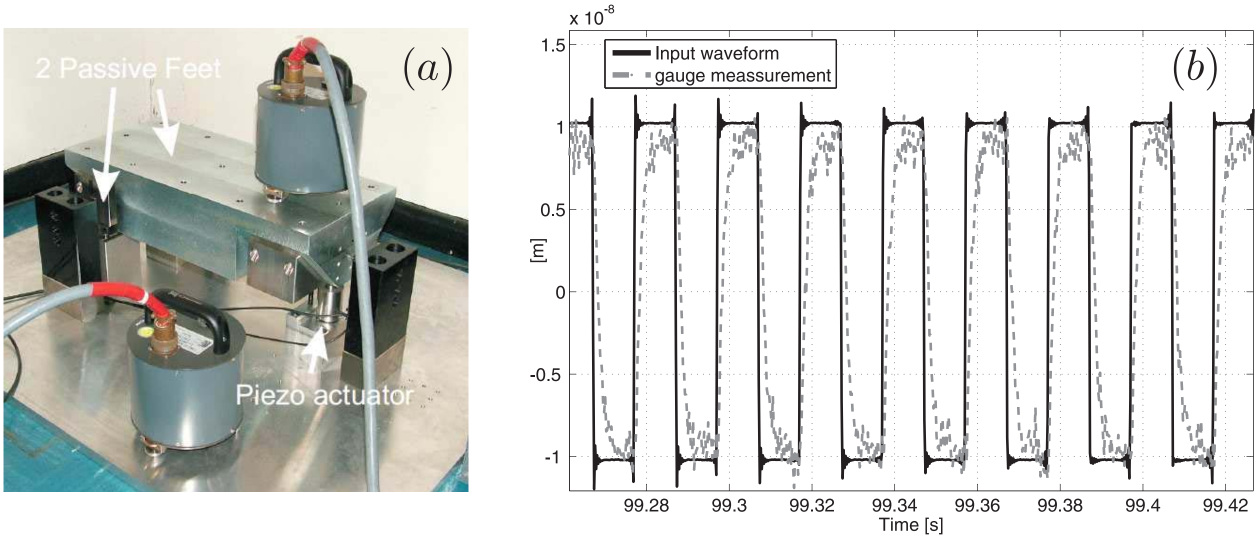

Figure 7: Full scale test bench (tripod): (a) Picture of the tripod; (b) Time history of the actuator stroke resulting from a square wave command.

The nano-positioning capability of the active support is presented in Fig.7b. The stabilization performances are similar to those obtained with the scaled test bench, but have not yet been tested in a quiet environment.

4 Conclusions and future work

First of all, a concept of six legs has been presented to support the quadrupole in order to fulfill alternatively the different requirements. Then, the strategy has been tested experimentally on two single d.o.f. set-ups. It has been shown that both can fulfill the nano-positioning requirements.

It has also been possible to stabilize the scaled set-up at the requested value, when it is placed in a quiet environment. The next steps are to test the control strategy, in both vertical and lateral direction, on the full scale set-up. Then, the stabilization of a long dummy quadrupole will be addressed.

Acknowledgements

The research leading to these results has received funding from the European Commission under the FP7 Research Infrastructures project EuCARD, grant agreement No.227579.

References

[1] A. Stochino, B. Abbot, Y. Aso, M. Barton, A. Bertolini, V. Boschi, D. Coyne, R. DeSalvo, C. Galli, Y. Huang, A. Ivanov, S. Marka, D. Ottaway, V. Sannibale, C. Vanni, H. Yamamoto, and S. Yoshida. *The Seismic Attenuation System (SAS) for the advanced LIGO gravitational wave interferometric detectors*. Nuclear Instruments and Methods in Physics Research A vol.598 (2009), pp. 737–753.

[2] A. Carter, G. King, T. Ulrich, W. Hasley, D. Alchenberger, and T. Perkins. *Stabilization of an optical microscope to 0.1 nm in the three dimensions*. Applied Optics Vol.46, No.3 (2007), pp. 46(3):421–427.

[3] K. Furutani, M. Suzuki, and R. Kudoh. *Nanometer cutting machine using a Stewart-platform parallel mechanism*. Measurement Science and Technology, Vol. 15 (2004), pp. 467–474.

[4] G. Riddone, D. Schulte, H. Mainaud-Durand, I. Syratchev, W. Wuensch, R. Zennaro, R. Nousiainen, and A. Samoshkin. *Technical specification for the CLIC two-beam module*. In *Proceedings of the EPAC 08, Genoa (Italy)* (2008).

[5] http://clic-stability.web.cern.ch/clic-stability/.

[6] D. Schulte. Beam based alignment in the new clic main linac. In *Proceedings of the 2009 Particle Accelerator Conference, Vancouver, Canada* (2009).

[7] S. Allison, L. Eriksson, L. Hendrickson, T. Himel, and A. Seryi. *Active vibration suppression r+d for the next linear collider*. In *Proceedings of the Particle Accelerator Conference, Chicago* (2001).

[8] J. Frisch, A. Chang, V. Decker, E. Doyle, L. Eriksson, L. Hendrickson, T. Himel, T. Markiewicz, R. Partidge, and A. Seryi. *Vibration stabilization of a mechanical model of a x-band linear collider final focul magnet*. In *22nd International Linear Collider Conference, 16-18 August (Germany)* (2004).

[9] D. Karnopp, M.J. Crosby, and R.A. Harwood. *Vibration control using semi-active force generators*. Engineering for Industry Journal, Vol. 96 (1974), pp. 619626.

[10] P.R. Saulson. *Vibration isolation for broadband gravitational wave antennas*. Review of Scientific Instruments, Vol. 55, No. 8 (1984), pp. 1315–1320.

[11] P.G. Nelson. *An active vibration isolation system for inertial reference and precision measurement*. Review of Scientific Instruments, Vol. 62, No. 9 (1991), pp. 2069–2075.

[12] M.J. Vervoordeldonk, T.A.M. Ruijl, and R.M.G. Rijs. *Development of a novel active isolation concept*. ASPE Spring Topical Meeting (2004).

[13] M.J. Vervoordeldonk and H. Stoutjesdijk. *Recent developments, a novel active isolation concept*. 6th euspen International Conference, Baden bei Wien (2006).

[14] M.J. Vervoordeldonk, T.A.M. Ruijl, R.M.G. Rijs, and J.C.A. Muller. *Actuator arrangemfor active vibration isolation comprising an inertial reference mass*. Technical Report Patent Number: US 2007/0035074 A1, United States Patent (2007).

[15] K. Kar-Leung Miu. *A Low cost, DC-Coupled Active Vibration System*. PhD thesis, Massachusetts Institute of Technology, September (2008).

[16] C.R. Fuller and A.H. von Flotow. *Active control of sound and vibration*. *IEEE*, 0272-1708 (1995).

[17] D.W. Schubert, A.M. Beard, and A.H. von Flotow. *A practical product implementation of an active/passive vibration isolation*. *SPIE*, 2264 (1994).

[18] D.W. Schubert, A.M. Beard, S.F. Shedd, M.R. Earles Jr., and A.H. Von Flotow. *Stiff actuator active vibration isolation system*. Technical Report Patent Number: 5,823,307, United States Patent, (1997).

[19] S. Redaelli. *Stabilization of Nanometer-Size Paricle Beams in the Final focus System of the Compact LInear Collider CLIC*. PhD thesis, University of Lausanne (2004).

[20] B. Bolzon. *Etude des vibrations et de la stabilisation l'échelle sous-nanométreique des doublets finaux d'un collisionneur linéaire*. PhD thesis, University of Savoie (2007).

[21] C. Montag. *Active stabilization of mechanical quadrupole vibrations for linear collider*. Nuclear Instruments and Methods in Physics Research A,Vol. 378 (1996), pp. 396–375.

[22] C. Montag. *Active Stabilization of Mechanical Quadrupole Vibrations in a Linear Collider Test Facility*. PhD thesis, Hamburg University (1996).

[23] K. Artoos, O. Capatina, C. Collette, M. Guinchard, C. Hauviller, F. Lackner, J. Pfingstner, H. Schmickler, M. Sylte, M. Fontaine, B. Bolzon, L. Brunetti, G. Deleglise, N. Geoffroy, and A. Jeremie. *Study of the stabilization to the nanometre level of mechanical vibrations of the CLIC main beam quadrupoles*. In *Proceedings of the 2009 Particle Accelerator Conference*, Vancouver, Canada (2009).

[24] C. Collette, K. Artoos, A. Kuzmin, S. Janssens, M. Sylte, M. Guinchard, and C. Hauviller. *Active quadrupole stabilization for future linear particle colliders*, *Nuclear Instruments and Methods in Physics Research A* (2010) In press, available online.

[25] J. Spanos, Z. Rahman, and G. Blackwood. *A soft 6-axis active vibration isolator*. In *Proceedings of the American Control Conference* Seatle, Washington (1995).

[26] A. Hanieh. *Active Isolation and damping of Vibrations via Stewart Plateform*. PhD thesis, University of Brussels (2004).

[27] A. Preumont. *Vibration Control of Active Structures An Introduction (2nd edition)*. Kluwer Academic Publishers, Dordrecht (2002).