Stabilization and nano-positioning of the CLIC Main-Beam Quadrupoles

S. Janssens [1], K. Artoos [1], C. Collette [2], M. Esposito [1], P. Fernandez Carmona [1], M. Guinchard [1], R. Leuxe [1]
[1] CERN, Engineering Department,
1211 Geneva 23, Switzerland
e-mail: **Stef.marten.johan.Janssens@cern.ch**

[2] Université Libre de Bruxelles, Active Structures Laboratory,
avenue F.D.Roosevelt 50, B-1050 Brussels, Belgium

Abstract
The Compact Linear Collider (CLIC), currently under study, aims to accelerate electrons and positrons towards the interaction point where they collide with a centre-of-mass energy of 0.5 to 3 TeV. In the main accelerator line, Main-Beam Quadrupole (MBQ) magnets are used to focus the beams. To have the best performance, these magnets need to be isolated from vibrations coming from the ground and external sources (water cooling, ventilation, etc.). The required integrated root mean square (r.m.s.) values of the vibrations down to 1 Hz are 1.5 nm vertically and 5 nm horizontally. The vibration isolation and positioning system has to integrate with the other control systems in the accelerator. This paper shows the proposed integrated design using stiff piezoelectric actuators, to be robust against external forces, and a new sensor especially developed for vibration isolation in an accelerator environment. The control system for the vibration isolation is validated in models and in tests with a prototype of the smallest type 1 magnet. These tests include the proposed water cooling and the nominal magnetic field. Additionally a prototype is used to prove the nano-positioning capability of the system.

1 Introduction

To perform fundamental particle research, a linear lepton collider, called the Compact Linear Collider (CLIC), is currently under study by the international scientific community and at CERN [1]. To ensure the collision brightness or luminosity, CLIC requires a very small beam cross section of 1 nm x 40 nm at the interaction point (IP). These requirements are reached by using 4000 Main Beam Quadrupoles (MBQ) with a magnetic field gradient of 200 T/m, along the two main beam accelerator lines of 24 km. The effectiveness of these quadrupole magnets is reduced by any dynamic or static misalignment of the magnetic axis in relation to a nominal reference. The main contributions to the misalignment are ground vibrations and external forces coming from water cooling in the magnet, ventilation in the tunnel, etc [2]. A mechanical alignment system based on cam movers reduces the static misalignment. Two systems will work together to mitigate the effects of the dynamic misalignments. First, the position of the beam is measured by means of Beam Position Monitors (BPMs) and then corrected with a dipole magnet or by repositioning the quadrupole magnets to create a desired dipole field. This system mitigates vibrations below 1 Hz and at multiples of 50 Hz which is the repetition rate of the beam pulses. Secondly, the vibrations are reduced by an active stabilization system under each quadrupole which is the subject of this paper.

Early beam simulations showed that the vibrations of the magnetic axis of the quadrupoles should be limited to 1.5 nm integrated r.m.s. at 1 Hz absolute displacement in the vertical direction and 5 nm in the horizontal direction [3]. These requirements were met with a position feedback and feedforward control system using commercial seismometers and inclined piezo actuator legs on a first test set up with a 100 kg dummy mass. The vibrations were reduced from 2.5 nm integrated r.m.s. to 0.5 nm [4]. More importantly, the luminosity loss due to ground vibrations was reduced from 68% to 6%[5]. From new beam simulations it was found that the influence on the luminosity could be further reduced to 1% by using a simple inertial reference mass which allows for stabilization up to higher frequencies [6]. This upgrade will be the subject of this paper. First the challenges of using a sensor which includes the modes of the system in its bandwidth is explained. In the second part, a concept of a multi d.o.f. system is presented to limit the amount of actuators. Finally test results are given.

2 Mechanical configuration and its challenges

The commercial seismometer has an unwanted phase drop towards -π due to higher order filters that limit the bandwidth (see Figure 1). This renders the control loop only conditionally stable. By using a reference mass, which has a sensitivity from its resonance to infinity, this phase drop is avoided but the mechanical modes of the system are included in the bandwidth of the sensor potentially causing stability problems.

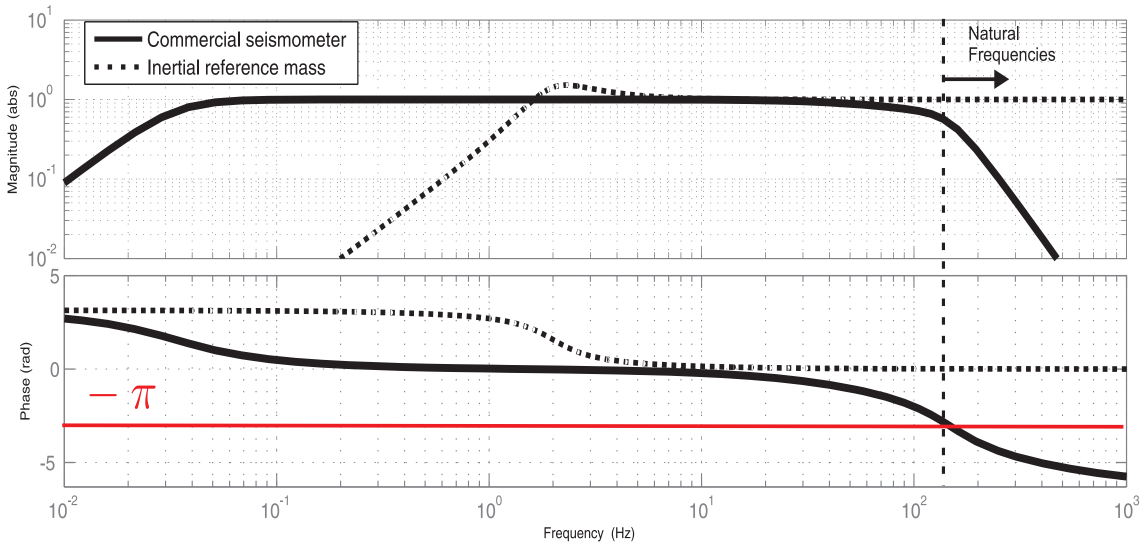

Figure 1: Sensitivity curve of the commercial seismometer and the foreseen inertial reference mass.

To investigate the effect of several expected modes on the system, an inertial reference mass was modeled on four different spring mass configurations. Figure 2 (a) shows the base configuration. In Figure 2 (b) a flexible appendage is added representing the flexibility of the longest Type 4 magnet. In configuration (c) a flexible support is included simulating the alignment system located under the stabilization system. Configuration (d) includes a flexible joint, to protect the actuator from moment and shear forces, between the actuator and the payload mass. The equations of motion of all these systems are given by

$$M\ddot{X} + C\dot{X} + KX = B\delta + Ew \tag{1}$$

with M, C and K the respective mass, damping and stiffness matrix of the system. Vector X includes the different position parameters. Matrix E is the input matrix for the ground motion w and B the input matrix for the actuator displacement which is given by $\delta = H(s)(x - x_g)$.

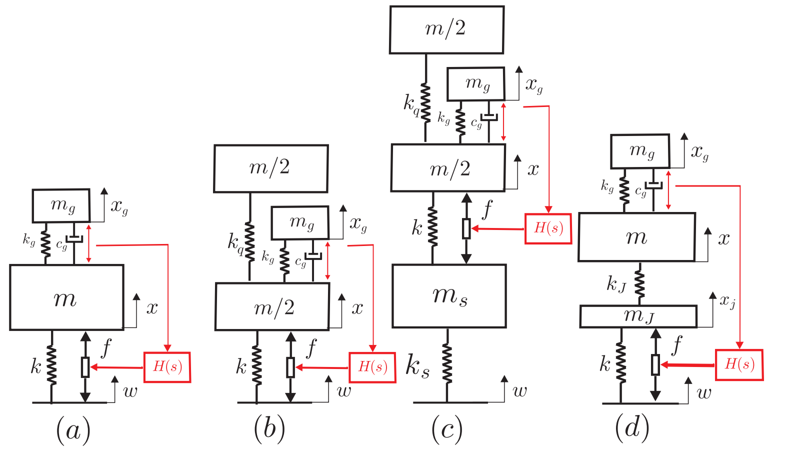

Figure 2: (a) Feedback with inertial reference mass m_g; (b) System with a flexible appendage of $m/2$; (c) System on a flexible support m_s and with a flexible appendage of mass $m/2$; (d) System with a flexible joint (m_j) between actuator and payload .

The controller $H(s)$ includes a lag and a lead to increase phase margins and damp the resonances. The resulting rootlocus plots for the different configurations are shown in Figure 3 with $k_q = 10k$ for configuration $2(b)$ and $k_q = 20k$ and $k_s = 10k$ for configuration $2(c)$.

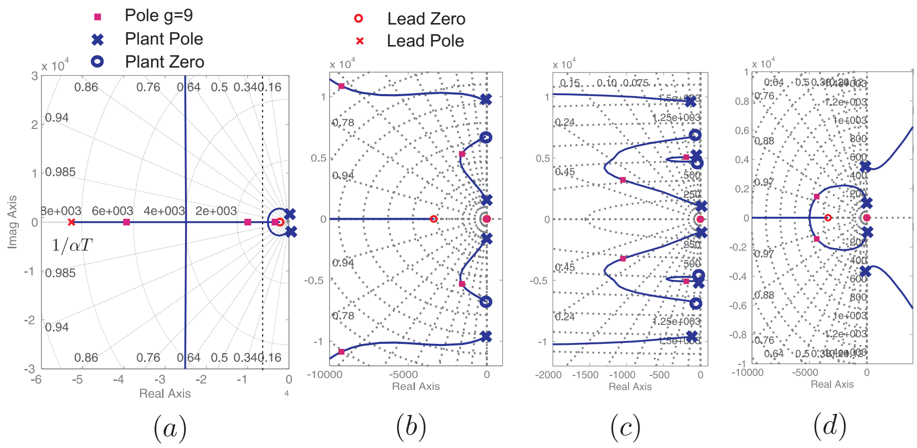

Figure 3: Rootlocus of (a) configuration 2 (a); (b) configuration 2 (b); (c) configuration 2 (c); (d) configuration 2 (d).

Configuration $2(a)$, $2(b)$ and $2(c)$ are always stable due to the alternating pole-zero pattern near the imaginary axis. This is caused by the collocation of the sensor-actuator pair. For collocation, the sensor and the actuator need to work on the same d.o.f. (x) and the sensor-actuator pair needs to be dual. They are dual when a translation sensor (displacement, velocity, acceleration) is associated with a force actuator or an angular sensor with a torque actuator [8]. The system with a flexible joint $2(d)$ is not collocated as the actuator works on x_j and the sensor measures x disrupting the alternating pole zero pattern. Mechanical poles that are not compensated by a zero will go to an asymptote. The angle of these asymptotes is given by the excess number of poles (n) in respect to zeros (m) by [7]

$$\phi_l = \frac{180° + 360°(l-1)}{n-m} \tag{2}$$

with $l = 1, 2, ..., n - m$. For configuration $2(a)$, $2(b)$ and $2(c)$ there are two poles more than zeros resulting in two asymptotes going to $\pm 90\,\mathrm{deg}$. The position of these asymptotes on the real axis is given by

$$a = \frac{\sum p_i - \sum z_i}{n - m} \tag{3}$$

Since the poles and zeros always appear in complex conjugate pairs, the imaginary parts add to zero. As the pole of the lead is located much further in the left half plane than the combination of the zeros, the two asymptotes will always be located in the left half plane. For the system with a joint there are eight poles and five zeros resulting in three asymptotes. One has an angle of $180\,\mathrm{deg}$ with a pole going to $-\infty$ and two with an angle of $\pm 60\,\mathrm{deg}$. The last two poles cause the system to be only conditionally stable as their poles will quickly wander into the right half plane.

The stability can be increased by adding damping and/or increasing the stiffness of the flexible joint, moving the pole further into the left half plane. Alternatively, the sensor and actuator can be made collocated again by relocating the reference mass on to m_j and measuring the relative displacement between x_j and the reference mass x_g removing the problem.

3 Multi d.o.f. configuration

The quadrupole magnets must be stabilized in the horizontal (x) and vertical (y) direction which also reduces the pitch and yaw motion if performed correctly with several pairs of in plane legs. There are however 3 d.o.f. in the plane perpendicular to the beam as the magnet can also roll with an angle θ. Normally the number of actuators is equal to the number of the degrees of freedom to control. Because 4000 magnets need to be stabilized the reduction of the number of actuators will result in a significant cost saving. To this end two actuators are positioned under an angle $\pm\beta$ with flexural hinges to protect the actuators. These hinges result in a very low lateral stiffness and a consequent low '4-bar'-mode. Using shear pins the '4-bar' mode can be increased to move it out of the sensitive stabilization bandwidth between 1 and 100 Hz and the longitudinal mode is also blocked. As there is no control over the roll, there will be a parasitic roll motion in the system. To quantify this drawback, an xy-guide prototype using a single actuator pair was built which is shown in Figure 4 (a). A schematic representation is given in Figure 4 (b).

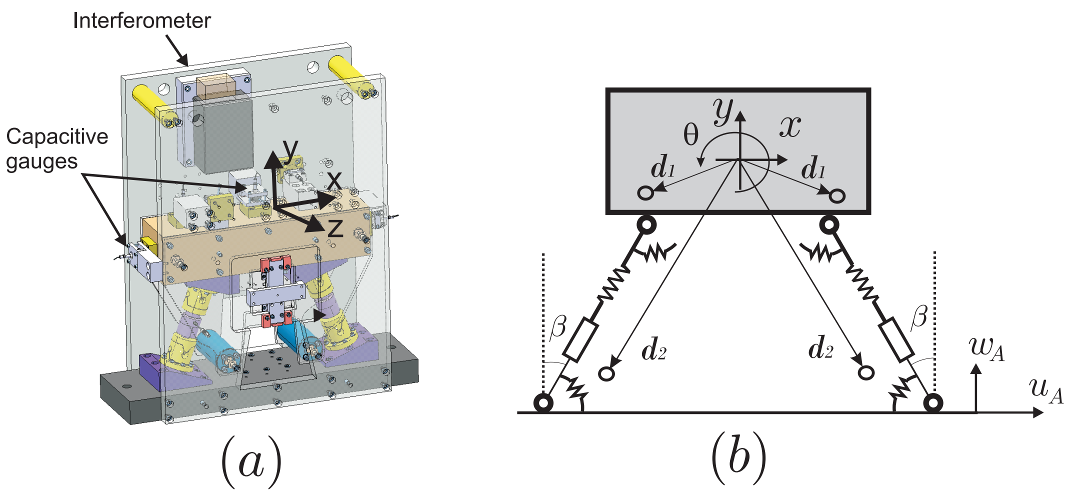

Figure 4: (a) CAD drawing of the xy-guide system including interferometer and capacitive gauge mounts; (b) Schematic representation of the xy-guide, the axial stiffness of the flexural hinges was omitted on the figure to keep it simple

The shear pins are positioned at a distance d_i from the rotation point adding a moment force to the system while limiting the contribution in the x and y-direction. The simplified kinematics of the system are given by

$$\delta_1 = \sin(\beta)x + \cos(\beta)y + (d_v \sin(\beta) - d_h \cos(\beta))\theta \tag{4}$$

$$\delta_2 = -\sin(\beta)x + \cos(\beta)y + (-d_v \sin(\beta) + d_h \cos(\beta))\theta \tag{5}$$

$$\alpha_1 = -\frac{\cos(\beta)}{r}x + \frac{\sin(\beta)}{r}y + (-d_v\frac{\cos(\beta)}{r} - d_h\frac{\sin(\beta)}{r})\theta \tag{6}$$

$$\alpha_2 = -\frac{\cos(\beta)}{r}x - \frac{\sin(\beta)}{r}y + (-d_v\frac{\cos(\beta)}{r} - d_h\frac{\sin(\beta)}{r})\theta \tag{7}$$

with δ_1, δ_2 the extension of the actuators and the angles α_1, α_2 are the changes in angle of the legs due to the movement of the magnet. The angle β is the angle of the leg from the vertical line and d_v and d_h are the vertical and horizontal distance from the rotation point to pin i ($d_i = \sqrt{dv_i^2 + dh_i^2}$). The potential and kinetic energy are given by

$$V = 1/2kq_1^2 + 1/2kq_2^2 + 1/2k_e\alpha_1^2 + 1/2k_e\alpha_2^2 + 1/2k_e(\alpha_1 - \theta)^2 + 1/2k_e(\alpha_2 - \theta)^2 \tag{8}$$

$$+ 2k_{pl}(x^2 + y^2) + (k_{pr1} + k_{pr2}^2)\theta^2 \tag{9}$$

$$T = 1/2M\dot{x}^2 + 1/2M\dot{y}^2 + 1/2I\dot{\theta}^2 \tag{10}$$

with k the longitudinal stiffness of the actuator and flexible joint, k_e the rotational stiffness of the joint and k_{pl} and k_{pr} the translation and rotational stiffness of the pins. By implementing the kinematic equations, the lagrangian

$$L = T - V \tag{11}$$

can be used to calculate

$$\frac{d}{dt}(\frac{\delta L}{\delta \dot{s}}) - \frac{\delta L}{\delta s} = 0 \tag{12}$$

for s equal to x, y and θ. This results in a set of equations of motion

$$M\ddot{X} + C\dot{X} + KX = 0 \tag{13}$$

with $X = [x, y, \theta]^T$, M the mass matrix, K the stiffness matrix. The damping matrix C is calculated by using modal damping

$$\phi^T C\phi = diag(2\xi_i\mu_i\omega_i) \tag{14}$$

with ϕ the mode shape matrix, ξ_i the damping ratio of the mode (1%), μ_i the modal mass and ω_i the angular frequency of the mode. These angular frequencies are found by solving

$$-\omega^2 M + K = 0 \tag{15}$$

for ω. The resulting frequencies of the first three modes for the conceptual designed xy-guide with a mass of 38 kg are shown in Table 1

Due to the limited stiffness in the x-direction, caused by the flexible joints, the '4-bar'-mode is 11 Hz. The added stiffness from the pins increases this mode to 156 Hz. The increase for the vertical mode is only limited as the stiffness of the piezo actuator is much higher than the stiffness of the pins. The roll mode has been increased from 128 Hz to 551 Hz.

	'4-bar' mode	Vertical mode	roll mode
without pins	11 Hz	366 Hz	423 Hz
with pins	156 Hz	389 Hz	551 Hz

Table 1: The modes of the xy-guide with and without pins

4 Test set-up and results

4.1 xy-guide test results

4.1.1 Stabilization

In reality the xy-guide, shown in Figure 4 (a), will have a number of extra modes. These are caused by the attachment plates for the pins which are not perfectly rigid, the longitudinal modes which were not in the model, etc. These extra modes are determined by performing an openloop measurement. A sweep sine was injected in the piezo actuators and the output was measured with the seismometer and with the inertial reference mass. The openloop transfer functions are shown in Figure 5.

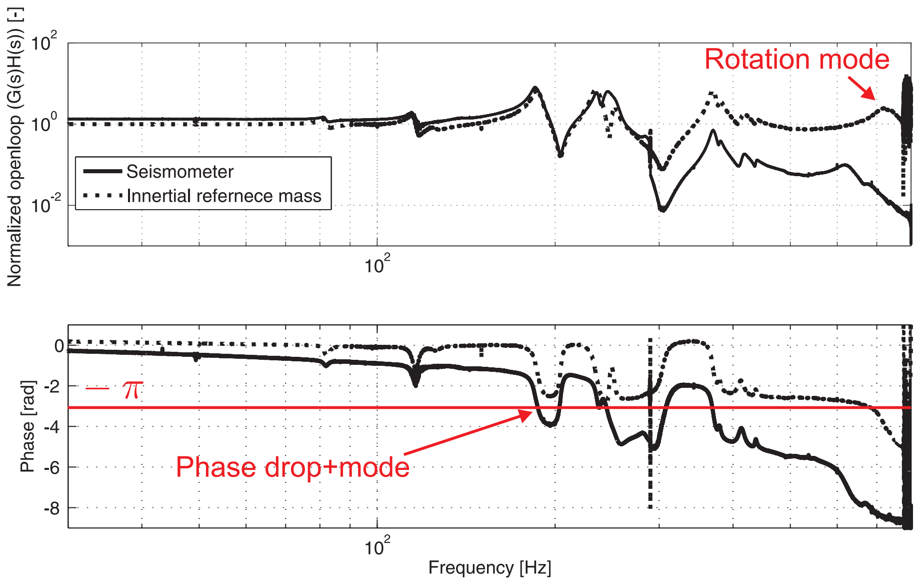

Figure 5: The normalized openloop transfer function for the xy-guide in y-direction using the commercial seismometer and the inertial reference mass.

By definition, a feedback system becomes unstable when the phase of the openloop drops below -π rad for a magnitude greater than one [7]. As was shown in Figure 1, the high order filters of the seismometer cause a phase drop resulting in an instability at the first mode. This is counteracted by double lag pushing the modes out of the sensor bandwidth [5]. With the double lag in the feedback configuration and adding a sensor to perform feedforward, the xy-guide was stabilized to 0.45 nm integrated r.m.s. at 1 Hz coming from 4.3 nm integrated r.m.s as is shown in Figure 8.

The stability limit for the inertial reference is caused by a mode at 600 Hz. This is the rotation mode that is picked up by the sensor as it could not be positioned exactly in the middle. The mode causes a 180 degree phase drop while there is no compensation zero before it rendering the control system unstable for small gains. Usually this is compensated by using a notch which adds a zero near the poles of the resonance and two compensation poles far away not to affect the rest of the system. This will however not be convenient for 4000 quadrupoles with different weights as the notch needs to be tuned precisely to render the feedback control system stable. Therefore it is suggested to go from one sensor on top of the magnet to a sensor for each leg directly in series with the actuator making it a Single Input Single Output (SISO) system and collocated. This will solve the destabilizing effect of the roll mode and eliminate the destabilizing effect of the hinge if the sensor measures the output of the actuator directly, making it collocated.

4.1.2 Positioning

In order to define the exact motion of the dummy mass for the positioning and determine the parasitic roll, the xy-guide includes mounting positions for multiple sensors to measure the motion of the dummy mass representing the magnet. Two capacitive gauges measure the movement in x and y-direction. A three beam interferometer is used to complement the measurement in the y-direction and measure the roll around the x and z-axis by calculating the difference between the three beams. Figure 6 (a) shows the lateral and vertical movement measured by the capacitive gauges during a test with the prototype xy-guide. At the same time the resulting roll was measured around the z-axis by the interferometer (see Figure 6 (b)).

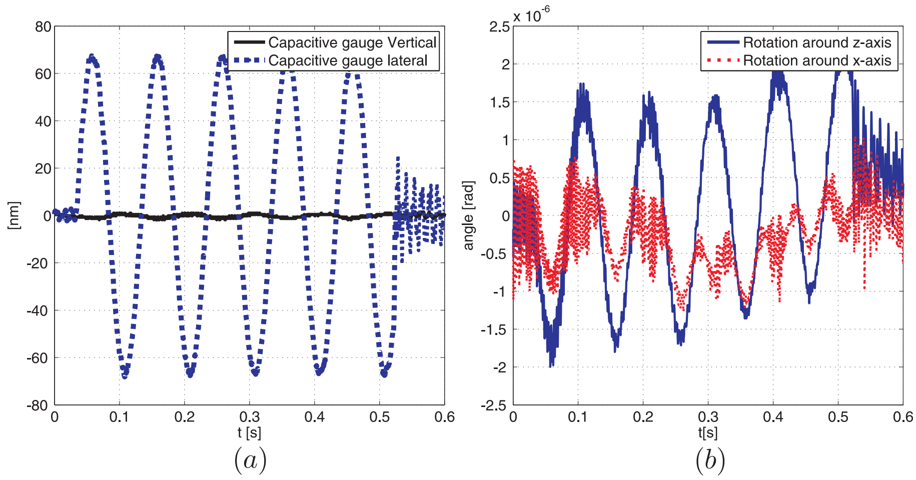

Figure 6: (a) Vertical and horizontal motion of the block measured with capacitive gauges in the vertical and lateral direction; (b) The angle around the x-axis and z-axis measured at the same time with a three beam interferometer.

The measured roll is 1.5 μrad for a lateral motion of 66 nm movement in the x-direction while the model presented in section 3 predicts a roll of 2 μrad for the same horizontal motion. Beam simulations will be performed to define the effect of this roll motion on the luminosity. For now, it is well in the pre-alignment requirement of 100 μrad.

To achieve precise positioning, straingauges in the legs measure the exact extension of the piezo actuators. This measurement is used in a Proportional Integral (PI) control loop in order to eliminate any position offset due to the reaction forces of the pins and other hardware effects. The resulting performance with the PI-controller on and off for a square wave of 30 nm is shown in Figure 7.

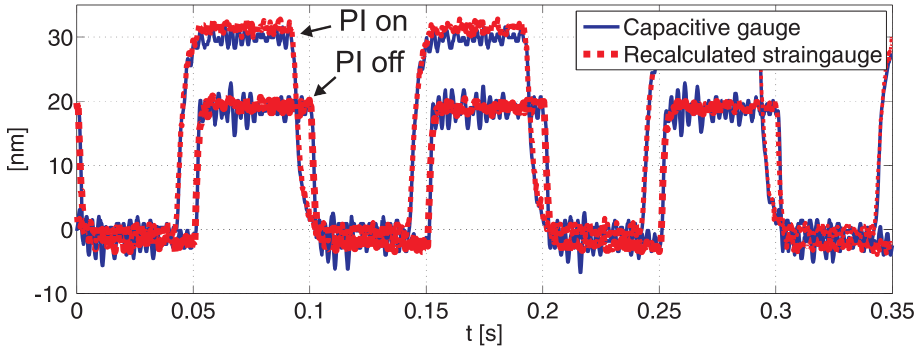

Figure 7: The position in the y-direction caused by a square wave input of 30 nm, measured by the capacitive gauge in y-direction and by recalculating the measurement of the straingauges in the legs for the y-direction with the PI-controller on and off.

4.2 Type 1 magnet stabilization under nominal conditions

To probe the performance of the stabilization system in an accelerator environment, a set up was created with a type 1 quadrupole magnet supported by two inclined active legs and two passive supports in the back allowing movement in 2 d.o.f. for small displacements. This test bench is shown in Figure 8 (a). During the stabilization test the quadrupole magnet was powered to the nominal magnetic field with the nominal water cooling flow of 4 l/minute going through the coil exerting direct forces. Figure 8(b) shows the measured integrated r.m.s. displacement for the feedforward-feedback configuration with commercial seismometers.

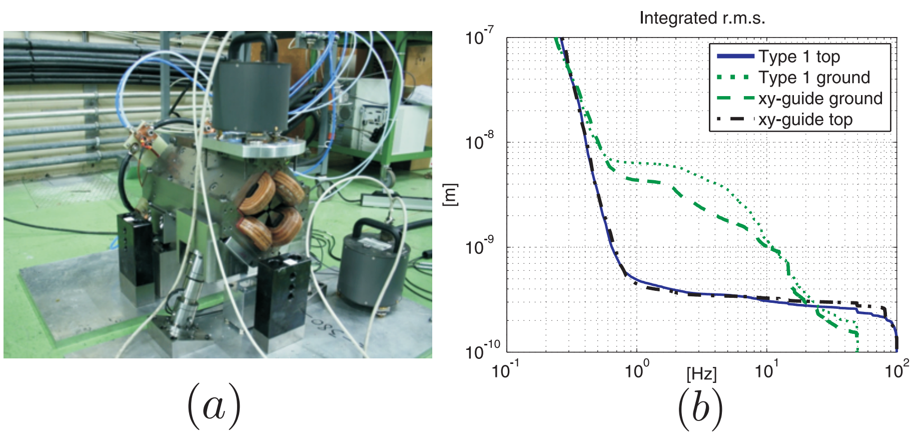

Figure 8: (a) Type 1 magnet with two inclined legs allowing for tests with magnetic field and water cooling; (b) The resulting integrated r.m.s. graph for the type 1 magnet with nominal water cooling and magnetic field and for the xy-guide prototype.

The stabilization system decreased the integrated r.m.s. vibrations from 6.3 nm to 0.5 nm or a reduction factor of 13, this with a nominal magnetic field and water cooling.

5 Conclusion

This paper has shown that by changing from a seismometer to an inertial reference mass, the destabilizing modes of the system are included in the sensor bandwidth. Adding a flexible appendage or support does not destabilize the system but adding a flexible joint makes the system non-collocated causing stability problems. This problem can be reduced by increasing the stiffness of the joint or solved by making the system collocated by putting the sensor directly on the actuator. Stabilization was performed with a commercial seismometer and an xy-guide prototype down to 0.45 nm integrated r.m.s. A model of the guide showed the increase in the natural frequencies of the system by the implementation of shear pins and the parasitic roll was confirmed in both the model and measurements to be a few μradians. The rotational mode can cause a problem when the inertial reference mass is not centered. By making the actuator and sensor collocated this can be avoided. The stabilization requirements were also reached for a type 1 magnet with nominal magnetic field and water cooling.

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References

[1] W.P. Delahaye, *Towards CLIC Feasibility*, IPAC10, Kyoto, Japan (2010).

[2] C. Collette et al., *Seismic response of linear accelerators*, Physical Review Special Topics Accelerators and Beams 13 (2010), 072801

[3] C. Collette et al., *Nano-motion control of heavy quadrupoles for future particle colliders: an experimental validation*, Nuclear Instruments and Methods in Physics Research Section A 643 (2010), pp. 95-101.

[4] K. Artoos et al., *Status of a study of stabilization and fine positioning of quadrupoles to the nanometer level*, IPAC'11, San Sebastian, Spain, (2011).

[5] S. Janssens et al., *System control for the CLIC Main Beam Quadrupole stabilization and nano-positioning*, IPAC'11, San Sebastian, Spain, (2011).

[6] J. Pfingstner et al., *Recent improvements and robustness studies of the integrated simulations of the orbit controller for CLIC*, IPAC'12, New Orleans, United States of America, (2012).

[7] G.F. Franklin, et al.,*Feedback Control of Dynamic systems (5th edition)*,Pearson Prentice Hall, Upper Saddle River, New Jersey, 2006.

[8] A. Preumont, K. Seto, *Active Control of Structures*, Jhon Wiley & Sons Ltd., West Sussex, UK, 2008, ISBN: 978-0-470-03393-7