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IMPROVING HIGH PRECISION CAM MOVER'S STIFFNESS

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Abstract

Pre-alignment is a key challenge of the Compact Linear Collider (CLIC) study. The requirement for CLIC main beam quadrupole (MBQ) alignment is positioning to within 1 µm from target in 5 degrees of freedom (DOF) with ± 3 mm travel. After motion, the position should be kept passively while the system's fundamental frequency is above 100 Hz. Cam movers are considered for the task. Traditionally they are used for the alignment of heavier magnets with lower accuracy and stiffness requirement. This paper presents a new CLIC prototype cam mover with design emphasis on the fundamental frequency. A finite element method (FEM) model predicts the mode shapes and eigenfrequencies of the system and can be used for further improving the design. Experimental modal analysis (EMA) of the prototype shows that the prototype's fundamental frequency is at 44 Hz. It also validates the FEM model.

INTRODUCTION

Cam movers are widely used in particle accelerator alignment. They can position loads up to several tons with the precision of tens of µm or below. In most cases this is enough, but at the Taiwan Photon Source, also dynamic stiffness was important. They are using a six-axis cam mover, with axis movement resolution of 1 µm. After positioning, they use separate clamping devices to stiffen the structure. They reached a fundamental frequency of 24 Hz without and 30 Hz with the clamping. [1]

In the Compact Linear Collider (CLIC) study, the design considerations are somewhat different. Cam movers with five degrees of freedom (DOF) are planned to be used in active pre-alignment of the main beam quadrupoles (MBQ). There are four types of MBQs to align, masses of which vary between 200 kg and 800 kg. The CLIC positioning requirement is to have both ends of the MBQ along the beam line within 1 µm in X- and Y-directions. In addition, the beam line roll should be below 100 µrad and translation along the beam should be blocked (not controlled). [2]

After positioning, the cam mover should provide stiff support so that the fundamental frequency is above 100 Hz. In this study, the design considerations that bring cam movers closer to the CLIC stiffness requirement while preserving the positioning accuracy are discussed. Emphasis is on the lightest CLIC MBQ because it poses challenging space requirements to the cam mover. The dimensions of the lightest MBQ together with its accessories are 460 mm x 377 mm x 500 mm (WxLxH).

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CAM MOVER DESIGN

Stiffness was taken into consideration in all design aspects of the new CLIC cam mover prototype. The left side of Fig. 1 shows a previous CLIC prototype version where only positioning resolution and space restrictions were taken into account. Each of the five axes has its own body which also includes the gearbox and stepper motor. The axes are bolted together on a support plate.

The middle of Fig. 1 shows the new prototype cam mover. Extra space allowance was found in the vertical direction and this was taken advantage of. At the same time, it was decided that all five axes should be in one body. The polymer concrete body is manufactured by Schneeberger Mineralgusstechnik s.r.o. The technology is expensive when only a prototype is manufactured due to the design and manufacturing costs of the mould. But in CLIC, hundreds if not thousands of cam movers would be needed. In serial production, polymer concrete is a cost-effective choice. The actual common axis housing is made of EN AW-7075 (T6) aluminium and it is bolted and glued to the polymer concrete body.

Figure 2 shows the difference between a single axis of the old CLIC design (left) and the new one (right). The camshaft of the new prototype is thicker than in the old one and it is made of hardened 42CrMo4 steel. The camshaft is attached to the housing with an SKF NU 2305 ECJ radial roller bearing (radial support) in one end and a back-to-back configuration of SKF 31305 J2 tapered roller bearings (radial and axial support) in the other end.

The right side of Fig. 1 shows a CLIC MBQ together with its accessories. It sits on the cam mover so that the five interface planes — four inclined and one horizontal — are in contact with the eccentric disc parts of the five camshafts. This means that five contact regions define the MBQ position and, to a large extent, the system's stiffness. Possibly the most important difference between the old axis design and the new one is that the bearing around the eccentric part of the camshaft is removed. The bearing around the eccentre makes positioning smooth because there is no sliding but rolling between the two surfaces in contact. However, the bearing is detrimental to the system's stiffness.

It can be seen in Fig. 2 that the new camshaft is crowned. This is in order to make sure that the location of the contact region is controlled. The crown radius was originally set to 500 mm because big radius means large contact area and thus high contact stiffness. It was later changed to 130 mm which is easier to manufacture.

Each axis is driven by a stepper motor through a worm gear and a belt drive. The worm gear's gear ratio is 70 which makes it self-locking. This is a useful feature as it both keeps the cam mover's position and provides the axis with torsional support.

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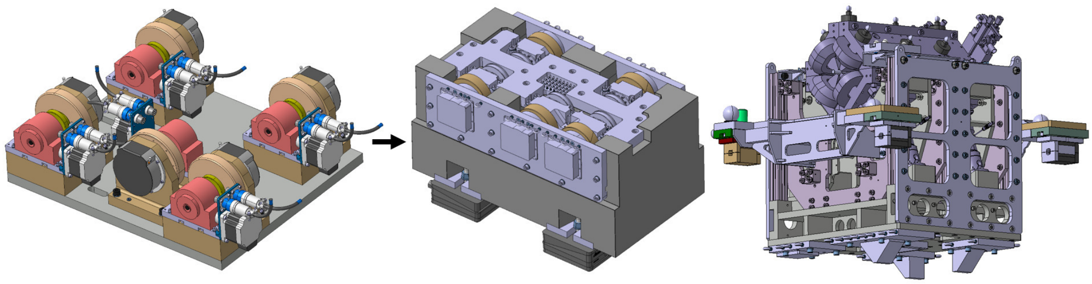

Figure 1: CLIC cam mover prototypes. The previous prototype (left) had five separate actuators as axes whereas the new prototype (middle) has all axes integrated in one body and housing. CLIC MBQ including the planes that interface it with the cam mover is shown in the right.

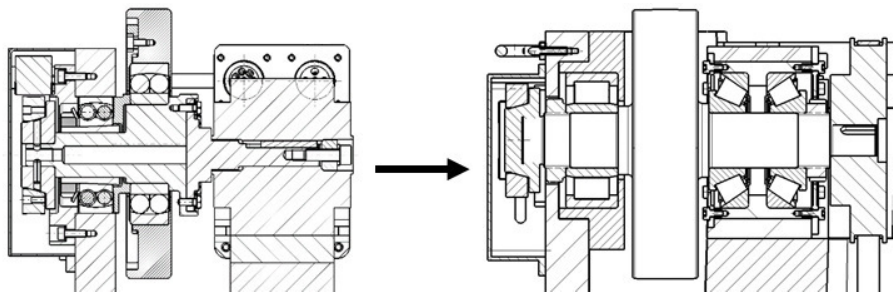

Figure 2: Section view of an axis of the previous CLIC cam mover prototype (left) compared to the new prototype (right).

FINITE ELEMENT METHOD MODEL

A finite element method (FEM) model of the whole system containing the cam mover and the MBQ was created (Fig. 3) using Ansys Workbench software. The contact areas between the camshafts and the interface planes are modelled as frictional contacts that are brought together before the simulation. Because this is an important area considering the natural frequencies, a fine mesh is created using a pinball region. Coefficient of friction is estimated to be 0.15. Two greased steel surfaces are in contact.

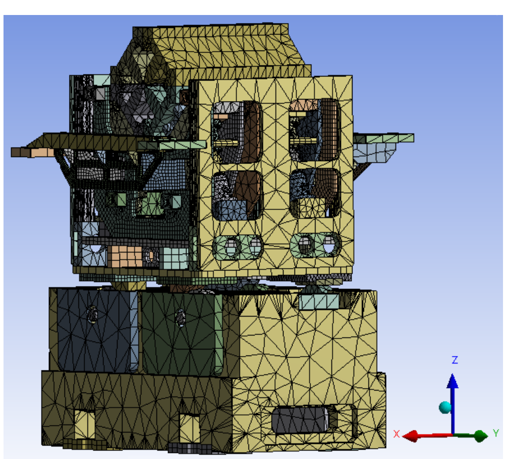

Figure 3: FEM model of the cam mover and the MBQ.

All bearings are modelled with bushings between the housing and the camshafts. The stiffness matrices have been calculated based on formulas that have been combined from literature sources [3, 4]. For the radial roller bearings, preload is designed through an interference fit whereas the tapered roller bearings are mounted on the axis with the aim for zero clearance. The design values were used in calculation of the stiffness matrices, as well as radial forces that the weight of the MBQ causes to the camshafts.

A static structural analysis is done first so that the effects of gravity and preload are taken into account. The results are given as input to the modal analysis. The system's eigenfrequencies grow when preload is added between the MBQ and the cam mover.

The eigenfrequencies of the system's first three modes are listed in the first column of Table 1 when there is no preload and in the first column of Table 2 when there is a preload of 730 N. In each cell, the mode number is followed by the eigenfrequency.

The MBQ tilts around the Y-axis in mode 1, X-axis in mode 2 and Z-axis in mode 3. In mode 3, there is also torsion in the MBQ frame. The cam mover body is almost stationary while the movement is mostly between the cam interface planes and the camshafts. The camshafts also move inside the bearings.

EXPERIMENTAL MODAL ANALYSIS

An experimental modal analysis (EMA) of the prototype system consisting of the cam mover and the MBQ was performed by inspire AG. Figure 4 shows the prototype system with accelerometers attached to it for the EMA measurements.

The EMA was performed on the system both without and with preload between the MBQ and the cam mover.

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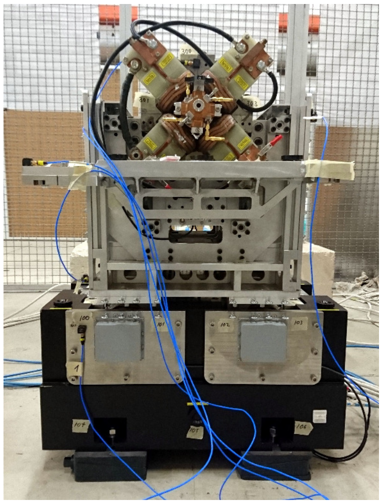

Figure 4: Cam mover and MBQ prototype ready for experimental modal analysis.

Table 1: Cam Mover and MBQ Prototype System FEM and EMA Comparison without Spring Preload

FEM	EMA	Modified FEM
1 - 43.5 Hz	1 - 43.9 Hz	1 - 36.4 Hz
-	2 - 44.5 Hz	2 - 46.0 Hz
2 - 58.9 Hz	3 - 52.7 Hz	3 - 52.3 Hz
-	4 - 71.5 Hz	-
-	5 - 95.7 Hz	-
3 - 111 Hz	6 - 99.0 Hz	4 - 109 Hz

Table 2: Cam Mover and MBQ Prototype System FEM and EMA Comparison with Spring Preload

FEM	EMA	Modified FEM
1 - 49.1 Hz	1 - 44.5 Hz	1 - 39.9 Hz
-	2 - 44.9 Hz	2 - 46.3 Hz
2 - 67.1 Hz	3 - 53.3 Hz	3 - 59.2 Hz
-	4 - 71.7 Hz	-
-	5 - 95.9 Hz	-
3 - 121 Hz	6 - 99.1 Hz	4 - 118 Hz

Preload was introduced using several springs in parallel and nominally the combined spring force was 730 N. The second column of Table 1 shows the results without and the second column of Table 2 with the spring preload.

The EMA modes 1, 3 and 6 are similar to the FEM modes 1-3 respectively, with a difference that in EMA, also the cam mover body moves. In mode 2, the whole system is displaced in Z-direction. In mode 4, the MBQ frame tilts around X relative to the cam body. Mode 5 consists of torsion of the frame and rotation of the magnet around Z while there is displacement of the MBQ frame relative to the cam body in Z.

DISCUSSION

The FEM model predicted three out of the six first EMA modes. In addition, the effect of preload is not visible in the EMA results. The FEM model is still being developed in order to explain better the EMA results.

In the current FEM model version, fixed supports between the cam mover body and foundation have been replaced with bushings. The results are visible in the third columns of Tables 1 and 2. The displacement mode in Z-direction (mode 2 of EMA and modified FEM) is likely explained by compliance in the foundation or in wedgemounts that were used to level the cam mover body. Especially since the preload does not have effect in the eigenfrequency of this mode in the modified FEM model. The EMA modes 4 and 5 and the missing effect of preload in EMA modes 1, 3 and 6 are still under investigation. Also, an effort is put to elimination of the differences between the eigenfrequencies of similar modes betweeen EMA and FEM.

The goal of 100 Hz fundamental frequency has not yet been reached. Simulations indicate that adding a blocking device in X-direction would help significantly since the first mode is tilting around the Y-axis. Adding a sixth axis would increase the fundamental frequency the most, but it would increase also the cost and complexity of the design significantly.

CONCLUSIONS

A new CLIC prototype cam mover has been developed. The emphasis in its design was on stiffness while preserving positioning accuracy of below 1 µm. Fundamental frequency of 43.9 Hz was found experimentally. After FEM model update, the first three modes correspond fairly well between FEM and EMA when there is no spring preload in the system. The test site foundation seems to have an effect on the EMA results.

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