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LAPP-TECH-2007-01
September 2007

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Presented by L. Brunetti
at the International Linear Collider Workshop LCWS/ILC 2007
Hamburg (Allemagne), 30 May - 3 June 2007

Active Stabilization of a Mechanical Structure

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This article [1] refers to a particular stage of our attempt to reach the stabilization of the linear collider final focus quadrupole. All along this final focus, an absolute displacement has to be lower than the third of nanometre above a few hertz. The presented intermediary step consists in doing active vibrations control of an elementary mechanical structure in cantilever mode which is similar to the final focus. We consider mainly the active compensation and the latest results on a large prototype. Other aspects are also treated such as modelling, active isolation and instrumentation dedicated to the ground motion.

1 The context and the approach

In order to attenuate the motion at the end of a mechanical structure in cantilever mode, the sources of the displacement have to be analysed. In fact, there are two different sources of disturbance, as presented by figure 1.

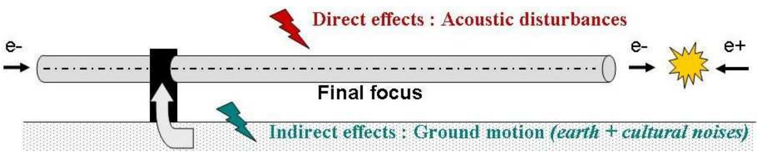

Figure 1: Different sources of disturbance which excite a mechanical structure

The main perturbation is the ground motion, which is in fact a combination of the earth motion lower than 1 hertz and of the cultural noise (pumps, motors, etc.) starting above 1 hertz. This disturbance will be transmitted by the clamping to the structure. Consequently the whole system (clamping and structure) is mainly excited vertically. The second influence is the acoustic perturbations which have a direct effect on the structure and which will excite it in all directions, affecting all resonant modes. Figure 2 represents the ground motion (left figure) and the acoustic pressure without added external disturbances (right figure) measured at LAPP.

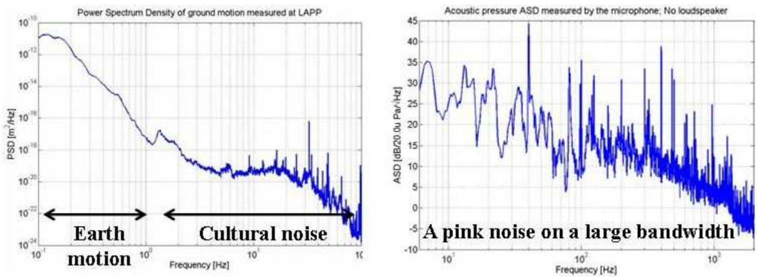

Figure 2: Example of ground motion (left) and acoustic pressure (right) measured at LAPP

As a consequence, we can differentiate two types of motion: the motion of the entire system via the vertical displacement of the clamping due to ground motion [2] and the motion of the mechanical structure itself due to acoustic disturbances [3]. First of all, the stabilization requires isolating the whole system from the ground using active and passive isolation in order to attenuate the vertical displacement of the clamping. In this manner, an almost null displacement of the clamping can be obtained. There is still to cancel the direct effects applied to the structure itself, in order to maintain it in a straight horizontal position along its axis. The only possible solution is then the active compensation, meaning to apply a force that creates a motion in opposition with the motion created by the acoustic disturbance. This study refers mainly to the active compensation, because there are already industrial solutions for the active isolation. The former will be nevertheless treated in the last part.

2 First approach for active compensation

2.1 The first algorithm

For the purpose of controlling the structure motion using the active compensation, one needs an adapted algorithm. In automatics, a controller is generally based on a representative model of the process by optimising the transfer of the system. Considering the final focus, it is too complicated to compute a fine model which is able to reproduce with accuracy the behaviour of this complex mechanical structure, so we consider that the model is unknown. The innovation of this algorithm consists of working only with the measurable behaviour of the process. Furthermore, we have taken into consideration that the measured motion of the mechanical structure does not present itself as a white noise, but is composed of many independent frequencies, some of which are amplified. We have then decided to control independently every main frequency. That means there must be as many algorithms that run at the same time as there are frequencies to reject.
This algorithm is based on the estimation of the effects of disturbances and it computes a sinusoidal control for each disturbance frequency [4]. The principle for one frequency is described in figure 3.

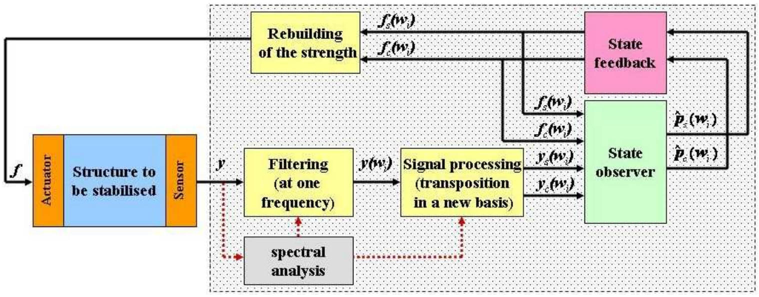

Figure 3: The initial developed algorithm for active compensation

2.2 The experimental setup

In order to validate this algorithm and to analyse its efficiency and robustness, the prototype presented in figure 4 was built [4].

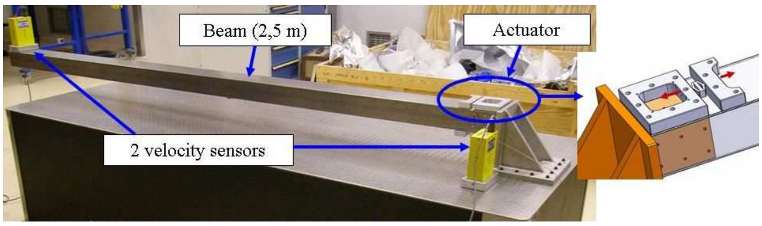

Figure 4: The built mock-up which has similar geometry to the final focus quadrupole

The prototype is composed of a 2,5 meters long steel beam in cantilever mode, just as planned for the final focus. To define the instrumentation, one has to consider the required accuracy and study the impact on the mechanical structure. The velocity sensor SP500B has been chosen, because of its light weight, its small size and its magnetic insensitivity. This sensor is able to measure ultra-low level vibrations because of its very low noise: 0.085 nm integrated noise from 4Hz to 75Hz measured at LAPP. Concerning the actuators, assemblies of piezoelectric patches (APA 25XS from the CEDRAT Company) are used. They allow creating very low displacements at a nanometre scale all along the beam.

2.3 The first results at a nanometre scale

The sensor used for the measurement in the feedback loop was placed at the end of the beam, the part that needs to be stabilised. A second sensor is placed on the clamping in order to measure the perturbation given by the ground motion. We also use an actuator located as close as possible to the clamping, in flexion mode. Figure 5 represents the result of the stabilization in a natural environment, with no added external disturbances. The first two modes of flexion of the beam can be recognized (large peaks) and a lot of unknown other disturbances can be noticed (narrow peaks).

For this test, one of the narrow peaks has been arbitrarily selected (the surrounded peak) and we can observe that the rejection is efficient. It is possible to parallelize the algorithms that reject each of these narrow peaks, in order to reduce as much as possible the motion of the mechanical structure. As a conclusion, we can state that this algorithm is able to reject narrow peaks at a nanometre scale. However, for the eigenfrequencies, this method is quite limited, because working at selected frequency is not sufficient enough to treat a bandwidth.

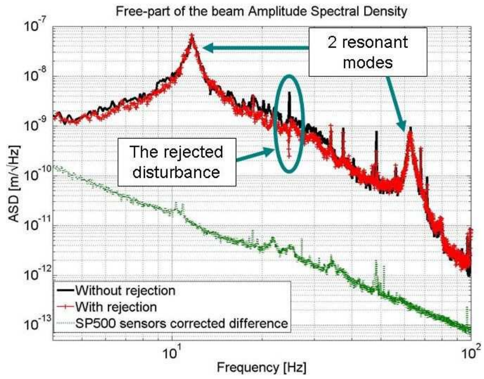

Figure 5: ASD of the displacement at the end of the beam with and without rejection

3 Improvement by using a new algorithm

3.1 The method

The aim of this chapter is to complete the previous method in order to be able to treat all sorts of disturbances, including the frequencies which correspond to the resonant modes of the structure. Because of its complexity, a complete model of the system is too difficult to obtain. Moreover, it is quite limited to work only at a selected frequency, so an intermediary solution was chosen, which consists in using a local model for the different large peaks (meaning the resonant modes of the structure). This algorithm is based on a command with internal model control [5] as described in figure 6 :

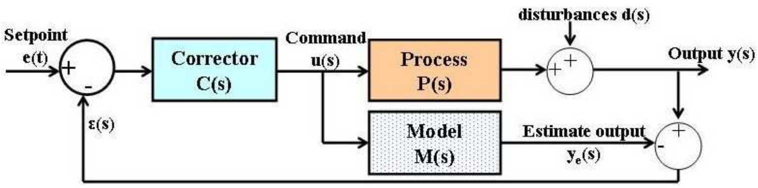

Figure 6: Principle of a classic command with internal model control

It is necessary to consider the transfer function in closed loop of this algorithm :

$$Y(s) = \frac{C(s)P(s)e(s) + [1 - C(s)M(s)]d(s)}{1 + [P(s) - M(s)]C(s)} \quad (1)$$

This equation allows revealing the two fundamental rules of this algorithm. First, in order to follow the setpoint, the controller should be exactly equal to the inverse of the model. Next, to cancel the disturbances, the model doesn't require being exactly equal to the process but only to be an approximation of it, which is an important advantage considering the difficulty in estimating an accurate model. To reach our specific problem of stabilization, this method has been adapted to our needs. The particularity of our approach is that one algorithm is dedicated only to a bandwidth, so that, instead of using a complete model, only a local one which represents the process on the desired bandwidth is defined. The advantage is that this local model can be defined, having only a basic knowledge of the process behaviour. This knowledge is easy to estimate in experimental mode. The adaptation of the command with internal model control for one bandwidth is described in figure 7 . Notice that the measured and the estimated signals are filtered with a band pass filter in order to process only the desired bandwidth without disturbing the neighbour frequencies. Furthermore, different algorithms can be run in parallel in order to stabilize different bandwidths.

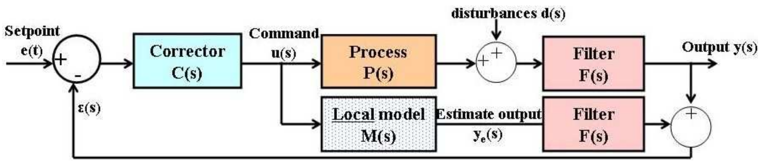

Figure 7: Adaptation of a classic command with internal model control

3.2 Test of the algorithm using a finite element model of the prototype

First of all, this feedback loop algorithm has been tested using a numerical approach in order to evaluate its efficiency. For this step, we computed a finite element model of the prototype under the software SAMCEF. This allowed us to obtain the dynamics equation:

$$M\ddot{u}(t) + C\dot{u}(t) + K(t) = f_p(t) \tag{2}$$

where M is the mass matrix, C is the damping matrix and K the stiffness matrix.

This method allows predicting with accuracy the mechanical structure response in all points in terms of displacement, velocity and acceleration. From the previous dynamics equation, a state-space model is created, using the MATLAB software [6], which can be integrated in a SIMULINK application. It can then be executed with the feedback loop in a simulation test application, as presented in figure 8. An efficient tool is obtained this way. It allows adjusting the feedback loop, increasing the test possibilities (multiple configurations for instrumentation) and analysing the behaviour of the entire beam during an active vibration control.

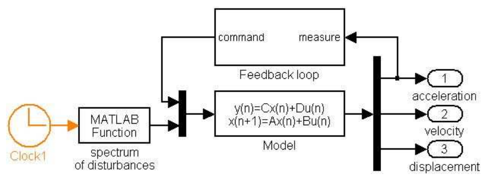

Figure 8: Simulation of the feedback loop with the computed model of the prototype

3.3 Experimental tests

Once this simulation step was completed, the developed feedback-loop algorithm was tested on the large prototype, in the same configuration as previously, with a natural environment.

Two bandwidths were processed, each of them corresponding to a resonant mode of the mechanical structure (12 and 68 Hz). Figure 9 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).

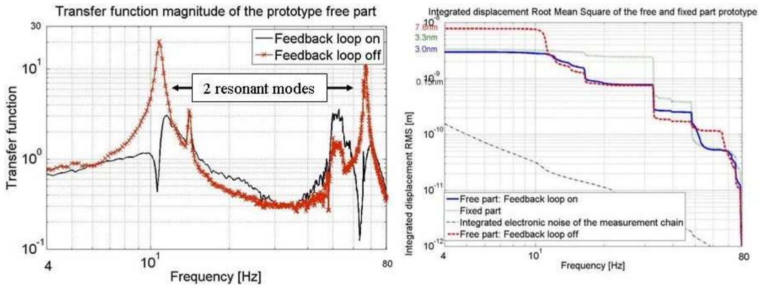

Figure 9: Transfer function between the motion at the end of the beam and the one at the clamping (left) and the integrated displacement RMS with and without rejection (right)

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. Even without additional optimization, the displacement at the end of the beam is equal or even lower than the displacement of the clamping. It is expected that when the adjustments will be finished, the absolute displacement of the mechanical structure will be lower than the nanometre, by performing only the active compensation.

4 Combination between active compensation and active isolation

There are two manners of attenuating the motion of a mechanical structure: active compensation and active isolation. The aim of this part is to present the latest experiment done by combining active compensation with active isolation using an industrial solution: an active table with 4 STACIS active isolators, produced by the company TMC [7] (see figure 10).

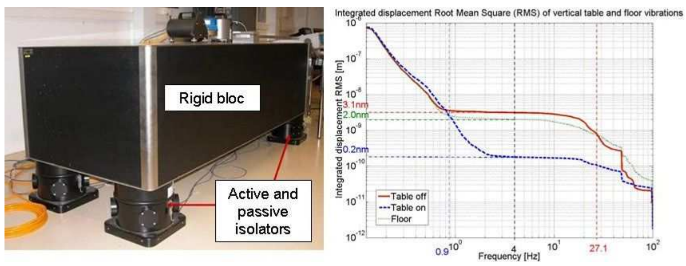

Figure 10: The active table TMC and an obtained integrated displacement RMS

When the beam is placed on this active table, a very low displacement on the clamping is obtained, of about 0.16 nanometres (3 nm with the table OFF). The measured displacement at the end of the beam (without active compensation) is lower than a nanometre (0.25 nm). This displacement is already very low. We also applied the active compensation at approximately the same ratio as previously. The result is a very low displacement, actually an absolute stabilization about a tenth of nanometre, as can be observed in figure 11. This test proves that the instrumentation is not a limitation and that it is possible to stabilize at the tenth of nanometre scale. Now, the objective is to succeed in obtaining the same results but in an environment with more perturbations, meaning when the motion of the structure, is about a few tens of nanometres. Another target is to obtain these results not only on a selected point of the beam, but along the whole length.

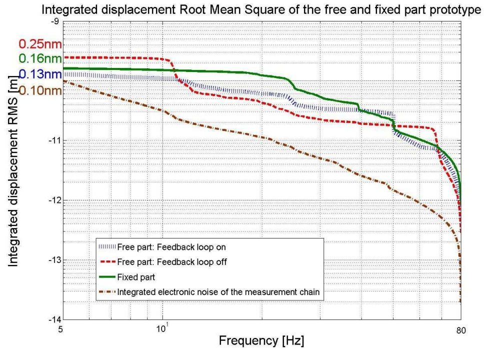

Figure 11: Integrated displacement RMS obtained with the combination of active compensation and active isolation

5 Other development

The active compensation is not enough for treating the case of the future linear collider final focus, because the ground motion (so the clamping displacement) in many measured sites [8] is already greater than the imposed tolerance. In this context, because of the price and the magnetic sensitivity of industrial active tables, we have begun to study the possibility of developing a low cost table [9].

In fact, the principle is based on a mixture of passive isolation and an active solution using actuators. The passive isolation material acts like a low resonant filter, meaning that it attenuates all the high frequency disturbances but amplifies the low frequency ones. Consequently, the high frequency disturbances are already attenuated by the passive solution, so it remains only to reject the low frequency amplified disturbances using the active isolation. In order to reproduce this phenomenon, a small elementary active table has been developed (see figure 12).

This table is composed of two superposed layers. The first one for passive isolation with rubber and the second one for active isolation with actuators (APA25XS). Furthermore, two accelerometers (Endevco 86) are placed on the base and on the table in order to measure the motions at these given points. The figure 13 presents the transfer function between the displacement measured by the sensor on the base of the small table and the displacement measured by the second sensor on the top of it.

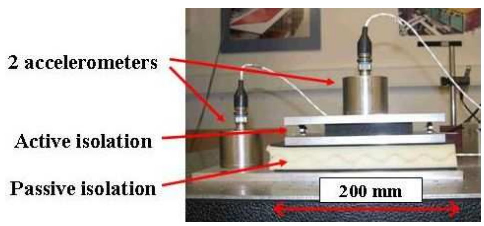

Figure 12: The small elementary active table

We can notice that the behaviour of this rubber (passive isolation) is similar to the behaviour of a resonant low pass filter.

For the active part, an adapted algorithm is required. Contrary to the active compensation, in this case it is possible to determine a model of the built table, so a classical algorithm can be used. We have selected an algorithm (LQG) [10], adapted to noisy systems, so to the seismic domain. This part is currently under development and the next step will be to transpose this study on a large mock-up with industrial rubber, in order to offer a complete solution with the active compensation.

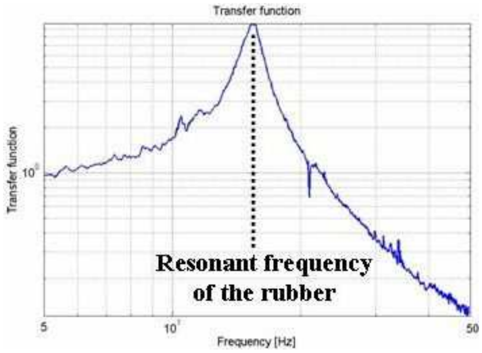

Figure 13: Transfer function on our table

6 Acknowledgments

This work is supported by the Commission of the European Communities under the 6th Framework Programme "Structuring the European Research Area" (RIDS-011899).
We also thank the CERN team for allowing us to use their TMC active table.

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