MODAL ANALYSIS AND MEASUREMENT OF WATER COOLING INDUCED VIBRATIONS ON A CLIC MAIN BEAM QUADRUPOLE PROTOTYPE*

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Abstract

To reach the Compact Linear Collider (CLIC) design luminosity, the mechanical jitter of the CLIC main beam quadrupoles should be smaller than 1.5 nm integrated root mean square (r.m.s.) displacement above 1 Hz. A stiff stabilization and nano-positioning system is being developed but the design and effectiveness of such a system will greatly depend on the stiffness of the quadrupole magnet which should be as high as possible. Modal vibration measurements were therefore performed on a first assembled prototype magnet to evaluate the different mechanical modes and their frequencies. The results were then compared with a Finite Element (FE) model. The vibrations induced by water-cooling without stabilization were measured with different flow rates. This paper describes and analyzes the measurement results.

INTRODUCTION

In CLIC, currently under study [1], about 4000 Main Beam Quadrupoles (MBQ) with a magnetic field gradient of 200 T/m are used to maintain ultra low beam emittance and size along the two linear accelerators. To reach the design luminosity, a vertical beam size of 1 nm is required at the interaction point (40 nm in the horizontal plane). To preserve the beam emittance, a beam based orbit feedback system based on Beam Position Monitors (BPM) and corrector dipoles will be combined with an active vibration stabilization system under each MBQ. Beam dynamics simulations showed that the movements of the magnetic axis should be limited to the nanometre level. As an indicative value, the integrated r.m.s. absolute displacement [2] should not exceed 1.5 nm in the vertical direction and 5 nm in the lateral direction at 1 Hz. To reach such a level the MBQs need to be isolated from ground motion comprising seismic low frequency motion and technical vibrations [3]. In addition, direct vibration forces acting on the magnet are induced by water cooling, ventilation, or transmitted through inter-connections with other components. The stabilization system was designed very stiff to be robust against such direct forces.

A magnet mock-up without water-cooling was stabilized to the required level [2] based on ground motion typical for particle accelerators. The first resonance frequency of the quadrupole itself or of the quadrupole on its support was identified as a limiting factor for the controller stability [4] and should be as high as possible, preferably well above 100 Hz.

A first estimation of the water cooling influence, expected to be the most important vibration source, was measured on the first MBQ prototype and is presented in this paper.

MAGNET DESCRIPTION

There are four different types of MBQs with identical cross-sections and with a length between 332 mm (Type 1; 87 kg) and 1827 mm (Type 4; 424 kg). This paper discusses the analysis and measurements of the longest type 4. Each magnet is composed of four identical steel quadrants which are bolted together (76 M8 bolts, 15 Nm). The aperture of the magnet is 10 mm in diameter. The size of the cross-section is 232 by 232 mm. The coils are wound copper conductors with internal water flow, impregnated with epoxy resin and the total mass of the coil is about 88 kg. Shims and insulating material are inserted between each yoke and coil. The coil is pre-stressed by the quadrants being bolted together. The combined coil modulus was not measured and the coil pre-stress is therefore not known.

MODAL ANALYSIS

FE models of type 4 MBQ (Fig. 1) were developed [5] [6] to determine the natural frequencies and modal deformation of the assembled magnet with free boundary conditions and to verify the influence of the coil and magnet support stiffness on the modes.

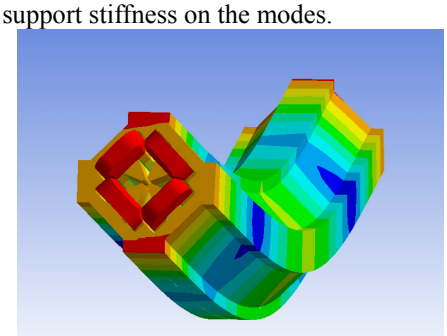

Figure 1: FE model of a type 4 MBQ including the coil [6] with a first mode at 281 Hz for free boundary conditions (measured at 264 Hz).

To validate the FE models, an experimental modal analysis was performed on the first type 4 MBQ prototype [7]. The magnet was tested in free support conditions by hanging it vertically from a crane from slings on one end. An impact hammer with integrated force transducer was used to excite the magnet structure at several locations, in

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three directions. The dynamic response was measured with five tri-axial accelerometers fixed on the magnet and recorded by a signal analyser. Modal analysis software was used to calculate the transfer functions and to extract the modal parameters such as frequencies, modal shapes, and damping ratios from the measurements.

The first two (orthogonal because symmetrical) modes are the beam bending modes at 264 Hz, shown in Fig. 1. The next two beam bending modes were measured at 628 Hz and the first longitudinal torsion mode at 656 Hz (Table 1). The prototype magnet is thus sufficiently stiff and the results confirm the conclusion from the models whereby an assembly by bolts with enough bolt tension is very similar to a fully welded assembly. The small differences (~6% for modes 1 to 4) found between the measured and calculated frequencies can be explained by differences in mass, Young modulus, and the modelling of the coil. The coil modulus (1 GPa) to be applied in the FE model to obtain the measured frequencies indicates that the copper coil does not participate in the stiffness of the magnet. With this low coil modulus, the torsion mode in FE model 2 becomes 691 Hz, closer to what was measured.

Table 1: Comparison between calculated and measured modes with free boundary conditions

Mode	Freq. (Hz) Measured [7]	Damping (%) Measured [7]	Freq. (Hz) Model 1 [5]	Freq. (Hz) Model 2 [6]
1 & 2	264	1.26	279	281
3 & 4	628	3.32	676	669
5	656	3.54	684	905

Although the magnet is sufficiently stiff, the dynamic behaviour of the magnet will depend on the applied boundary conditions. Lower natural frequencies of rigid body modes can limit the stabilization performance. To allow analysis including the stabilization support before its final design and construction, simple "equivalent" supports were used in the FE model and for testing. The stiffness of the equivalent supports corresponds to the maximum vertical stiffness reachable with the stabilization support design. To reduce the stiffness, cuts are applied in the equivalent supports in the model. Such cuts can also be applied on the supports built for measuring the magnet with different support stiffness but this has not yet been done.

FE models with three equivalent supports show rigid body modes just above 100 Hz. The exact measured frequency values and modes are at this stage less important since the structure is over-constrained. The static vertical sag of the magnet on two supports at the Airy points is of the order of 1 μm and the mounting tolerances of the supports will consequently pre-stress the magnet and change the frequencies. More important is the identification of a longitudinal, a lateral, and a yaw rigid body mode with frequencies between 100 and 150 Hz, giving important input to the design of the stabilization support and the alignment system.

The defined MBQ vibration stability is the stability of the magnetic axis. The vibrations of the MBQ can only be measured on the outside of the quadrupole. In the FE models there were, however, no modes of the single pole tip (not measurable on the outside) below 1 kHz.

WATER COOLING

The measurement of water-cooling induced vibrations was made with seismometers (Guralp GMC 6T) placed on the magnet. The magnet was positioned horizontally on two "equivalent" supports at the Airy points and one support in the middle of the magnet. To avoid pre-stress which might affect the frequencies, each support contacts the magnet freely (no screws) in the middle of the inclined sides of the two lower quadrants. For the first series of measurements the magnet was placed on its supports on the marble stone of an optical table. From analysis of the measurements, several resonances were observed at low frequencies (< 50 Hz) raising the r.m.s. displacements between floor and magnet to levels far above the requirements. The white noise vibrations induced by water-cooling were further amplified at these low frequency resonances. This increase of more than 10 nm is, however, due to the optical table and hence not representative (although very instructive).

Therefore, a second series of measurements was made with the magnet placed on equivalent supports at the same positions, directly on the floor (Fig. 2).

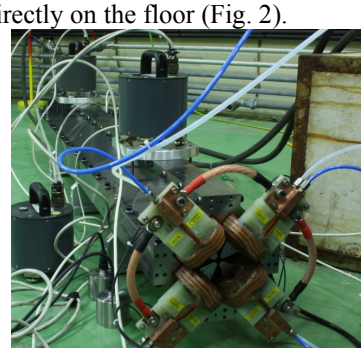

Figure 2: Water cooling measurements with the magnet on equivalent supports on the floor

Seismometers measured the vibrations on the magnet and just next to it on the floor, at different longitudinal positions. Each test was made during 500 s and the Power Spectral Density (PSD) was calculated for 1 l/min, 2 l/min, and without flow. The nominal flow of 0.7 l/min (turbulent regime) could neither be set nor measured with the valve and flow gauge available. Displacements are obtained by integrating the velocity measured by the seismometers. The integrated r.m.s. displacement was calculated from the PSD. More information on the analysis techniques is given in Ref. [8].

Determining the small difference of vibrations due to water flow is not straightforward. First, one can compare the r.m.s. results [9] with and without water flow (Fig. 3). Differences are in this case very small and only visible at

higher frequencies. Measurements with and without flow are, however, not measured at the same time and include ground motion. As ground motion can change from one measurement to the next, direct comparison is not correct as shown, for example, for the vertical direction in Fig. 3 where the integrated r.m.s. below 4 Hz seems to decrease with water flow. The change of integrated r.m.s. displacement of the ground motion was small between the different measurements, but so was the change caused by water-cooling on the magnet.

A longitudinal mode at 32 Hz has not yet been explained. No significant differences were measured at different positions on the magnet. No significant increase was measured for a flow rate of 2 l/min.

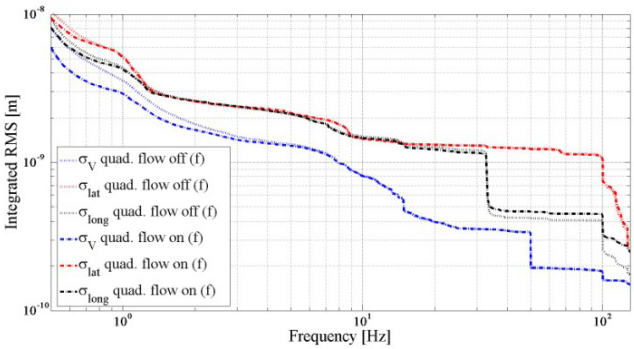

Figure 3: The integrated r.m.s. displacement of the magnet without and with water cooling with a flow of 1 l/min for a stiff magnet support

From the measurements without water flow (off), the transfer function $\mathbf{T}^{off}_{wx}$ of vibrations between the ground (w) and the magnet (x) can be calculated for each position and each direction. The increase of vibrations Φ_δ caused by the water flow can then be calculated from:

$$\Phi_\delta = \Phi_x^{ON} - \left|T_{wx}^{OFF}\right|^2 \Phi_w^{ON}$$

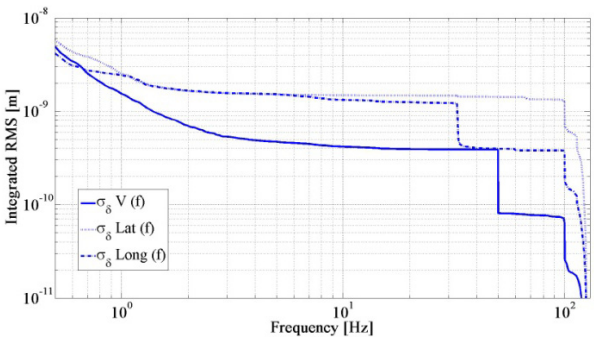

Figure 4: Calculated r.m.s. increase due to water flow

Although this method is more correct it is also less precise because measurement uncertainties and noise are added up. The integrated r.m.s. calculated from Φ_δ (Fig. 4) is about 2 nm at 1 Hz, well above what is directly measured (Fig. 3). Nevertheless, it gives an upper limit for the increase of integrated r.m.s. displacement at 1 Hz caused by water flow, for a magnet on stiff supports and without stabilization.

CONCLUSIONS

To be robust against vibration forces acting directly on the magnet and to avoid low-frequency resonances that would limit the stabilization performance, the CLIC MBQs and its supports should be as stiff as possible. An experimental and FE modal analysis confirms that the magnet assembled with bolts is sufficiently stiff with a first bending mode at 264 Hz for free boundary conditions. Magnet suspension modes on the magnet support occur, however, at lower frequencies. For supports with stiffness equivalent to the stabilization system under development, longitudinal and lateral modes were calculated with frequencies between 100 and 150 Hz.

The measurements on the magnet with nominal water flow stressed the importance of a stiff support without low-frequency rigid body modes. For the highest reachable stiffness with the current stabilization system design, the increase of integrated r.m.s. displacement caused by water-cooling, without stabilization, is very small and difficult to measure. As a very conservative upper limit, an r.m.s. of 2 nm was calculated for a close to nominal water flow and must be removed by the stabilization system.

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