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LAPP-TECH-2008-05
September 2008

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Presented by **B. Bolzon**
at NANOBEAM'08, Advanced Beam Dynamics Workshop,
Novosibirsk (Russia), 25-30 May 2008

CENTRE NATIONAL DE LA RECHERCHE SCIENTIFIQUE
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STUDY OF VIBRATIONS AND STABILIZATION AT THE SUB-NANOMETRE SCALE FOR CLIC FINAL DOUBLETS*

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Abstract

CLIC is one of the current projects of high energy linear colliders. Vertical beam sizes of 1nm at the collision point and fast ground motion of a few nanometres impose an active stabilization of the final doublets at a tenth of a nanometre above 5Hz.

The majority of our work concerned vibrations and active stabilization studies of cantilever and slim beams in order to be representative of the CLIC final doublets, which is one of the originality of the work. For that, we determined appropriate sensors and actuators, developed an active control algorithm, modelled the phenomenon, and finally constructed a prototype.

In a first part, measured performances of different types of vibration sensors associated with an appropriate instrumentation showed that accurate measurements of ground motion are possible from 0.1Hz up to 2000Hz even if vibrations are low. Some non magnetic electrochemical sensors compatible with the specifications of CLIC can be incorporated in the active stabilization at a tenth of a nanometre.

In a second part, a study of the impact of ground motion and of acoustic noise on beam vibrations showed that an active stabilization is necessary at least up to 1000Hz.

In a last part, results on the active stabilization of our prototype at its two first resonances are shown down to amplitudes of a tenth of a nanometre above 5Hz by using in parallel a commercial system performing passive and active isolation from ground motion.

INTRODUCTION

The luminosity of CLIC collider is planned to be of the order of $10^{35}cm^{-2}s^{-1}$, which imposes a vertical beam size of 1nm. In order to maximise the luminosity at the interaction point, the relative motion between the last two focusing magnets, the final doublets, should not exceed a tenth of the beam size above 5Hz [1].

Major vibration sources like ground motion [2] and acoustic noise can induce displacements of a few nanometres above 5Hz. Thus, an active isolation of the ground and of the final doublets at their resonance frequencies must be carried out [3].

First, in order to stabilize final doublets to the sub-nanometre level, we have to compensate for the nanodisplacements induced by cultural noise. We consequently need sensors and actuators which are able to measure and create displacements of mechanical structures at the sub-nanometre level while being placed in a harsh environment composed of high magnetic fields and radiation. We also need a feedback loop which controls actuators from sensor data. In addition, mechanical simulations and dynamic response calculations are included in this study for defining the active stabilisation feedback loop. Also, although ground motion decreases with frequency, acoustic noise can be very high at high frequencies and some vibration studies have been consequently done on a canteliver beam to estimate the need of final doublets stabilization above 300Hz.

SENSOR ASSESSMENT

We started by assessing very sensitive, commercial vibration sensors, acquisition systems and signal conditioning for displacement measurements at the sub-nanometre level.

Instrumentation

When measuring nanodisplacements, resolution of the measurement chain is limited by internal noise of the chain itself, mainly composed of sensors and acquisition system noises. Consequently, these noises have been measured to evaluate sensors' and acquisition systems' performances. In table 1 and table 2, the characteristics and the measured noise of the three types of vibration sensors used by our team are given.

Table 1: Geophone characteristics

Type of geophones	**Electromagnetic**	**Electrochemical**
Model	GURALP CMG-40T	SP500-B
Company	Geosig	PMD Scientific
Sensitivity	1600V/m/s	2000V/m/s
Range (Hz)	[0.033; 50]	[0.0167; 75]
Measured noise for f>5Hz (nm)	0.05	0.05

Table 2: Accelerometer characteristics

Type of sensors	**Piezoelectric accelerometers**	
Model	ENDEVCO 86	393B12
Company	Brüel & Kjaer	PCB Piezotronics
Sensitivity	10V/g	10V/g
Range (Hz)	[0.01;100]	[0.15; 1000]
Measured noise for f>5Hz (nm)	0.25 >50Hz: 0.02	11.19 >300Hz: 0.005

*Work supported by the Commission of the European Communities under the 6th Framework Programme "Structuring the European Research Area", contact number RIDS-011899.
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Low frequency vibrations

Two types of commercial vibration sensors which are liable to measure nanodisplacements have been acquired: electromagnetic geophones using a servo loop to control the mass position and piezoelectric accelerometers coupled with sensitive charge amplifiers.

Because one measures velocity and the other measures acceleration, performances of these two types of sensors were compared to know in which frequency range they are the most sensitive with respect to ground motion.

Two GURALP geophones [4] and two ENDEVCO accelerometers [5] have been put side-by-side on the floor and their signals registered by an acquisition system (PULSE system [5] configured in 16 bits resolution from Brüel & Kjaer Company) of very low noise due to its integrated state-of-the-art electronics.

From these measurements, coherences [6] between the signals of the two GURALP sensors and of the two ENDEVCO sensors have been calculated. Coherence well below 1 means that signals are contaminated by instrumental noise because ground motion is coherent between two points close to each other and instrumental noise is not. Also, signal to noise ratios of these two types of sensors have been calculated for consistency. Results are shown in figure 1.

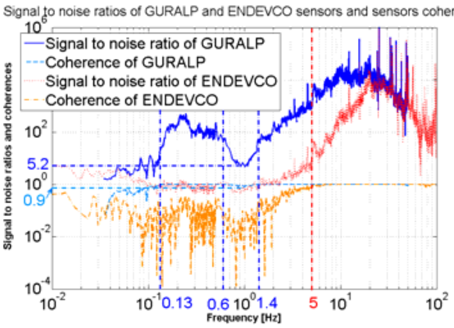

Figure 1: Signal to noise ratio of GURALP and ENDEVCO sensors and coherences

Signal to noise ratios and coherences of GURALP sensors are very good from 0.1Hz up to the upper limit of their frequency range (50Hz) but the ones of ENDEVCO sensors are good only above 5Hz.

Note that there is a very good consistency of results between signal to noise ratios and coherences even if calculations of coherences and of noises are from different measurements. The first calculation is from coherence measurements done the day. The noise measurements are done the night in order to get less ground motion signals and to have consequently a better estimation of sensor noise. This shows that the noise estimation done by either method gives good results.

To understand such difference of performances at low frequency between these two types of sensors, raw signals have to be analyzed because accelerometers measure acceleration and geophones measure velocity. In figure 2, the solid and solid thick curves represent respectively Power Spectral Density (PSD) [6] of ground velocity measured by GURALP geophones and PSD of ground acceleration measured by ENDEVCO accelerometers. The dashed and dash-dot curves represent respectively PSD of measured noises. As shown above, because PSD of ground acceleration could not be measured by ENDEVCO sensors at low frequencies, PSD of ground acceleration (dotted curve) has been computed by deriving ground velocity measured by GURALP sensors.

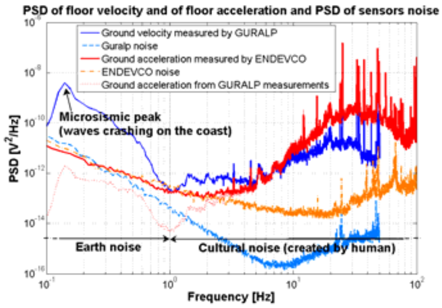

Figure 2: PSD of floor velocity, of floor acceleration and of sensors noise

At low frequencies, ground acceleration is very low compared to ground velocity and is below sensor noises (identical for both types of sensors), which explains why signal to noise ratio and coherence of ENDEVCO sensors were very bad. That means that ground motion has to be measured by geophones below 1Hz and can be measured by both types above a few Hertz. In order to perform different vibratory studies in a wide frequency range, for instance ground motion study or evaluation of the STACIS commercial active isolation system presented in figure 14, we use GURALP geophones to measure vibrations below 1Hz to 50Hz and ENDEVCO accelerometers to measure vibrations from a few Hertz up to 100Hz.

High frequency vibrations

Another model of accelerometers, the high frequency 393B12 accelerometers [7], has been acquired by our team to perform vibratory studies of a cantilever beam at high frequencies (see the *Acoustic noise study* section).

Figure 3 represents the floor acceleration PSD measured by these sensors with the PSD of their measured noise (left plot) and the signal to noise ratio of the sensors with their coherence (right plot).

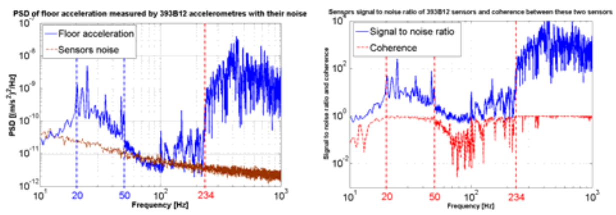

Figure 3. Floor acceleration PSD measured by 393B12 sensors with their noise PSD (left) and signal to noise ratio of the sensors with their coherence (right)

Ground acceleration increases above 200-300Hz and 393B12 noise decreases with frequency, which allows having a high signal to noise ratio and consequently accurately measuring ground motion at high frequencies.

This is confirmed by the good coherence obtained above the same frequency. Note that signal to noise ratio and coherence results are consistent as mentioned in the *Instrumentation* section.

Sensors and instrumentation for active rejection

Because electromagnetic geophones and piezoelectric accelerometers are sensitive to high magnetic fields, a collaboration with PMD Scientific Company and SLAC laboratory has been created to develop electrochemical vibration sensors, the SP500 sensors [8], not sensitive to such environment for the active stabilisation of the future linear collider final doublets.

To acquire data of SP500 sensors and to control actuators for the active rejection of our prototype, a basic acquisition system (the DAQ PCI6052E from NI [9] with 16 bits resolution) but compatible with Matlab/Simulink (the software used to develop our feedback loop) has been used. It has been equipped with signal conditioning, that is to say active high-pass and low-pass filters and active amplifiers from Krohn Hite Corporation [10], in order to have sufficient resolution and very low noise.

To know if the sensitivity of SP500 sensors and of the acquisition system is sufficient, their noise has been measured. For that, we measured noise of the complete measurement chain by using the Corrected Difference method [11]. For the acquisition system noise, measurements were done by putting 50 ohm adapters on its inputs. By subtracted the acquisition system noise to the measurement chain noise, we obtained SP500 noise. Results of integrated Root Mean Square (RMS) [12] of the measurement chain noise and of the SP500 noise are given in figure 4.

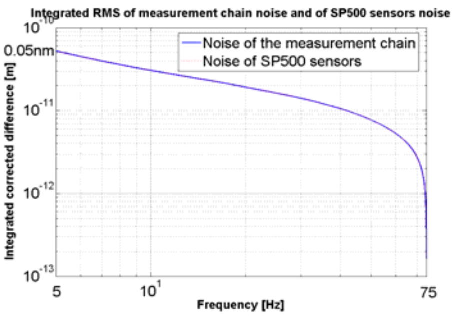

Figure 4: Integrated RMS of the measurement chain noise and of SP500 noise

This figure shows that noises of the measurement chain and of SP500 sensors are quite the same and are of 0.05nm above 5Hz. Consequently, DAQ PCI6052E system has a very low noise compared to sensor noise and consequently does not degrade sensor performances. This shows that our signal conditioning was very efficient.

Also, our measurement chain allows us to measure displacements from 0.05nm to 500nm above 5Hz. This dynamic range is sufficient to measure vibrations of structures subjected to ground motion down to the sub-nanometre level and this measurement chain has been consequently used in the vibration active rejection of our prototype.

Now that the sensor performances have been found to be compatible with ground motion measurements, vibration study of a canteliver beam subjected to ground motion and to acoustic noise has been performed at high frequencies.

ACOUSTIC NOISE STUDY

Ground motion decreases with frequency and vibration studies have been consequently focused until now on the highest motions, that is to say below 300Hz.

However, contrary to ground motion, acoustic noise does not decrease with frequency and can be very high at high frequencies in a collider with equipment switched on.

Therefore, because of final doublet resonances induced at high frequencies, ground motion and acoustic noise may make these final doublets vibrate beyond tolerances for frequencies above 300Hz.

In this section, the impact of ground motion and of acoustic noise on the vibrations of a structure representative of one of the linear collider final doublets has been studied in the bandwidth of a resonance in the aim to estimate the need of stabilization at high frequencies.

Choice of the resonance of study

The study was done on a canteliver beam of one meter long made of aluminium in order to have the same boundary conditions than those fixed for the linear collider final doublets.

In order to identify resonant frequencies of the beam and to choose upon which the study will be focused, some vibration measurements have been done outside the working hours with two accelerometers (of type 393B12) fixed to the free end of the beam and to the clamping.

In addition, to have an idea of the acoustic noise, some measurements of acoustic pressure have been done with two microphones (of type 4189 [5]) put near the clamping and near the free end of the beam.

The experimental set-up was done in our laboratory which has very low noise: it is of 5nm on average above 5Hz [13]. A photography of the set-up is shown in figure 5.

Figure 5: Canteliver beam instrumentated with two accelerometers and two microphones

From the accelerometer measurements, the displacement Amplitude Spectral Density (ASD) of the beam free end and of the clamping have been calculated and plotted in figure 6. Because of the low amplitudes of measured vibrations, the noise ASD of the whole measurement chain has been plotted in the same figure in order to show that these accelerometers have a sufficient resolution to measure the vibrations at the free end of the beam and at the clamping.

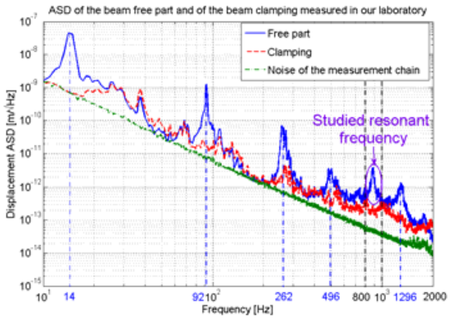

Figure 6: Displacement Amplitude Spectral Density (ASD) of the beam free end and of the clamping

This figure shows that the noise of the measurement chain (green curve) is lower than the measured signals (red and blue curves). Consequently, the resolution of the measurement chain is sufficient to measure vibrations of the beam and of the table in these conditions.

Also, the free end of the beam has some vibration peaks (blue curve) that the clamping does not have (red curve). These peaks correspond thus to the resonances of the beam. Among others, the beam has a resonance at 881Hz with a bandwidth of [800; 1000] Hz.

The displacements of the clamping are very low in the bandwidth of the resonance at 881Hz compared to the displacements below 300Hz.

Noise amplitude

In order to have a value of the level of acoustic noise, of the amplitude of the clamping displacement and of the impact that these ones can have on the value of the beam displacements in the frequency range of study ([800; 1000] Hz), the integrated Root Mean Squares (RMS) of displacement and of acoustic noise have been calculated from measurements done in the previous section.

In figure 7, results are shown for the acoustic noise measurements.

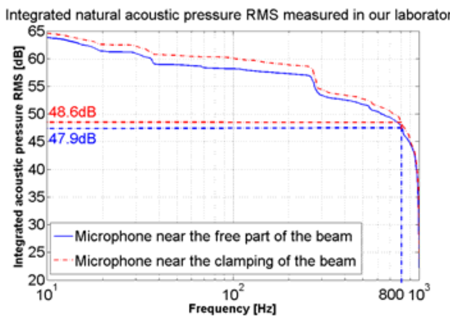

Figure 7: Integrated Root Mean Square (RMS) of acoustic pressure near the beam free end and near the clamping

In the frequency range of study, that is to say between 800Hz and 1000Hz, the acoustic level was of 48dB near the free end of the beam and near the clamping, which represents a low acoustic level.

The integrated displacement RMS of the beam and of the clamping have been calculated and plotted in figure 8. The integrated RMS of the measurement chain noise has been plotted in the same figure in order to have a value of the measurement chain resolution in the frequency range of study.

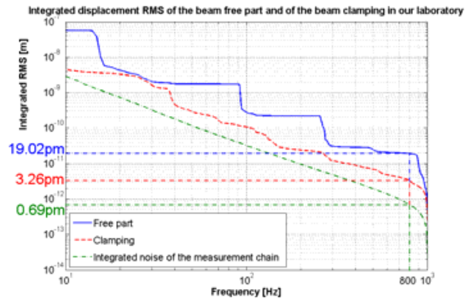

Figure 8: Integrated displacement RMS of the beam free end and of the clamping

First of all, figure 8 shows that the resolution of the measurement chain is only of 0.69pm in the frequency range of study.

The displacement of the clamping (red curve) is of 3.3pm between 800Hz and 1000Hz, which is very low compared to tolerances of relative motion fixed in a linear collider (one tenth of a nanometre).

However, the displacement of the beam free end (blue curve) is of 19pm between 800Hz and 1000Hz, which is just a factor 5 below tolerances fixed in a linear collider.

Because acoustic noise was very low in our laboratory and certainly much lower than in the future linear collider, it is very important to evaluate the impact of acoustic noise on the displacements of the canteliver beam when this noise has higher amplitudes.

Acoustic noise of higher levels

In order to obtain acoustic noise of higher levels than those of our laboratory, a loudspeaker was used to create acoustic noise.

We have chosen to use the loudspeaker to create a sinusoidal noise at 881Hz at different levels which is similar to the noise of a pump.

Figure 9 shows the experimental set-up in our laboratory.

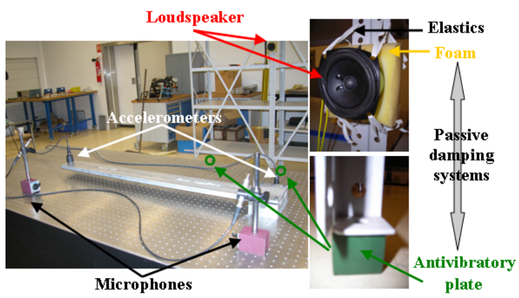

Figure 9: Experimental set-up to study the impact of acoustic noise coupled with ground motion on the vibrations of a beam

The canteliver beam was fixed to the honeycomb table of the TMC Company to provide a clean working surface.

An accelerometer (of type 393B12) was fixed at the free end of the beam while the other one was fixed to the clamping in order to measure one of the two excitation sources of the beam: the mechanical vibrations of its support.

Two microphones (of model 4189) were used to measure the other excitation source of the beam: the acoustic noise applying a pressure directly on it. One of them was fixed near the free end of the beam while the other one was fixed near the clamping in order to check that acoustic pressure was uniformly spread along the beam.

The loudspeaker has been put at two meters from the beam. Because the mechanical vibrations of its membrane can be transmitted through the support to the floor and make the beam clamping vibrate, we therefore isolated the loudspeaker in order to reduce mechanical vibrations by using some passive damping systems filtering vibrations of the loudspeaker at medium and high frequencies. For that, the loudspeaker was fixed with elastics and some foam was put between the loudspeaker and its support. Also, each foot of the loudspeaker support was put on an anti-vibratory plate of type B32 [14] of Bilz company beginning damping vibrations above 10Hz.

For each level of noise created by the loudspeaker, a simultaneous acquisition of the four sensor measurements was performed by the PULSE system outside the working hours.

Acoustic noise impact on the beam vibrations

In order to have the displacement values of the beam free end and of the clamping versus the levels of acoustic noise created by the loudspeaker, the integrated displacement RMS of the beam free end and of the clamping have been calculated versus the integrated RMS of acoustic pressure (in dB) in the frequency range of study ([800; 1000] Hz). These calculations have been performed from the measurements done for different levels of noise. Results are shown in figure 10.

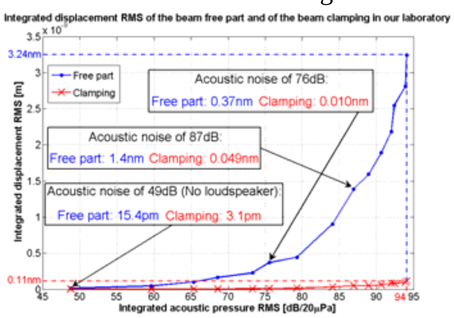

Figure 10: Integrated displacement RMS of the beam free end and of the clamping versus the integrated RMS of acoustic pressure (in dB) near the beam free end

This figure shows that the displacements of the beam free end increase significantly with the levels of noise created by the loudspeaker while the displacements of the clamping stays almost constant.

For an acoustic pressure going from 49dB to 76dB (representative of what can be found in a collider), the displacement of the clamping goes from 3.1pm to 10pm, that is to say an amplification factor of 3 whereas the displacement of the beam goes from 15.4pm to 0.37nm, that is to say an amplification factor of 24. Consequently, the beam displacements increase of a factor 3 due to the increase of the clamping displacements and of a factor 7 due to the acoustic noise applying a force directly on the beam. Consequently, if just taking into account the increase of acoustic noise and assume that ground motion is very low (3.1pm), the beam displacements would increase from 15.4pm to 0.11nm (factor 7) and would be thus above the relative motion tolerances of CLIC final doublets (0.1nm).

In order to see the increase of the beam displacements only due to the direct impact of acoustic noise for its different levels, the ratio between the integrated displacement RMS of the beam free end and the one of the clamping has been calculated versus the integrated RMS of acoustic noise. Results are shown in figure 11.

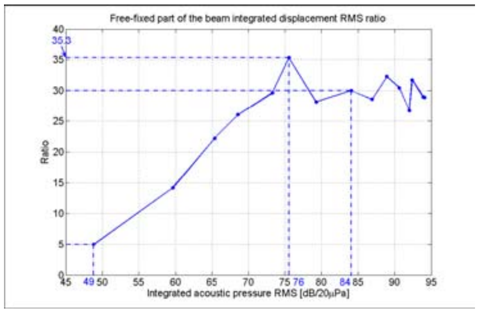

Figure 11: Ratio between the integrated displacement RMS of the beam free end and of the clamping versus the integrated acoustic pressure RMS (in dB) near the beam free end

For an acoustic pressure of 46dB (no loudspeaker and measurements outside working hours), acoustic noise has probably a very low impact on the beam displacements. The ratio between the beam displacements and ground motion is of 5 and represents the amplification factor of the beam resonance because the system is composed of only one input, ground motion and only one output, the beam displacements.

Now, when acoustic noise increases up to a level of 76dB, this ratio increases up to a factor 35 and shows that acoustic noise has a direct impact on the beam displacements.

To conclude, acoustic noise can have a big impact on the displacements of a canteliver structure at high frequencies. For an acoustic pressure representative of what can be found in a linear collider and even for a linear collider site where ground motion is very low (like the one of our laboratory), final doublets can vibrate above relative motion tolerances of CLIC final doublets up to at least 1000Hz.

Now that some hands-on experience has been acquired on the vibratory behaviour of a simple canteliver beam, numerical simulations of a canteliver structure closer to the design of future linear collider final doublets have been performed in order to control the vibrations of this structure afterwards.

NUMERICAL SIMULATIONS

Numerical simulations can be a great help to test the efficiency and the robustness of the active control algorithm in realistic conditions. The main objective is to obtain a state-space model of the structure to control, in order to use it in Matlab/Simulink to get dynamic response of this structure under predefined loads.

To do this the first step can be the finite element modelling of the structure.

Finite Element Model

Finite element modelling is of prime importance, insofar as the finite element model is required for the future results to be representative. Indeed, the state-space model will use the formulations of the finite element model (FE model).

In order to get the most realistic results (in terms of dynamic and control), the FE model must be as accurate as possible.

Consequently, updating the FE model is a step of the utmost importance. Thus, experimental vibration measurements are required to get, on the one hand the different eigenfrequencies and their corresponding mode shapes, and on the other hand their level of damping. Then a model updating can be performed.

Note that most of the time, the use of Super-Element can be realized to reduce the size of the system to solve, which is a non-negligible aspect for the future dynamic computations.

State-Space Model

The State-Space model results exclusively from the FE model (1), namely the mass, damping and stiffness matrices without forgetting the external applied loads. The latter act as the input of the model, the output being the motion of some predefined locations (in terms of acceleration, velocity and displacement). The fundamental equation describing the dynamic behaviour of a structure is:

$$M\ddot{q}(t)+B\dot{q}(t)+Kq(t)=g(t) \tag{1}$$

where the q(t) state vector collects the displacements of the structure by degree of freedom, while the g(t) vector indicates the corresponding applied loads. The M matrix corresponds to the mass, the B matrix to the damping and the K matrix to the stiffness.

The state-space model will have the following form, assuming that only external forces can be applied to the model:

$$\begin{Bmatrix} \dot{x}=Ax+Bu \\ y=Cx+Du \end{Bmatrix} \tag{2}$$

where the state vector x, the input vector u, the output vector y, the different matrices A, B, C, D and the acceleration vector $\ddot{q}$ are defined in [15].

In the general method, it is assumed that only external forces are applied to the structure. Nevertheless, an extended method has been proposed [15], in which external disturbances can be not only pinpoint forces, but also prescribed acceleration for instance. Then, the system to solve is now written:

$$\begin{aligned} &\begin{bmatrix} M_{11} & M_{12} \\ M_{21} & M_{22} \end{bmatrix}\begin{Bmatrix} \ddot{q}_1 \\ \ddot{q}_2 \end{Bmatrix}+\begin{bmatrix} B_{11} & B_{12} \\ B_{21} & B_{22} \end{bmatrix}\begin{Bmatrix} \dot{q}_1 \\ \dot{q}_2 \end{Bmatrix}+\dots \\ &\dots\begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix}\begin{Bmatrix} q_1 \\ q_2 \end{Bmatrix}=\begin{Bmatrix} g_1(t) \\ -r_2(t) \end{Bmatrix} \end{aligned} \tag{3}$$

where the index 2 stands for the *dof* where acceleration is applied and index 1 for all other *dof* of the model. Moreover, g_1(t) represents the possible external point forces. The first equation of the system (3) allows the dynamic response computation, provided one carries over the right hand side forces of inertia, dissipation and stiffness associated with the prescribed motion:

$$\begin{aligned} &M_{11}\cdot\ddot{q}_1+B_{11}\cdot\dot{q}_1+K_{11}\cdot q_1=\dots \\ &\dots g_1(t)-M_{12}\cdot\ddot{q}_2-B_{12}\cdot\dot{q}_2-K_{12}q_2 \end{aligned} \tag{4}$$

Hence, in a general way, the input vector u of the state-space model will put together the terms of point forces and the terms of forces of inertia, dissipation and stiffness, which corresponds to the right hand side terms of (4).

Finally, by correctly initializing the different matrices of the State-Space model in Simulink, it is possible to get the dynamic response of the structure under prescribed acceleration. The active control can be coupled to this computation by adding for instance pinpoint forces (if pinpoint actuators are required) in the input vector of the state-space model. Figure 12 shows a vibration active control simulation of a canteliver structure at its free end, as a function of time. Vibrations of the structure are due to a sinusoidal load at its first resonant frequency and, for reasons of understanding, the controller is activated when the process is in steady state. Without active control, the beam free end has consequently the same velocity magnitude and when the active rejection is applied, its velocity magnitude decreases.

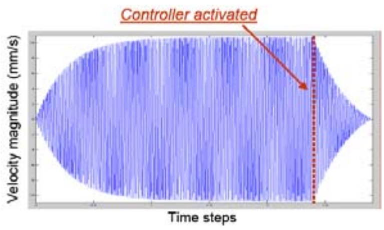

Figure 12: Simulation of canteliver structure velocity at its free end, without and with active control, as a function of time

The active control algorithm will be described in the next section.

ACTIVE STABILIZATION

In order to obtain a very low displacement of the future linear collider final doublets (one tenth of a nanometre in the vertical axis and in the desired frequency range [5; 100] Hz), a lot of constraints have to be considered. The complexity of the mechanical structure and the multitude of perturbation sources are the two main aspects of this problem.

Concerning the mechanical structure, the design of the future linear collider is not yet finalised. However, this will be very complex, so a few intermediate stages are necessary. This is why this study aims to obtain a very low displacement all along an elementary mechanical structure which is similar to the future final doublets in the main aspects that concern our work. The prototype used for this experiment is a 2.5m long steel beam in cantilever mode, respecting the elementary parameters planned for the final doublets. Furthermore, the eigenfrequencies of this linear structure are included in the desired frequency range. The measurement of the motion is performed with the SP500-B electrochemical geophones presented in the *sensor assessment* part. Concerning the actuators, assemblies of piezoelectric patches (APA 25XS from the CEDRAT Company) are used. They allow creating very low displacements at the sub-nanometre scale all along the beam. The built prototype is presented in figure 13.

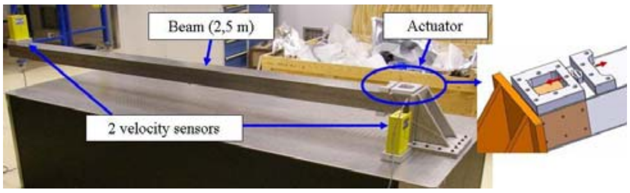

Figure 13: The built prototype with clamping and free end

In order to attenuate the motion of this prototype, the influence of the perturbations has to be analysed. In fact, there are two types of motions that can be identified:

- The vertical motion of the clamping created by ground motion. Its effect excites indirectly the mechanical structure, mainly its resonant modes.
- The motion of the mechanical structure itself created by acoustic perturbations which excite it in all directions.

Active isolation from the ground

In order to deal with the two aspects of the problem, two methods are used. First of all, the purpose of our study is to obtain a very low displacement of the clamping by the use of passive and active isolation, in order to isolate the whole system from ground motion [16]. For that, an industrial active table has been tested [17]. This is an active table produced by TMC Company with four STACIS active isolators (see figure 14). Even if this table is really efficient (For example: 2.7nm on the floor versus 0.17nm on the table above 5Hz), this solution is not sufficient given the very strict allowed tolerances (1/10nm) and is too large considering the crowded environment.

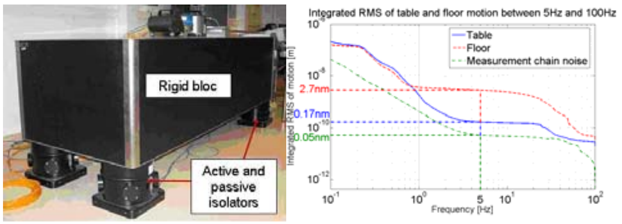

Figure 14: The active table TMC and an example of an obtained integrated displacement RMS

Active rejection of resonances

Indeed, this approach does not consider the acoustic perturbations and even the slightest motion of the clamping will be amplified by the structure, mainly at its resonant modes. This is why active compensation has been developed and the approach and the results are presented in this paper. The proposed method consists in applying a force that creates a motion in opposition with the motion created by the perturbations. This will maintain the mechanical structure in a straight horizontal position along its axis.

Because of the complexity of the structure, it is considered that it is too complicated to compute a fine model representative of the system. The originality of the proposed algorithm is that it takes into consideration only the measurable behaviour of the system and that they do not require an accurate complete model of the structure.

This algorithm is based on a command with internal model [18]. It uses only an elementary model which is representative for the structure behaviour and for a given bandwidth corresponding to a resonant mode. For the purpose of controlling all the desired range, there are as many algorithms as there are frequencies or bandwidths to process. The adaptation of the command with internal model control for one bandwidth is described in figure 15.

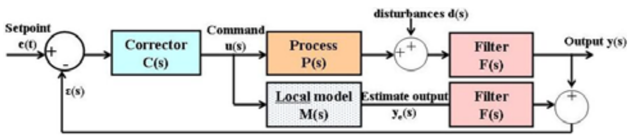

Figure 15: Adaptation of a classic command with internal model control

This algorithm was tested in simulation, then with the large prototype at the nanometre scale in a natural environment. Two bandwidths were processed, each of them corresponding to a resonant mode of the mechanical structure (12Hz and 68 Hz). Figure 16 represents the transfer function between the measured displacement at the end of the beam and the measured displacement at the clamping, with and without rejection (left plot) and the integrated displacement Root Mean Square at the clamping and at the end of the beam with and without rejection (right plot).

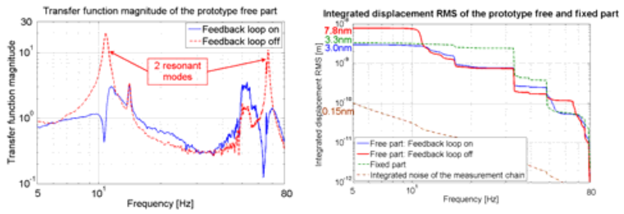

Figure 16: Transfer function between the motion at the end of the beam and at the clamping (left) and the integrated displacement RMS with and without rejection (right)

These results reveal that for the two treated bandwidths the algorithm is efficient, since the amplification is considerably reduced. However, the results can be improved, because the processing of a bandwidth has a small detrimental influence on neighbouring frequencies. Considering these results, the combination of active compensation with active isolation was tested in order to investigate if the approach can be applied at the sub-nanometre scale.

In this prospect, the prototype was fixed on the active table. Figure 17 represents the obtained results.

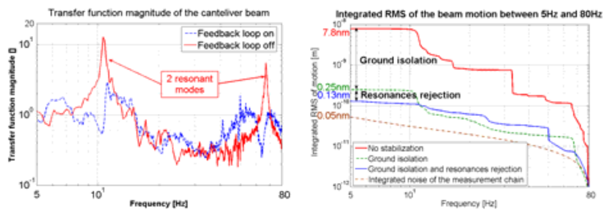

Figure 17: Transfer function of the beam (left) and the integrated displacement RMS obtained with the combination of active compensation and active isolation (right)

Because of the active isolation, the measured displacement at the end of the beam (without active compensation) is lower than a nanometre (0.25 nm). Even if this displacement is already very low, we also apply the active compensation and the obtained results reduce the motion at approximately the same ratio as before. The result is a very low displacement, actually an absolute stabilization about a tenth of nanometre. This test proves that the instrumentation is not a limitation and that it is possible to stabilize at the tenth of nanometre scale. The next objective is to obtain these results not only on a selected point of the beam, but all along its length.

CONCLUSION

Thanks to some electrochemical vibration sensors and piezoelectric actuators associated with an appropriate instrumentation, a control algorithm developed by our team and a real time apparatus, the feasibility of actively rejecting structure vibrations down to 0.1Hz has been proven by using in parallel a commercial system performing passive and active stabilization of the clamping. The design of the linear collider final doublets is not finished but the tools developed by our team, including the simulation of the whole system, will allow us to follow their evolution. Moreover, the mechanical modelling will give us information about optimal location of sensors and actuators for the active rejection of structures all along their length. To finish, a study of acoustic noise done on a canteliver beam showed that this type of noise can make the final doublets of the future linear collider vibrate above CLIC tolerances up to at least 1000Hz. Stabilization of these magnets may have to be done even at high frequencies.

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