

ACTIVE STABILIZATION STUDIES AT DESY

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Abstract

All Linear Collider schemes currently under study are aiming at luminosities of some $10^{34} \text{ cm}^{-2} \text{ sec}^{-1}$. These can only be achieved by very tiny beam sizes of some 100 nm width and 10 nm height at the interaction point (IP). This requires high beam position stability in order to provide central collisions of the opposing bunches. Since ground motion measurements indicate that amplitudes are likely to exceed the required tolerances for the S-Band Linear Collider SBLC, some means of active stabilization of mechanical quadrupole vibrations is necessary to achieve sufficient orbit stability. Therefore an inexpensive active stabilization system to be installed in the main linacs has been developed and successfully tested. This system is capable of damping quadrupole motion in the frequency range between 2 Hz and 30 Hz by up to 14 dB, thus resulting in rms vibration amplitudes well below the required tolerances. As an alternative method, electronic feedforward of seismometer signals to corrector magnets is also briefly discussed.

1 Introduction

To achieve high luminosities of some $10^{34} \text{ cm}^{-2} \text{ sec}^{-1}$, all Linear Collider schemes currently under study require very low emittance beams which are focused to beam sizes of roughly 100 nm width and 10 nm height at the interaction point. These tiny beam dimensions require extremely tight mechanical jitter tolerances in order to keep the beams in collision. Assuming a β -function scaling with energy as

$$\beta \propto \gamma^k, \quad k < 0.8, \quad (1)$$

and a constant phase advance μ per FODO cell, the maximum tolerable uncorrelated quadrupole jitter rms amplitudes σ_q in the main linac, corresponding

E/GeV	500	1000
$\bar{\beta}_{v,\text{end}}/\text{m}$	165	233
$\bar{\beta}_{h,\text{end}}/\text{m}$	165	233
$\gamma\epsilon_{v,\text{end}}/\text{m}$	$2.5 \cdot 10^{-7}$	$0.5 \cdot 10^{-7}$
$\gamma\epsilon_{h,\text{end}}/\text{m}$	$5 \cdot 10^{-6}$	$5 \cdot 10^{-6}$
k	0.5	0.5
μ	$\pi/2$	$\pi/2$
$2L/\text{km}$	29.4	29.4

Table 1: Main parameters of the S-Band Linear Collider SBLC [3].

to 3% luminosity degradation, can be expressed as [1, 2]

$$\begin{aligned} \sigma_q &= 0.25 \sqrt{\frac{\epsilon_{\text{end}} \bar{\beta}_{\text{end}}}{N_q} \cos \frac{\mu}{2}} \\ &= 0.25 \bar{\beta}_{\text{end}} \sqrt{\frac{\epsilon_{\text{end}} (1-k) \sin \mu}{2L} \cos \frac{\mu}{2}}, \quad (2) \end{aligned}$$

where ϵ_{end} , $\bar{\beta}_{\text{end}}$, N_q denote the geometric emittance at the end of the main linac, the average β -function of the last FODO cell, and the total number of quadrupoles in the linac. L is the total length of the main linac.

Using the parameters given in Table 1, the required tolerances in the case of the S-Band Linear Collider SBLC are therefore 85 nm for a c.m. energy of 500 GeV and 43 nm in the case of the 1 TeV version.

As a comparison with measurements in the HERA tunnel shows, these tolerances are exceeded at frequencies higher than those pulse-to-pulse beam based orbit correction schemes are capable of compensating, see Figure 1. Therefore beam-independent methods are required to keep beam orbit jitter within tolerable limits.

This article describes two different approaches of vibration compensation, namely mechanical feedback and electronic feedforward of measured ground motion signals. It should be emphasized here that

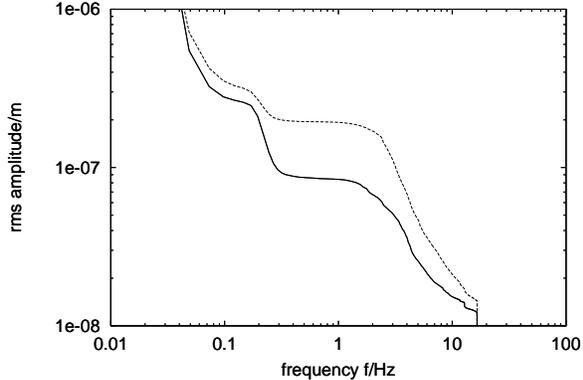


Figure 1: Measured vertical rms ground motion amplitudes in the HERA tunnel, during the night (dashed line) and during the rush hour (solid curve) [4].

these studies are aimed at the main linac with its large number of identical optical elements, which requires a solution at low cost. In the final focus with its comparably small number of elements but even tighter mechanical tolerances [5, 6], more sophisticated and expensive methods may be applied.

2 Mechanical Feedback

For compensation purposes, the ground motion spectrum may be divided into three frequency domains. At very low frequencies, beam-based orbit correction methods are applicable on a pulse-to-pulse base. As the experience with the SLC has shown, these methods are effective at frequencies below some $f_{\text{rep}}/25$, where f_{rep} is the repetition rate of the Linear Collider [7].

At high frequencies, rms vibration amplitudes tend to be rather small due to the approximate $1/f^4$ dependency of ground motion spectra, and are therefore negligible.

If rms vibration amplitudes for frequencies above $f_{\text{rep}}/25$ exceed the required vibration tolerances, some means of vibration damping is therefore required.

The simplest passive damping system utilizes some kind of “spring” to support the magnet. Together with the magnet mass m , the stiffness D of this

spring results in a resonance frequency f_r , and hence the transfer function of this passive vibration absorber can be expressed as

$$H_r(s) = \frac{\omega_r^2}{s^2 + 2\delta\omega_r s + \omega_r^2}. \quad (3)$$

Here δ denotes the damping constant, while $s = i\omega = i \cdot 2\pi f$ is the Laplace variable, and $\omega_r = 2\pi f_r$. As Equation 3 shows, high frequency ground motion amplitudes well above the mechanical resonance frequency f_r are suppressed by a factor $1/\omega_r^2$. To apply such a scheme in the case of the S-Band Linear Collider SBLC with its 50 Hz repetition frequency, it would be necessary to design a passive damping support with a resonance frequency well below $50/25 \text{ Hz} = 2 \text{ Hz}$. To achieve a mechanical resonance frequency of $f_{r\text{mr}} = 1 \text{ Hz}$ for a magnet mass of 100 kg, a spring constant $D = 4000 \text{ N m}^{-1}$ is required.

Though such a scheme would be capable of attenuating ground motion amplitudes to the required values due to its small transmissibility, it would nevertheless be extremely sensitive to forces acting on the magnet directly, like cooling water pressure fluctuations or even air flow due to air conditioning. For example, a static force of just $4 \cdot 10^{-3} \text{ N}$ would result in a static magnet displacement of $1 \mu\text{m}$. As an analysis in the frequency domain shows, the resulting amplitude $A(s)$ per unit force $F(s)$ as function of the Laplace variable s can be expressed as

$$\begin{aligned} A(s) &= \frac{\omega_r^2/D}{s^2 + 2\delta\omega_r s + \omega_r^2} F(s) \\ &= H_r(s) \frac{F(s)}{D} \\ &= H_r(s) \frac{F(s)}{\omega_r^2 m}. \end{aligned} \quad (4)$$

These considerations led to the development of an active stabilization system with a vibration sensor on top of each magnet and some means of actuator to move the magnet in order to keep its center at rest, as schematically shown in Figure 2.

At this point, again a decision had to be made about the mechanical resonance frequency of the active support. In the case of a soft support with a mechanical resonance frequency of some $f_r = 10 \text{ Hz}$, higher frequency ground motion amplitudes would be attenuated already just due to the passive mechanical behavior of that device. In the low

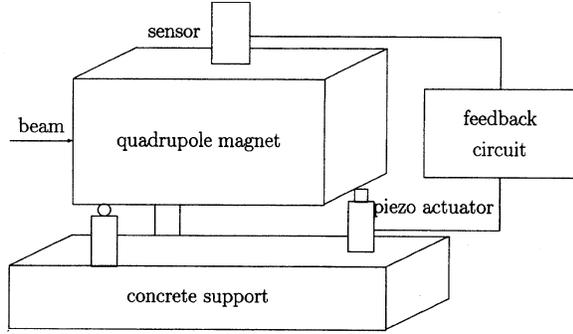


Figure 2: Schematical drawing of the active stabilization concept.

frequency range, active compensation has to be applied, for example by means of an electrodynamic transducer.

The transfer function describing the coupling of ground motion to magnet motion of this soft active support can be expressed in terms of the passive mechanical transfer function

$$H_r(s) = \frac{\omega_r^2}{s^2 + 2\delta\omega_r s + \omega_r^2} \quad (5)$$

and the transfer function $H_s(s)$ of the sensor, together with an arbitrary feedback algorithm $F(s)$, as

$$H_g(s) = \frac{H_r(s)}{1 + F(s)H_r(s)H_s(s)}. \quad (6)$$

As in the case of the pure passive support, the effect of forces acting on the magnet directly has to be taken into account. This can be expressed by the transfer function

$$H_m(s) = \frac{1/D}{1 + F(s)H_r(s)H_s(s)}. \quad (7)$$

All vibration sensors, like geophones, seismometers, piezoelectric accelerometers, etc., have a certain lower cutoff frequency f_s , hence

$$\lim_{s \rightarrow 0} H_s(s) = 0. \quad (8)$$

For high frequencies well above its mechanical resonance frequency f_r , the transfer function $H_r(s)$ of the mechanical support vanishes,

$$\lim_{s \rightarrow \infty} H_r(s) = 0. \quad (9)$$



Figure 3: Photo of the geophone plus corresponding preamplifier used for active stabilization.

Therefore such a soft active support responds to forces acting on the magnet directly like a pure passive support with the same resonance frequency f_r and spring constant D [8],

$$\lim_{s \rightarrow 0} H_m(s) = \lim_{s \rightarrow \infty} H_m(s) = \frac{1}{D}. \quad (10)$$

Thus, the resonance frequency and correspondingly the spring constant (or stiffness) of the support were chosen as high as possible. The magnet is supported and moved by a piezoelectric actuator.

To detect magnet motion in the frequency range above 2 Hz, inexpensive electrodynamic geophones are used, Figure 3. While the mechanical resonance frequency of these probes is 4.5 Hz, the transfer function is modified by several analog filters such that it is flat within the range from 2 Hz to 400 Hz. Obviously, this in turn results in a large phase change over this frequency range, see Figure 4. As measurements of the internal noise level of these devices plus their corresponding preamplifiers showed, these sensors are capable of measuring vibration amplitudes as small as about 1 nm at 2 Hz.

The entire active stabilization system is shown in Figure 5. The triangular cross section of the concrete support ensures the transfer of horizontal ground motion without additional amplification. The pre-amplified and filtered output signals of the geophone on top of the magnet are sent to a PC with 16 bit A/D board, where the signals are

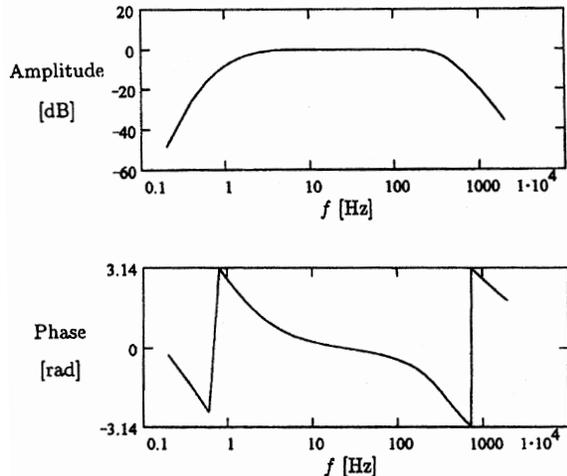


Figure 4: Transfer function of the geophone plus pre-amplifier.

digitized and processed in order to provide the required signals to be applied to the piezoelectric actuator. The system was set-up in an experimental hall at the DESY synchrotrons. This is considered a noisy environment with two synchrotrons, several transformers, and other equipment operated nearby. Therefore the ground motion conditions should be comparable to those to be expected in a real Linear Collider. The magnet coil was connected to the cooling water system, which is considered an additional vibration source. All measurements presented in this paper were taken at a cooling water flow of 220 l/h, which is well above the design value of 120 l/h.

To accurately investigate the performance of the active stabilization system, a second, identical sensor was placed on the ground just underneath the magnet support. The output signals of both the feedback sensor as well as this second one were simultaneously sampled at 400 Hz. After application of a Hanning window, sets of 1024 data points each were Fourier-transformed in order to determine the power spectra Φ_{magnet} and Φ_{ground} of both the magnet vibration and the ground motion. Figure 6 depicts an example of the power spectra obtained on top of the magnet and on the ground underneath when the feedback system is operating. Using these two spectra, the transfer function of the entire sys-

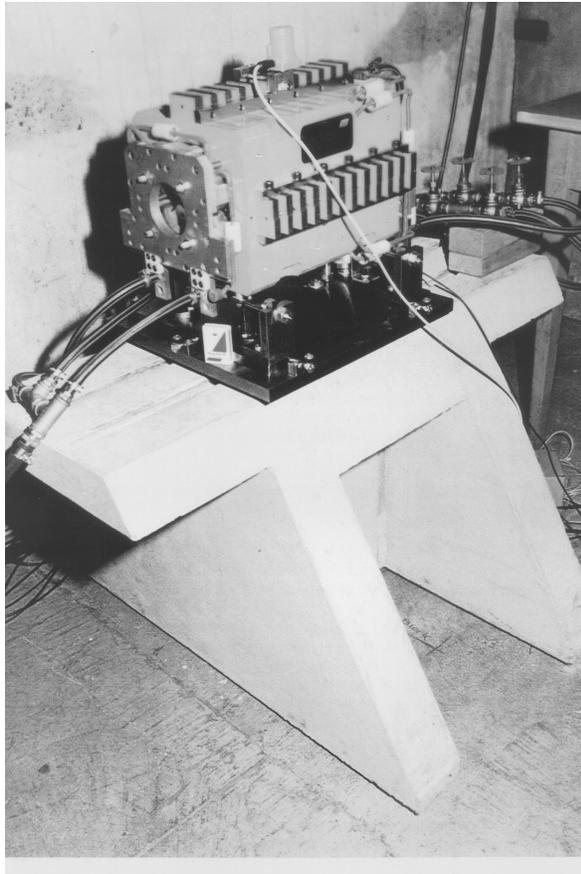


Figure 5: Photo of the experimental setup.

tem can be calculated as

$$H_{\text{total}}(f) = \sqrt{\frac{\Phi_{\text{magnet}}(f)}{\Phi_{\text{ground}}(f)}}. \quad (11)$$

As Figure 7 shows, the measured gain agrees very nicely with the theoretical transfer function.

The performance and maximum achievable attenuation factor is limited at the high frequency end by the first spurious resonance frequency of the piezo actuator, at $f_{\text{res}} = 143$ Hz. The origin of this resonance could not be detected. A calculation from the stiffness $D = 1300$ N/ μm of the piezo and the magnet mass $m = 100$ kg results in a resonance frequency of $f_{\text{th}} = 575$ Hz, which is far beyond the measured value. On the other hand, a static measurement of the stiffness is consistent with the specified value [9].

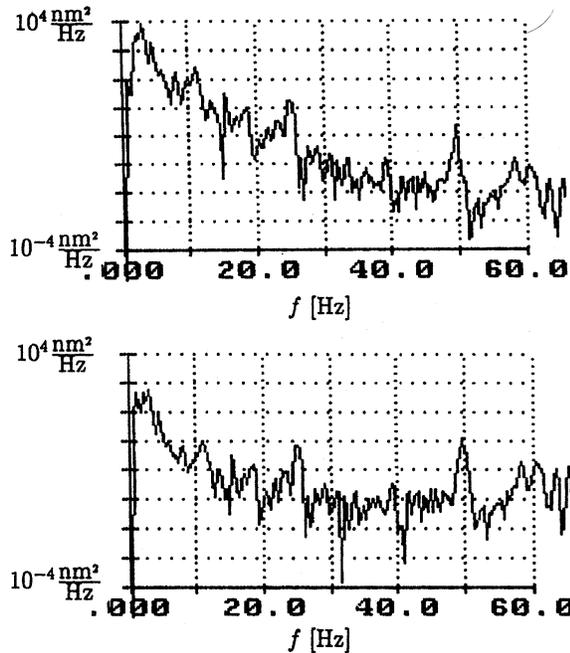


Figure 6: Simultaneously measured power spectra of ground motion (top) and magnet motion (bottom graph).

The low frequency limit is given by the lower cut-off frequency of the sensor. This could be improved by application of a broadband seismometer, but has not been tried yet.

The measured rms amplitudes for frequencies above a lower limit f_0 , calculated from the respective ground motion power spectra according to

$$\sigma(f > f_0) = \sqrt{\int_{f_0}^{\infty} \Phi(f) df} \quad (12)$$

are depicted in Figure 8. While the rms ground vibration amplitude for frequencies above 2 Hz is roughly 100 nm and therefore above the required tolerances, this value is damped down by the active stabilization system to some 25 nm, which is well below the required tolerances even for the 1 TeV case.

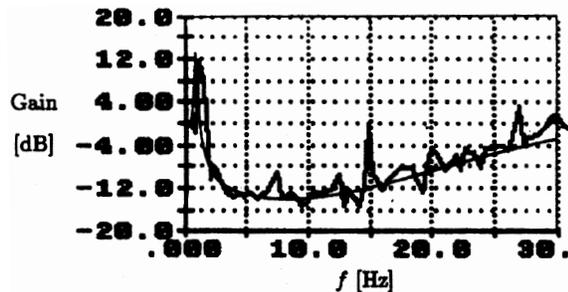


Figure 7: Feedback gain as calculated from the measured spectra of ground motion and magnet motion.

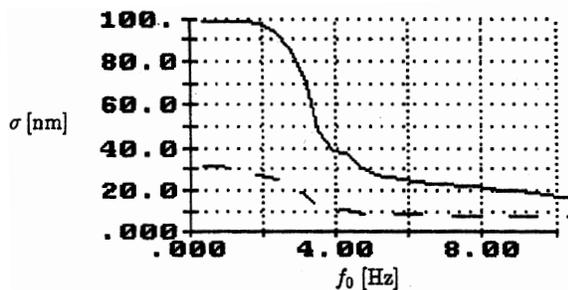


Figure 8: Measured rms vibration amplitudes of ground motion (solid line) and magnet motion (dashed curve), for frequencies $f > f_0$, plotted as a function of the lower frequency limit f_0 .

3 Electronic Feedforward

Mechanical devices usually tend to be rather expensive due to the long-lasting manufacturing process. Therefore one might think of an alternative method to compensate the effect of magnet vibration on the beam. Instead of using the measured magnet vibration signals in a closed feedback loop in order to mechanically stabilize the magnet motion, these signals could also be applied to dipole correction coils, thus compensating the beam orbit motion caused by the vibrating quadrupole.

In such a scheme, the phase of the transfer function must be close to zero (or 180°) over the entire frequency range of the system, while its amplitude must be constant. Obviously, the simple geophones used for the mechanical feedback system do not ful-

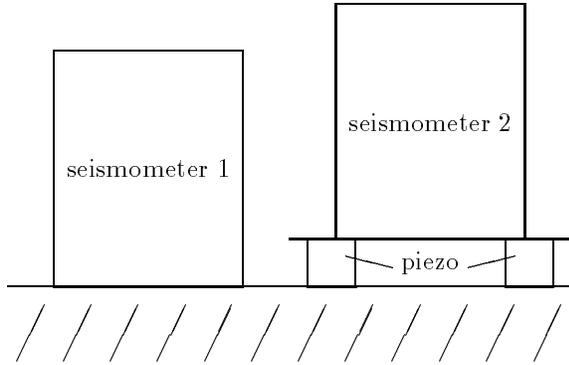


Figure 9: Mechanical test setup. The output signal of seismometer 1 is applied to several piezo actuators keeping a second seismometer at rest.

fill these conditions (see Figure 4) and are therefore not applicable in such an alternative scheme. Generally, broadband seismometers like the Guralp CMG-40 have the required properties, but are also more expensive than geophones.

The feasibility of the feedforward scheme can only be tested in an analog mechanical setup, as schematically shown in Figure 9. The ground motion is detected by a broadband seismometer, and the output signal of this instrument is applied to piezo actuators supporting a second seismometer, which measures the remaining vibration amplitudes.

4 Conclusion

The feasibility of active mechanical stabilization down to some 25 nm for frequencies above 2 Hz has been already successfully demonstrated. An alternative electronic feedforward system requires more sophisticated sensors, namely broadband seismometers with a flat transfer function, which are usually more expensive than passive geophones. A (mechanical) feasibility test of this scheme still remains to be done.

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