# DEVELOPMENT OF ADVANCED MECHANICAL SYSTEMS FOR STABILIZATION AND NANO-POSITIONING OF CLIC MAIN BEAM QUADRUPOLES

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*Abstract*

CLIC is a multi-TeV electron-positron linear collider currently under study. To reach the desired high luminosity, stringent alignment requirements should be satisfied, particularly for the Main Beam Quadrupoles (MBQ). An alignment stage will align the MBQ with micrometre resolution. Displacements due to ground motion and technical vibrations in the 0.1-100 Hz frequency range can however not be corrected with the alignment stage. An active vibration isolation system, based on piezoelectric actuators and inertial reference masses, will therefore be installed between the mechanical alignment stage and the magnet. This system can also be used for relative repositioning in between beam pulses with nanometre resolution and with a range of 10 micrometre. Compatibility between the actuating support, the alignment, fiducialisation and beam position measurements (BPM) should however be guaranteed. The actuating support should, in the same way as the alignment system, be robust against forces acting on the quadrupole and displacements created by the active support should not upset the initial alignment. Stiff piezo actuators with a fast response are therefore combined with flexural mechanisms and joints to create a guide for very precise displacements, to eliminate backlash and friction and to increase the frequency of internal modes. Precise measurement of the relative displacements and a good analysis of the kinematics are essential to ensure that the alignment of the MBQ is always well known. The performance and precision reached with a prototype of the actuating supports and a comparison between different solutions will be presented in this paper.

## INTRODUCTION

The Compact Linear Collider (CLIC), which is currently under study, is an $e^-e^+$ collider with a center-of-mass energy range from 0.5 to 3 TeV [1]. It will use accelerating structures with a very high gradient (100 MV/m). Even with this gradient the overall CLIC length for 3 TeV will be 48 km. One of the challenges is the emittance conservation and this imposes very tight alignment tolerances. Particularly critical from this point of view are the Main Beam Quadrupoles (MBQ). There are about 4000 of such magnets distributed along the two main beam lines and they will be mechanically aligned with micrometre precision by an active device based on supports with motorized eccentric cams [2]. The MBQ should be aligned with respect to an ideal straight line to within ±10 µm over a sliding window of 200 m [3]. The alignment stage will ensure a sufficiently accurate static positioning but it will not be able to correct the dynamic effects of ground motion or vibrations from the technical systems [4]. A piezo actuating support will therefore be placed in series above the alignment cams to mechanically stabilize the MBQ between 0.1 and 100 Hz.

External static forces act on the MBQ through the magnet interconnects, water cooled power or other cables. Both the alignment and stabilization should therefore have a sufficiently high stiffness. Passive vibration stabilization with a soft support is less adapted and a stiff actuator support with active position feedback using inertial reference masses was developed. The stiffness of the support makes it also robust against dynamic forces acting directly on the magnet and increases the frequency of low frequency resonances that would disturb the controller. The stabilization system reduced the integrated root mean square value of vibrations above 1 Hz of a water cooled magnet to values well below 1.5 nm [5] in the vertical direction and smaller than 5 nm in the horizontal direction. In addition the transfer function of the system can be optimized in function of luminosity [6].

Moreover, this strategy gives the possibility to perform nanometric relative positioning of the MBQ in between beam pulses in order to use the MBQ as a corrector dipole by offsetting it from the beam axis or to make fast small adjustments of the alignment in between beam pulses. Mechanical requirements for the active stabilization and nano-positioning supports can be summarized in the following points:

- Very high stiffness;
- Transportability;
- 4 degrees of freedom (DOF) of the magnet (vertical, lateral, yaw and pitch);
- Compatibility with alignment, fiducialisation and BPM measurements;
- Optimized cost;
- Compatibility with accelerator environment;

The design of the actuator support and geometry based on the above requirements are described in this paper. The stiffness, modal behavior and kinematics of the magnet on the actuating support are given. The compatibility with alignment, fiducialisation and BPM measurements will depend on the precise knowledge of the kinematics. The motion should be precise (repetitive) and accurate. The position of the magnet in the actuating support should be measured with high resolution and the relation between this position and the position of fiducials and BPM should be unique within their resolution. This is discussed and verified with measurements in a prototype. Four different sensors will be compared. Finally, future work will be briefly indicated.

## ACTUATING SUPPORT

### *Actuator geometry*

The studied solution for the MBQ active support consists of pairs of actuator legs lying in a vertical plane and inclined by 20° with respect to the vertical direction (Fig.1). The number of pairs depends on the length and mass of the magnet. Each leg is composed of a very stiff piezoelectric actuator and 2 flexural hinges. The piezoactuator is a commercial component [7] with a very high axial stiffness ($k_{ac}$=480 N/µm), a range of 15 µm and a resolution of 0.15 nm.

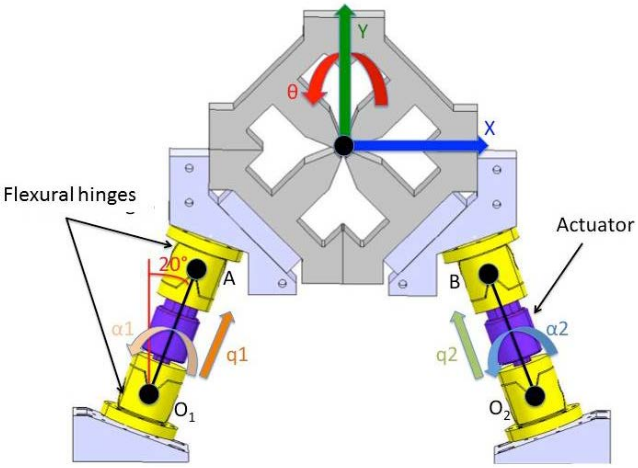


Fig.1: 4-bar system model & coordinate systems

The flexural hinges have been purposely designed to have a high rotational stiffness of $k_e$=220 Nm/rad and a high axial stiffness ($k_{al}$=300 N/µm).

The mechanism geometry with four links (joints) and four sides is a four linked bar geometry. Such a geometry is not fully constrained and can still move even without change in length of the legs (Kutzbach criterion) [8]. To reduce the number of degrees of freedom (d.o.f.), the four linked bar geometry will be constrained to a vertical plane by an x-y guide based on flexural guides, described later.

Restrained to a plane, the kinematics of the 4-bar system can be fully described by a set of Cartesian coordinates (x,y,θ) with origin in the center of the magnet. However a set of local "leg" coordinates ($q_1$,$q_2$, $\alpha_1$, $\alpha_2$) is important to have direct information on the position of the legs (Fig. 1). The relationship between the leg coordinates and the magnet coordinates can be written as follows:

$$q_1 = sin\beta\ x + cos\beta\ y + (D_v\ sin\beta - D_h\ cos\beta)\theta$$

$$q_2 = -sin\beta\ x + cos\beta\ y - (D_v\ sin\beta - D_h\ cos\beta)\theta$$

$$\alpha_1 = -\frac{cos\beta}{R}\ x + \frac{sin\beta}{R}\ y - \left(\frac{D_v cos\beta}{R} + \frac{D_h sin\beta}{R}\right)\theta \qquad (1)$$

$$\alpha_2 = -\frac{cos\beta}{R}\ x - \frac{sin\beta}{R}\ y - \left(\frac{D_v cos\beta}{R} + \frac{D_h sin\beta}{R}\right)\theta$$

where R is the initial length of the leg (distance between centers of 2 joints), β the initial actuator angle 20 degrees, $D_v$ and $D_h$ are the vertical and horizontal components of the distance between points A and B (Fig.1) and the center of the magnet.

The leg coordinates are not independent. An additional constraint is introduced as a constant distance between point A and B and three equations with 3 variables describe then the system. This means that the four linked bar geometry, constrained to a plane, and with the actuators fixed, still has one d.o.f. This d.o.f. is a rotation around the imaginary intersection point of the actuator leg axes. In such a movement the axial stiffness of the legs ($k_a$=115 N/µm) does not intervene and the only counteracting forces come from the rotational stiffness of the hinges (giving one reason for their designed high value). The lateral stiffness of the system can be calculated by using the principal of virtual work (Fig.2) with M the moment required to turn a hinge by angle α.

$$F*dx = M_{O1}*\alpha_1 + M_{O2}*\alpha_2 + M_A(\alpha_1 - \theta) + M_B(\alpha_2 - \theta) \qquad (2)$$

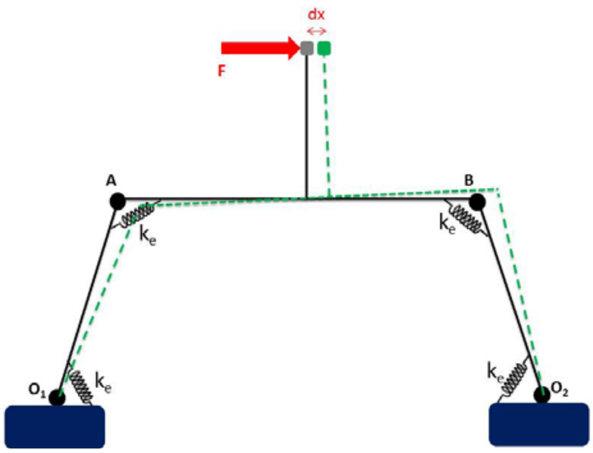


Fig.2: 4-bar system stiffness calculation

Despite the high rotational stiffness of the hinges, the lateral stiffness would still be too low (0.2 N/µm) and shall therefore be increased by the x-y guide.

In the same way, the vertical stiffness (~230 N/µm) is mainly determined by the axial stiffness of the two hinges and actuator in series and is rather low compared to the single actuator stiffness (480 N/µm). An increase of vertical stiffness, if needed, would require an increase of the axial hinge stiffness.

The x-y guide restrains each actuator pair with flexural guides (pins) between two fixed vertical support plates. This blocks the longitudinal d.o.f. of the magnet and the flexural stiffness of the pins will increase the vertical and lateral stiffness of the actuating support.

A symmetrical distribution of the pins around the magnet is needed for the stiffness of the longitudinal blocking of the magnet. This also allows introducing locking for transport in longitudinal direction, hence blocking the magnet mechanically without introducing tensile stress on the piezo actuators. By placing the pins far from the virtual intersection point of the actuator legs the stiffness for the rotation around this point will increase. Doing so, also the lateral stiffness will increase more than the sum of the lateral stiffnesses of the flexural pins. The initial choice of the stiffness of the pins (i.e. material and dimensions) comes from a compromise between the increase of stiffness and the decrease of the actuator range due to this. Eight pins (four on the front side and four on the back) are implemented for each support. The pins have a 5 mm diameter, 20 mm length and are made of aluminum. Their flexural stiffness is 3.2 N/µm.

### *Stiffness and modal behavior*

Finite element (FE) models were built to complete the stiffness calculations and to study the modal behavior, kinematics and dynamics of the magnet on the actuator support. An analytical modal analysis of the 4-bar system has been performed using the Lagrangian method [9], in which kinetic and potential energy can be expressed as follows:

$$T=\frac{1}{2}(m\dot{x}^2+m\dot{y}^2+I\dot{\theta}^2) \qquad (4)$$

$$V=\frac{1}{2}\{k_a(q_1{}^2+q_2{}^2)+k_e[\alpha_1{}^2+\alpha_2{}^2+(\theta\text{-}\alpha_1)^2+(\theta\text{-}\alpha_2)^2]\} \qquad (5)$$

Analytical and numerical calculations are in general in good agreement.

The x-y guide with flexural pins increases the lateral stiffness by more than a factor 200 while practically not changing the vertical stiffness and raises the first resonant frequency for a type 1 magnet (0.5 m long, 85 kg) from less than 10 Hz to more than 100 Hz.

The first longitudinal mode increases from 3.4 Hz without pins to 65 Hz with 8 pins connected to thick steel plates. That frequency can however be further increased by decreasing the mass of the plates and by introducing e.g. aluminium strut structures. For infinitely rigid vertical fixed plates the longitudinal first mode is at 280 Hz.

Figure 3 shows the frequency and shape of the three principal resonance modes (vertical fixed plate transparent for visibility).

**lateral mode**
f≈9 Hz (without pins);
f≈140 Hz (with pins);

**θ mode**
f≈250 Hz (without pins);
f≈300 Hz (with pins);

**Vertical mode**
f≈315 Hz (without pins);
f≈335 Hz (with pins);

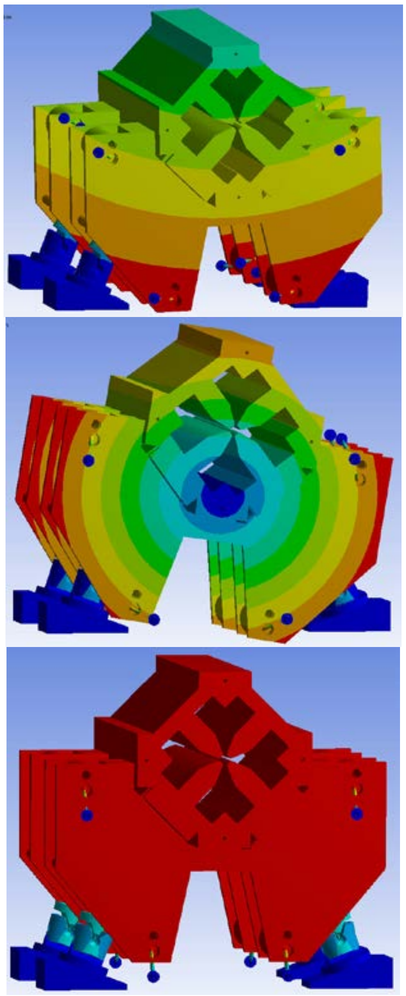

Fig.3: Natural modes of the MBQ on the 4-bar support

### *Simulated kinematics*

The FE models were also used to simulate the motion of the magnet with the piezo actuators. The Young modulus of the 3D actuators was set to have the same stiffness as the real piezo. The thermal expansion coefficient was set such that a free piezo (without flexural guides) expands by 1 µm for 1 degree temperature change. Piezo actuation was simulated by changing the temperature of each piezo. For the stabilization and nano-positioning, the actuating support must produce horizontal and vertical displacements of the magnet. Roll (θ) is undesirable and cannot be controlled by the actuators. The FE models showed however a parasitic roll when making a lateral motion. The same roll is also calculated by minimizing the potential energy in (5) with q1 and q2 equal but with opposite sign (for a lateral displacement). The roll is created by internal forces due to the rotational stiffness of the joints. For a vertical motion the hinge deformations are symmetric and no parasitic roll occurs. The calculated size of the parasitic roll during lateral motion is about 5 µrad/µm, i.e. maximum 25 µrad change from the aligned position for a 5 µm lateral displacement. Preliminary beam physics simulations with quadrupole and BPM roll (attached to magnet) indicated that roll should remain smaller than ±100 µrad [10].

A set of parameters was calculated to describe the quality of the motion (Table 1). The horizontal and vertical movements for a given actuator extension x/q and y/q express the coordinate relationship of (1) taking into account the influence of the stiffness of actuators and flexural parts. The coupling between horizontal and vertical motion x/y should of course be close to zero for simplicity of the control. The parasitic roll θ/y was described above. The consistency between analytical and numerical results is good.

Table 1: Actuator guide parameters

| | | x movement | | y movement | |
|---|---|---|---|---|---|
| 8 PINS | Analytical | x/q | 1.3 | x/y | 0 |
| 8 PINS | Analytical | y/x | 0 | y/q | 1.06 |
| 8 PINS | Analytical | θ/x [µrad/µm] | 5.15 | θ/y [µrad/µm] | 0 |
| 8 PINS | FE | x/q | 1.24 | x/y | 0 |
| 8 PINS | FE | y/x | 0.05 | y/q | 1.03 |
| 8 PINS | FE | θ/x [µrad/µm] | 5.25 | θ/y [µrad/µm] | 0 |

The possibility to apply pitch and yaw to the magnet was also verified with the FE models (figure 4).

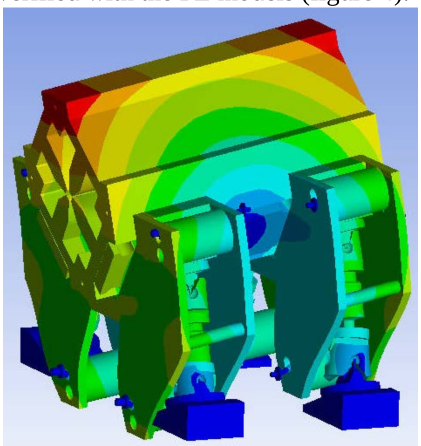

Fig. 4: FE analysis of pitch of type 1

As the flexural hinges are universal joints (two bending axes) pitch and yaw are not a problem for the actuator legs. It is however more complicated for the shape that the flexural pins must take. The simulations indicated that both pitch and yaw are possible, but the stiffness for yaw should be reduced. For the moment, not enough data is available from beam physics on how to calculate such corrections and the range that would be needed.

## NANO POSITIONING SENSORS

The condition for using a stabilization-nanopositioning system in CLIC is the compatibility of the resolution, precision and accuracy (semantic differences important in this case) [8] between the piezo actuating system, the alignment system, the magnet fiducialisation and the BPM measurements. The piezo actuating system can be an addition to the pre-alignment system by increasing the resolution for relative adjustments of the magnet position from the sub micrometre level of the alignment cam movers to the sub nanometre level of the piezo actuators. The 10 µm range of the nano positioning structure could allow for magnet position adjustments beyond the alignment resolution. This would increase availability of beam time because adjustments with the cam movers are too slow to be made in between beam pulses (20 ms). Since the BPM and the fiducials are fixed to the quadrupoles, concerns arise when moving them with the piezo actuating support. The exact motion of the magnet and the attached fiducials and BPM should be well known. For this purpose, the x-y actuating support should have a precision, i.e. repeatability of positioning at least one order better than the resolution of fiducials and BPM. Furthermore, when displacements are made within resolution of BPM or alignment sensors, the accuracy of the motion measurement should be as good, i.e. the measured displacement should correspond. Lever arms between magnet, fiducial and BPM supports should be kept to a minimum. Finally, the range of the displacement of the BPM should not alter its sensitivity or linearity.

Below 1 Hz, the ground motion can be in the sub micrometre and even micrometre range. This is measured by the inertial reference masses and the stabilization system will create the same order of displacements and in some cases slightly amplified. The same concerns as mentioned above for nano-positioning, will in this case apply.

Displacement sensors will therefore be incorporated in the magnet piezo support to measure the vertical and lateral displacements of the magnet, relative to the interface between alignment and the stabilization system.

To precisely measure the kinematics of the four linked bar geometry combined with an x-y guide based on flexural pins, an x-y guide prototype was built (Fig. 5) with a 52 kg block moved by one inclined piezo actuator pair.

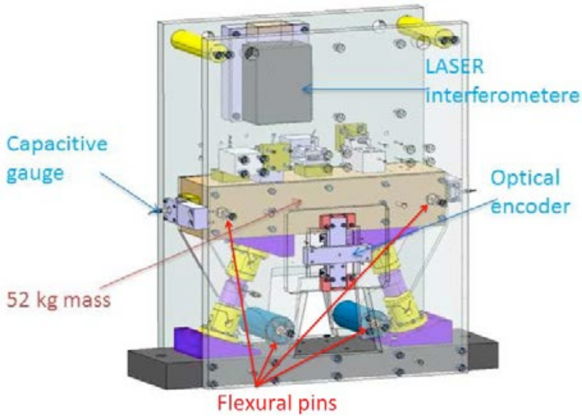


Fig.5: x-y prototype with sensors

In this x-y guide prototype, four different measurement systems were implemented to study the kinematics of the actuating support and to compare the different sensors.

First, the piezo actuators [7] have strain gauges to measure the extension of the actuators. Capacitive gauges [11] were installed to measure lateral and vertical displacements of the block with respect to the fixed support plate. A triple beam LASER interferometer [12] measured the vertical displacements, rolls and pitch. Finally, an optical grating encoder interferometer [13] measured in the vertical direction. Table 2 shows the different resolutions of the used sensors, as indicated in the catalogues.

Table 2: Nano positioning sensors resolution

| Sensor | Resolution |
|---|---|
| Strain gauge | 0.3 nm |
| Capacitve gauge | 0.1 nm |
| Interferometer | 10 pm |
| Optical encoder | 1 nm |

Different factors will affect the resolution, precision and accuracy of the sensors after installation in the actuating support. Such factors can be summarized as mounting tolerances of the sensors, quality and alignment of the actuating support, coupling between orthogonal motion, temperature stability, airflow and finally vibrations in the support. Of course, cabling and acquisition are important but not the subject of this paper. The actuator strain gauges have a smaller resolution and do not measure directly the motion of the magnet and are used in our prototype to measure the actuator input.

The capacitive gauges were until now the most spread sensors for actuation at the nanometre level and below. For this type of resolution the range and the gap between the electrode and target are however reduced to a few microns and the sensor alignment tolerance (<1 mrad) is difficult to obtain in larger actuating systems subjected to external forces. A misalignment will result in an important change of gain (> factor 10) and the capacitive gauges therefore need recalibration after installation. They are also not optimal for displacement perpendicular to the measurement direction due to irregularities on the target surface and small misalignment in the actuating support.

The triple beam LASER has an unmatched resolution but suffers from low frequency drift due to temperature changes and air flow, is too expensive for large scale use and has issues with alignment and orthogonal coupling. This device is however excellent as a calibration reference and to analyze the kinematics of the actuating support.

Finally, the optical encoder interferometer is easier to implement with achievable mounting tolerances (gap 0.1 mm, alignment 3.5 mrad) but requires nevertheless some care. Orthogonal coupling of only about 1% was measured in the x-y prototype. The device is sensitive to air flow but less than the triple beam interferometer and can be protected against this.

The optical encoder has the best advantages with respect to the other sensors. To illustrate the performance, vertical and lateral steps of 6 nm were made in twenty milliseconds with the x-y prototype and each position was kept for the same duration. The measurements of the steps in the vertical direction are shown in figure 6. From this test and several others it is clear that the optical ruler has the best resolution in this set-up compared to the other sensors. The measurements also show that the repeatability or precision of the x-y prototype is better than 1 nm. The repetitivity of the optical ruler is also slightly better than the actuator gauges and the capacitive gauges.

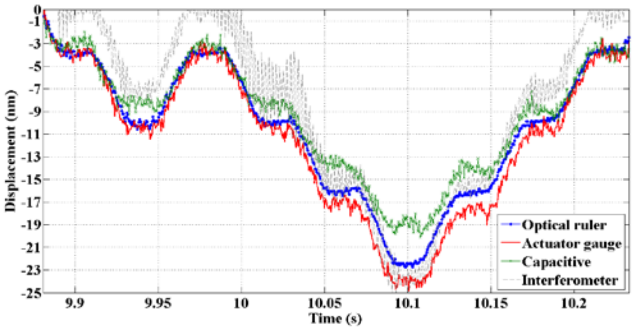


Fig.6: Vertical steps of 6 nm and 20 ms measured with four different sensors

The precision and accuracy of the three-beam interferometer is compromised by drifts due to air-flow. The gain of the capacitive gauge was corrected in this figure as mentioned earlier and hence only the accuracy of the actuator leg and the optical ruler can be really considered. By calibrating the MBQ with actuating support, an accuracy per measurement point better than 10 nm is achievable.

A main drawback for the long-term accuracy are however possible drifts of references: voltages for capacitive and actuator gauges and wavelength for the interferometer (periodic recalibration). The reference used for the optical rulers are edges etched on a glass ruler with small thermal expansion coefficient. However, if the counting of the edges is interrupted during displacements, e.g. during a power cut, the absolute position is also lost. This can be solved by using absolute optical encoders that also measure the absolute position etched on the ruler (hence no problem for reference drift). The accuracy of the etching is in the micrometer range. Again this can be improved by calibration. Such absolute sensors will be soon tested in the x-y prototype.

Temperature drift will influence of course the position and position measurement of the magnet and this is difficult to avoid. The best way to avoid such problems is a temperature stable environment.

Finally, mechanical vibrations and shocks during the measurements alter the effective resolution. Having a very stiff support improves the situation significantly and the sensor with the smallest influence of gap distance or orthogonal coupling (optical ruler) will have less problems. It is also clear from the tests that the main source of vibration excitation decreasing resolution is the input signal to the piezos. It should be avoided to have frequency components in the applied voltages that are resonant frequencies of the structure, square steps are not suitable. For figure 5, a low pass filter was applied to the input. Techniques for shaping the input from a limited number of frequencies are also possible.

### *Measured kinematics*

The kinematics and modal behaviour of the actuating system was verified with the x-y prototype and the instrumentation and a good agreement was found with the models on most parameters. The parasitic roll during lateral motion was measured with the 3 beam laser interferometer (Fig. 6). For a 1500 nm amplitude lateral motion, a roll of 3.3 µrad (amplitude) was measured or 2.2 µrad/µm for lateral motion. This is smaller than the calculated values and indicates that the limit specified by the beam dynamics can be met.

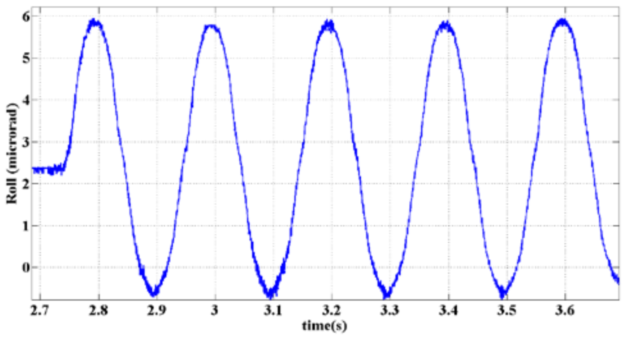


Fig. 6: Parasitic roll for a sinusoidal lateral motion of 1.5 µm

## FUTURE STUDIES & DEVELOPMENTS

Alternative solutions for an active support for the MBQs are currently under study and development. The main goals pursued are the improvement of robustness against external forces, higher precision response to actuator inputs and easier precise installation and mounting of the actuator support.

A promising solution is the construction of a support in a single monolithic design, decreasing the number of assembly tolerances.

An objective is to combine a quadrupole magnet with fiducials and a BPM in a stabilization support placed on an alignment system with alignment sensors and to do magnetic field and BPM measurements in order to determine the combined achievable precision and accuracy for implementation in CLIC.

## CONCLUSIONS

A stiff actuating support is under development for the CLIC main beam quad vibration stabilization. The latter can be used also for nano-positioning of the magnet in between beam pulses. The active support needs to be very robust against external forces and its mechanical behavior must be predictable and measurable with very high accuracy and precision to be compatible with the micrometric resolution alignment stage, fiducialisation and BPM measurements. This paper shows that the mechanics has been studied in detail using analytical and numerical procedures and that tests performed on a first actuating support prototype indicate that the obtained resolution, precision and accuracy can be sufficiently good for compatibility. Both the actuator support and some of the sensors can reach sub-nanometre resolution. More elaborate prototypes with improvements and combining alignment system, fiducials, magnetic field and BPM measurements will be developed and built to further confirm this.

## ACKNOWLEDGEMENTS


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