Module & Main LINAC Studies

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With thanks to:
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Andrea Latina, Daniel Schulte

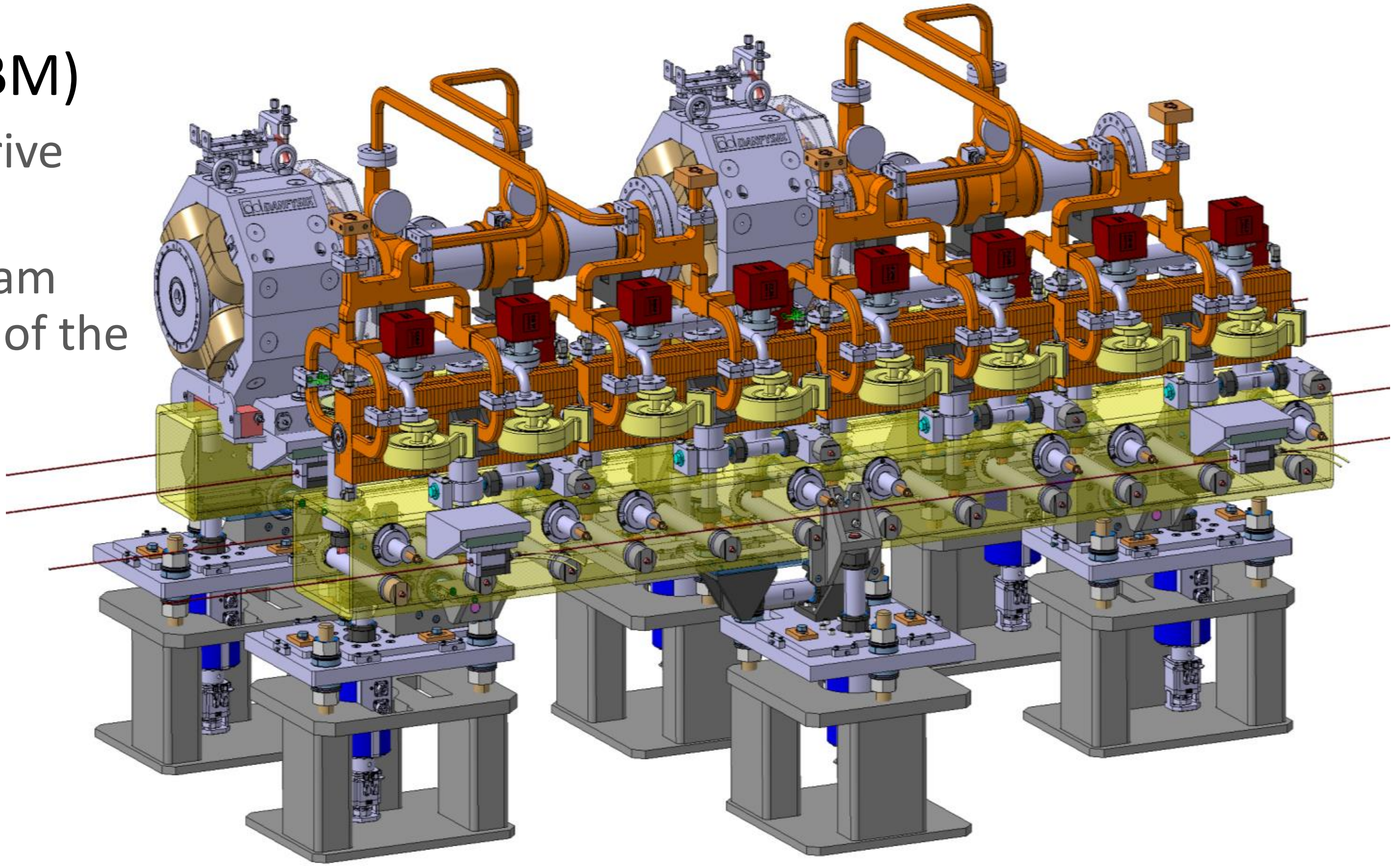
Contents

Module Design

A summary of the module systems, and an update on the current design and alignment systems.

Module Introduction

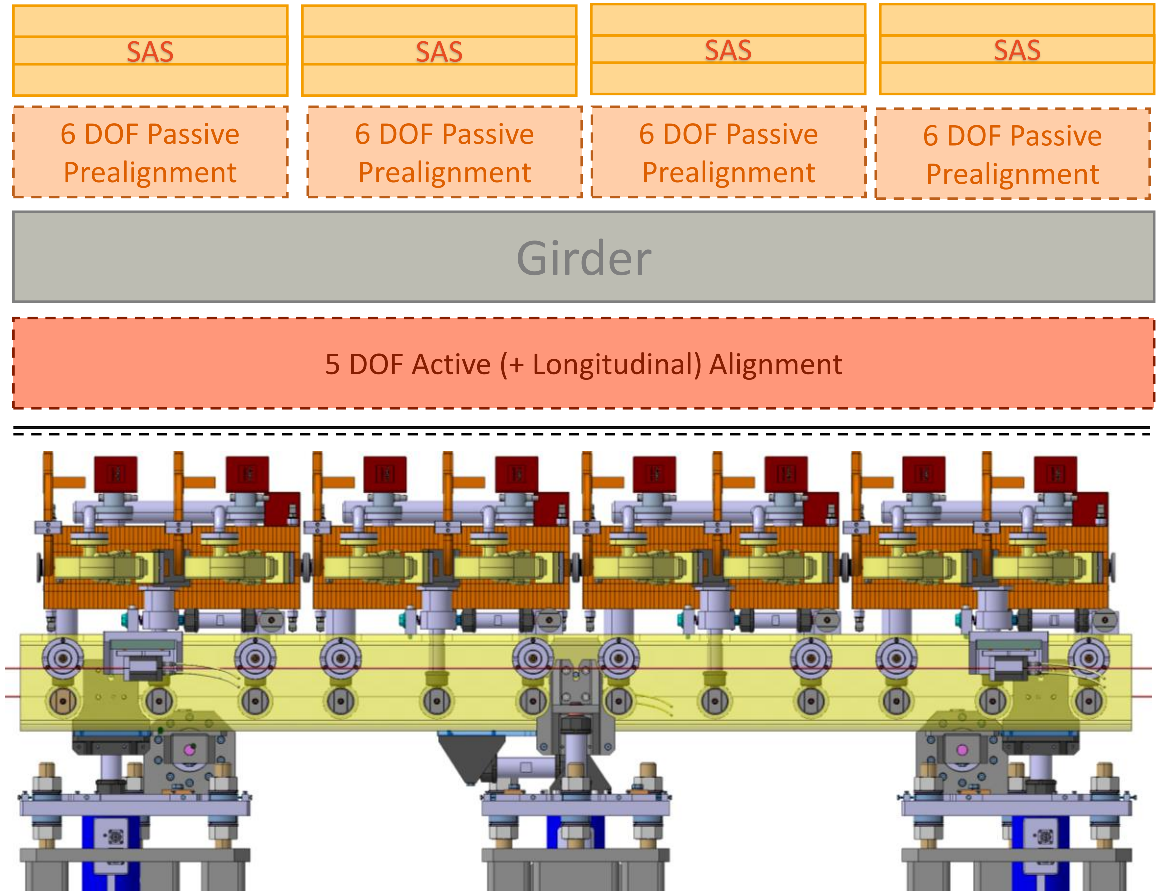
- The CLIC Two Beam Module (TBM)
  - Closely integrated assembly of Drive Beam and Main Beam sections
  - Along with the separate Main Beam Quadrupoles, forms the majority of the CLIC main LINAC
- Contains the positioning and alignment systems for the super accelerating structures
  - Passive prealignment
  - Active positioning
  - Stabilisation



Above: A CLIC T0 Two Beam Module (TBM)

Module Alignment Introduction

- The Super Acceleration Structures (SAS) are individually prealigned relative to the girder to within 10μm in all axes
- The girder is actively aligned using the same methodology & kinematics
  - Vertically and laterally adjusted automatically using linear actuators.
  - Longitudinal axis can be active or passive as required. Currently there is no proposed feedback mechanism.
  - Girder position monitored using WPS (Wire Position Sensors)
- Two-stage adjustment:
  - Removes the need for expensive precision girders e.g. SiC, Epument
  - Removes the misalignment induced due to thermal mismatch between the SAS and the supports
- Current design uses 'universal joints' for the SAS and girder positioning systems
  - Previously considered flexures and cam movers



Top: Main beam girder alignment schematic. Bottom: profile view of a main beam girder

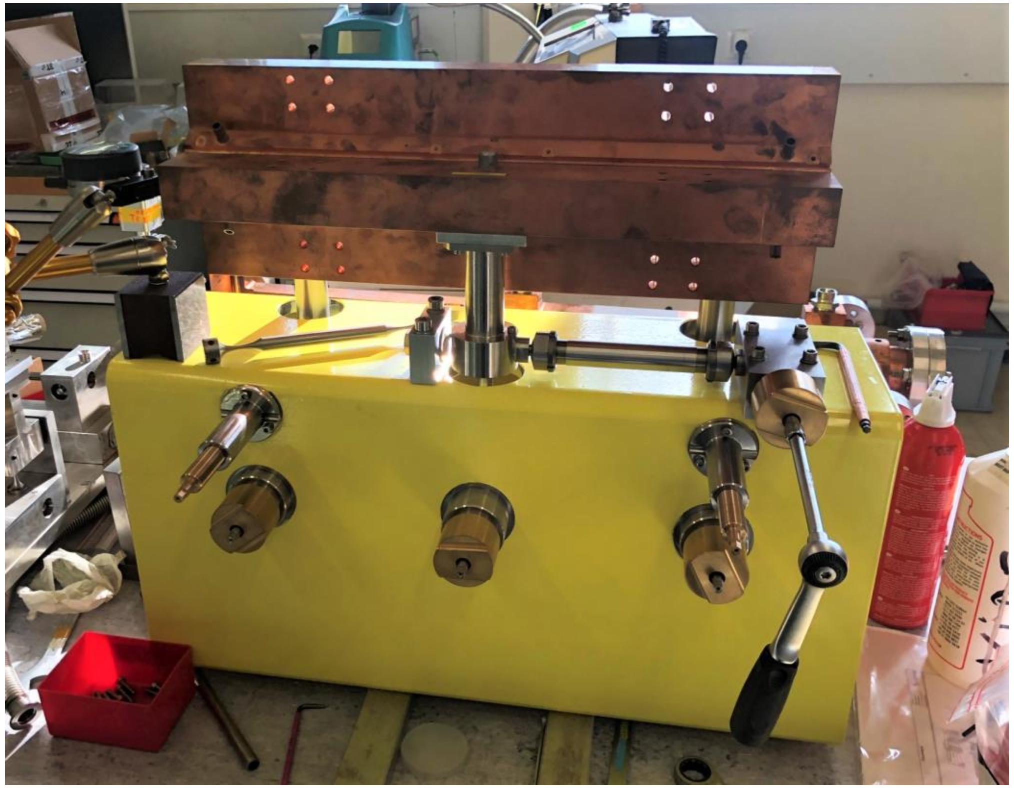
SAS Alignment Platform Prototypes

The manufacture and testing of SAS alignment platform prototypes.

Introduction to future prototype designs.

SAS Alignment Platform Prototypes

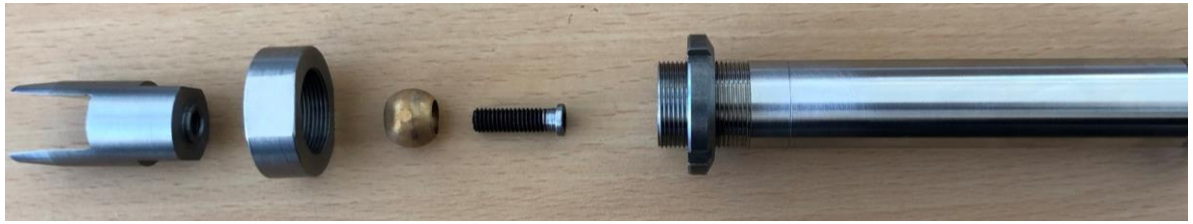
- This year we have manufactured and tested two versions of the SAS prealignment system
  - Consists of six mechanical flexures
  - This fully constrains the structure
  - Each flexure is then moved by a wedge or differential thread which provide a mechanical reduction and allow very fine precision adjustment in that direction
  - All adjustment points are on one side of the girder and intended to be compatible with a semi automatic adjustment system
- V3 prototype = 6 flexures
- V3.5 prototype = 5 flexures and one ‘universal joint’
  - Introduced to verify the universal joint and see if it impacted the operation of the system



Top: The SAS adjustment platform V3.5 during testing.

Universal Joints

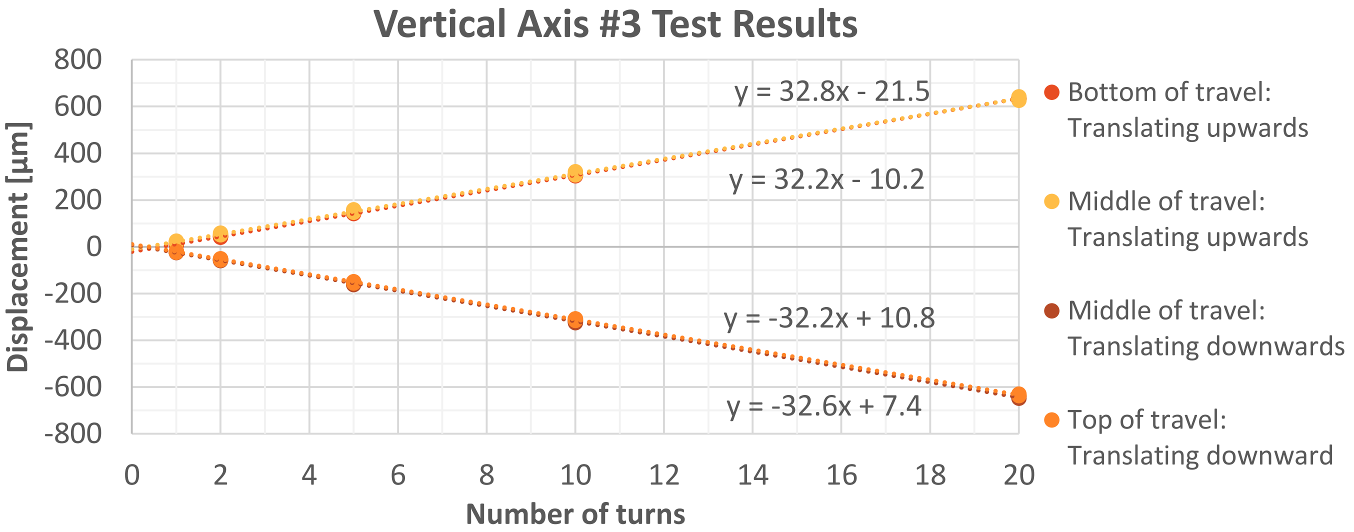
- Flexures (top right) are designed to be ridged in one axis, but flexible in all others.
  - Achieved by two narrow sections in the profile
  - This typically sacrifices the axial stiffness for off axis flexibility
    - Bad for stability
- Universal joints (centre and bottom right) replicate the kinematics
  - Achieved by two spherical bearings in series
  - Axial stiffness is dependent on bearing diameter, but independent of off-axis flexibility
- Originally developed for HL-LHC by BE-GM.
  - Thanks to Mateusz Sosin and Hélène Durand for helping with the design and sharing their test results



Top: A steel flexure from the adjustment platform and the fully assembled test joint. Bottom: The test joint disassembled, showing a spherical bearing.

SAS Alignment Platform Prototype Testing

- Testing procedure:
  - Each axis was manually independently adjusted by a set number of revolutions, and the displacement was measured
    - Resolution
    - Adjustment rate and linearity
    - Range and backlash
- Testing results:
  - Sub micron resolution in all axes
  - Adjustment rate close to design
  - Backlash <16µm
    - Current design does not attempt to eliminate backlash, but it can be avoided through correct operation
- There is no obvious limitation or impingement from the longitudinal joint on the other axes.
- All the test data is comparable to the V3 prototype



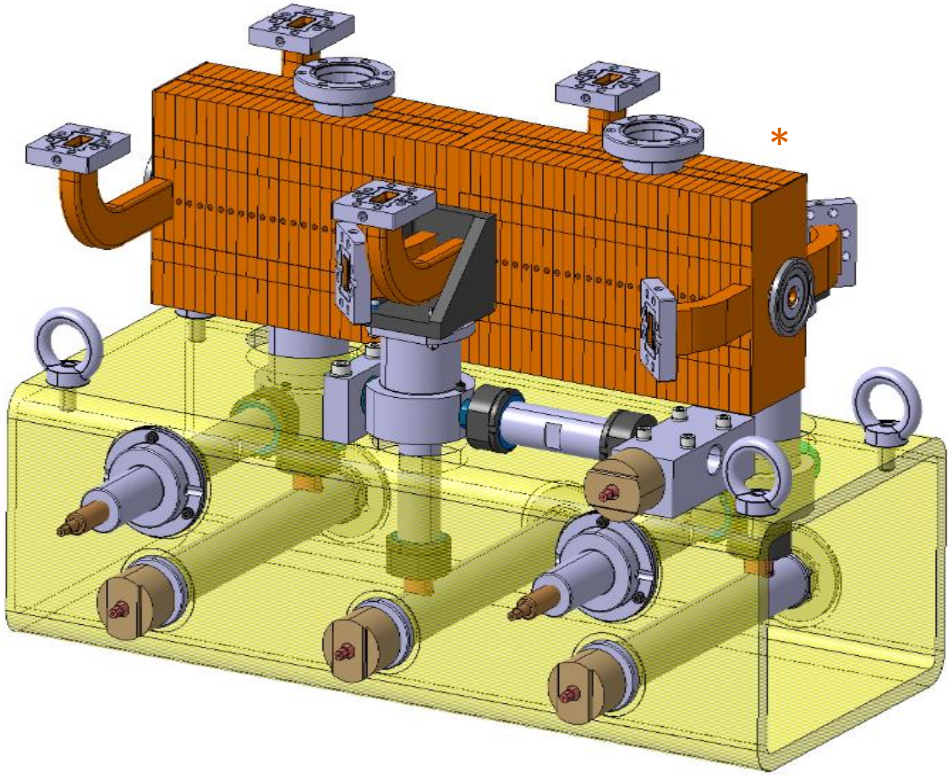
Above: A plot of the vertical axis #3 averaged test results

	Wedge Axes					Differential Thread Axes		
Axis	**V1**	**V2**	**V3**	**LN**	**Design**	**L1**	**L2**	**Design**
Resolution	<1	<1	<1	<1	<1	<1	<1	<1
Average Gradient	31.58	32.59	32.45	28.4	30	38.28	35.22	40
Average Backlash	12.43	15.79	12.48	7.2	--	2.84	0.01	--
Max Non-Linearity	6.80%	6.40%	1.70%	2.80%	--	6.20%	15.10%	--

Above: A summary table of all the axes test results

SAS Alignment Platform Future Prototype

- We are currently manufacturing a V4 adjustment platform prototype
  - Based around six universal joints
  - Based upon the results of the stability analysis and optimisation
    - 22mm diameter (commercially available) bearings
    - 260mm x 160mm landscape orientation girder
- Compatible with existing structures (round disk and manifolds), dummy structures (used in prototypes V1-3.5), and future SmartDisc structures*



Top: The SAS adjustment platform V4

* Rectangular disc structure currently in design and development, credit Pedro Morales Sanchez

Alignment & Stability Requirements

CLIC structure static alignment and stability constraints.

Suitability of current module design.

Alignment and Stability Requirements

- Structure Alignment Requirements (PIP):
    - Cavity offset relative to girder axis = 14µm
    - Cavity tilt relative to girder axis = 141µrad
  - The prototype structure alignment systems have demonstrated the ability to meet these requirements
- Structure Jitter Requirements (CDR)
  - RMS jitter tolerance which leads to a 1% luminosity loss
    - Accelerating structure horizontal position = 8µm
    - **Accelerating structure vertical position = 1.4µm**
    - Accelerating structure horizontal tilt = 6µrad
    - Accelerating structure vertical tilt = 1.1µrad
  - These tolerances are tighter (particularly the vertical position), and harder to compare our current design against, and require consideration of the local sources of vibration:
    - Known: e.g. ground motion, technical noise
    - Unknown: e.g. Structure water cooling, tunnel airflow, other equipment

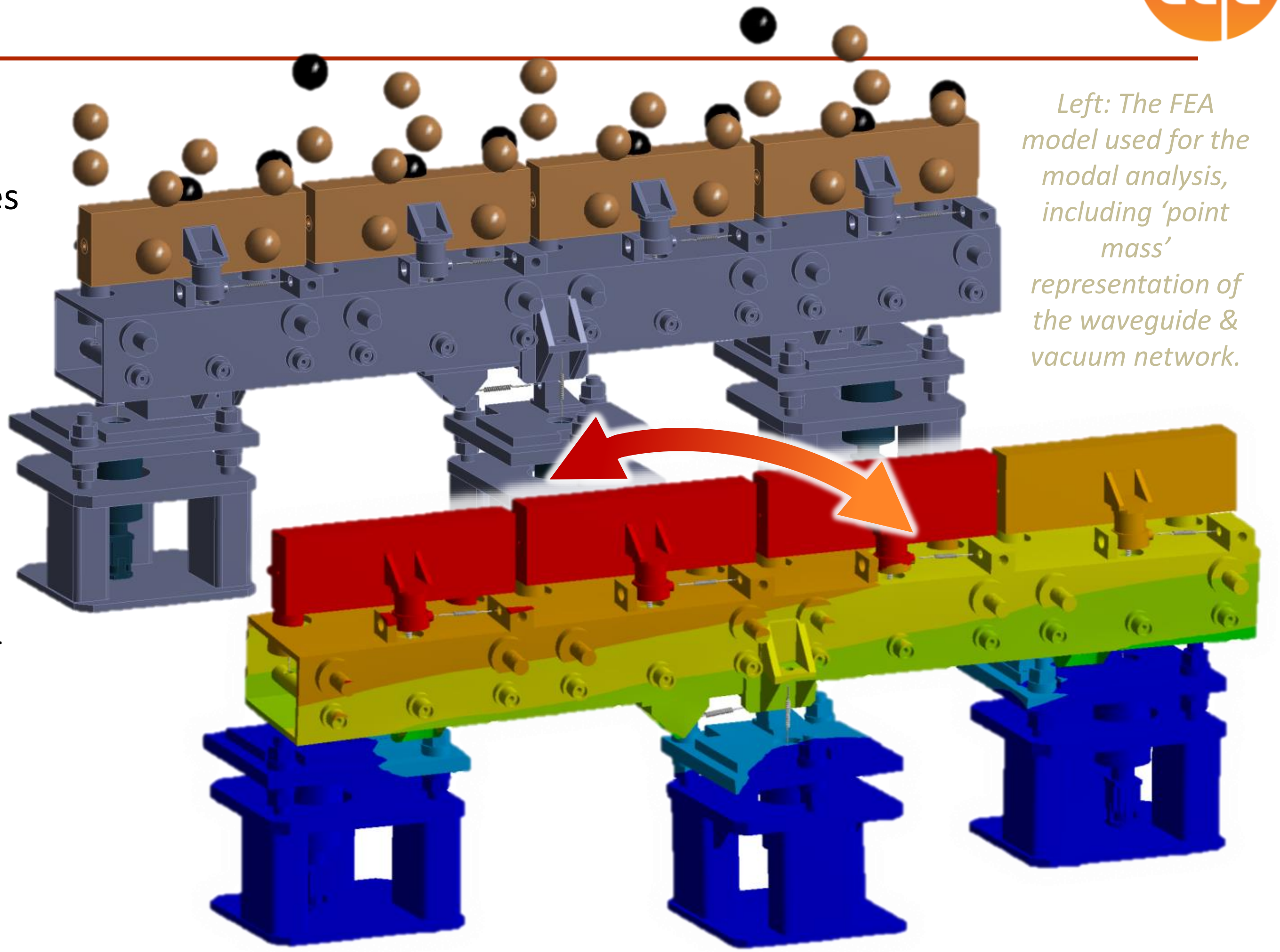
Alignment & Stability Analysis

Analyses performed: Modal, Harmonic, & Random Vibration.

Module alignment systems considered: Flexures, Cam, & Joints.

Modal Analysis

- Performed to determine the harmonic frequencies of the module.
- The TBM design uses a `hard-mount' passive vibration isolation system:
    - Similar to the base of the main beam quadrupoles (MBQs)
    - Unlike the pre-isolation of the CLIC final focusing magnets.
- The CLIC feedback system is good at suppressing frequencies below 1 Hz but amplifies the range 4-25 Hz, & immediately above and below the operational frequency [1].
- The goal of the optimisation is to increase the fundamental frequency to significantly greater than the 50Hz operational frequency



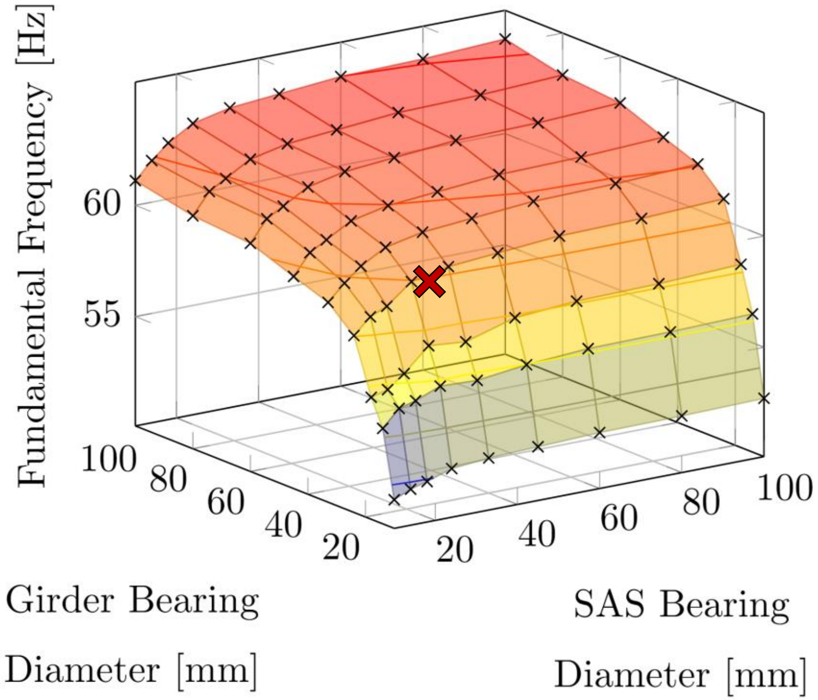
Left: The FEA model used for the modal analysis, including 'point mass' representation of the waveguide & vacuum network.

Above: A contour plot of the primary mode.

[1.] C. Gohil, Dynamic Imperfections in the Compact Linear Collider (2020) http://cds.cern.ch/record/2724824/files/405CERN-THESIS-2020-074.pdf

Modal Analysis Optimisation

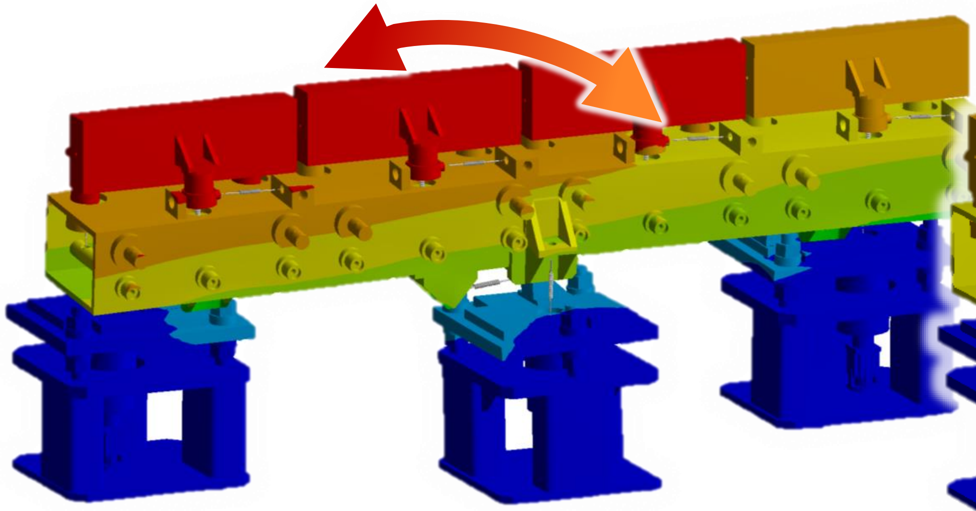
- The harmonic frequencies are extracted from the Finite Element stiffness matrix
  - Very low axial stiffness was the main motivation to move from flexures to universal joints
- The stiffness of the universal joints are closely related to the diameter of the spherical bearings
  - Joint stiffness determined through axisymmetric analysis, validated against test data
- Considering universal joints for both the girder support system, and the structure support systems, we can perform an impact study:
  - Increasing the both bearing diameters increases natural frequency
  - The girder support bearing is more significant
  - Very large bearings show diminishing returns
- Chosen design:
  - 22mm bearings for the SAS supports (commercially available)
  - 35mm bearings for the Girder supports (custom)
  - Increases the fundamental frequency to ~60Hz



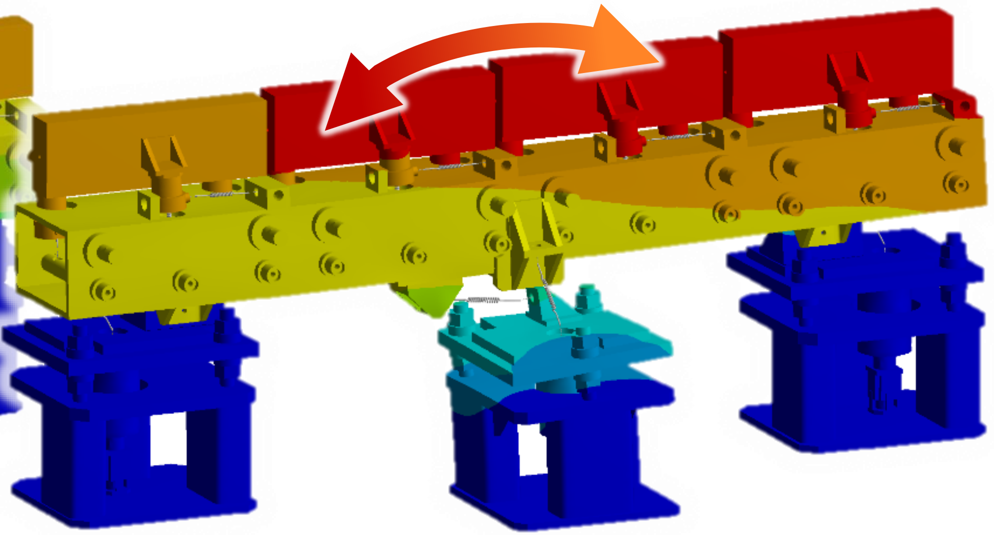
Top: The fundamental frequency of a module compared to the diameter of the spherical bearings used in the girder positioning system, and the SAS positioning systems

Modal Analysis

- The two lowest harmonic frequencies are very close (59.6Hz and 60.6Hz) and result in a lateral and longitudinal swaying:
  - Unsuprising as the support system relies upon three vertical joints, but two lateral and one longitudinal joints
  - The vertical jitter tolerance is much tighter than the lateral or longitudinal tolerances, however both these harmonic modes also result in displacements in the vertical axis, so must be considered



Above: A contour plot of the primary mode.



Above: A contour plot of the secondary mode.

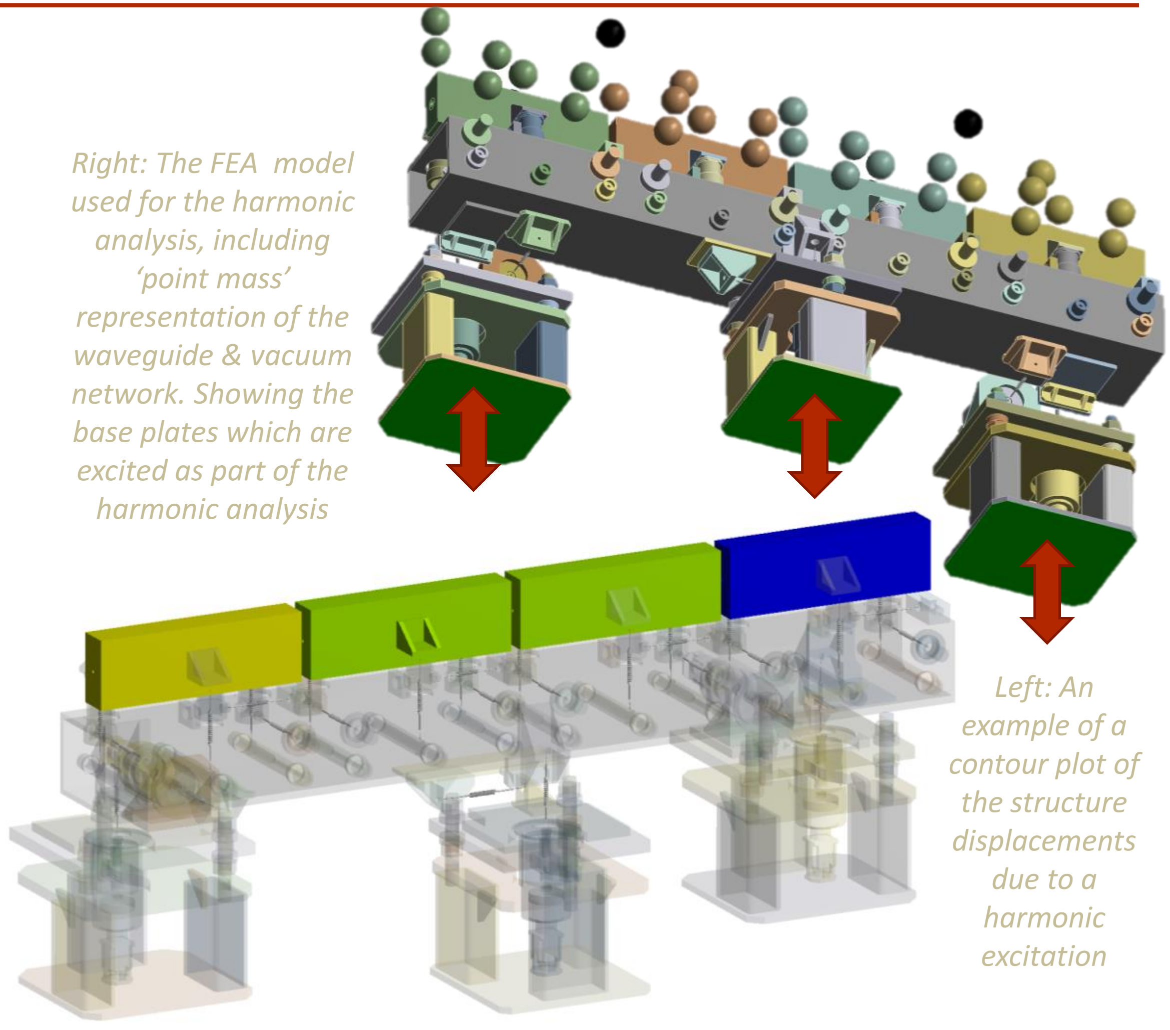
Harmonic Modes	
Mode #	**Frequency [Hz]**
ω_0	59.6
ω_1	60.6
ω_2	81.0
ω_3	103.0
ω_4	126.0
ω_5	130.0

Above: Table of the harmonic frequencies

Harmonic Analysis

- A nominal oscillation is applied to each of the support base plates, and sweep across a frequency range
  - 0 – 300 Hz
  - The three bases can be in-phase or out-of-phase
  - A 3% damping ratio is assumed
- The average displacement of each structure a the beam axis can be calculated
  - This can be averaged across all four structures
- Plotting these displacements against the input frequency produces the Frequency Response function of each structure

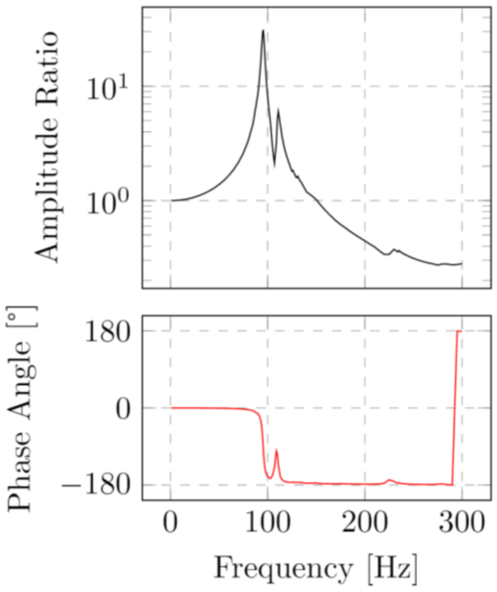
Right: The FEA model used for the harmonic analysis, including 'point mass' representation of the waveguide & vacuum network. Showing the base plates which are excited as part of the harmonic analysis



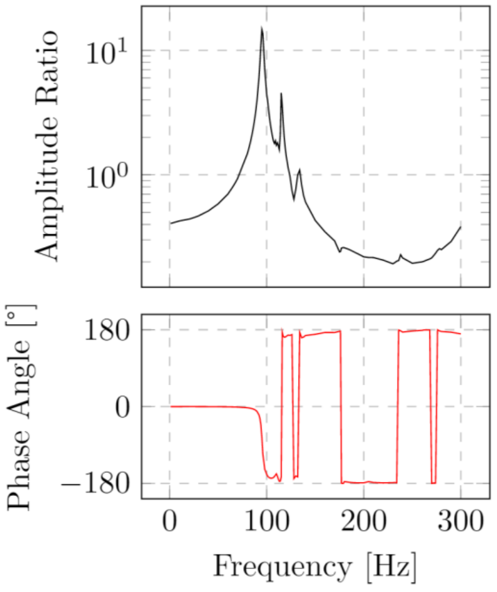
Left: An example of a contour plot of the structure displacements due to a harmonic excitation

Harmonic Analysis

- For a vertical excitation, the average beam-line position of the module experiences an amplification of this displacement up to a peak at 100Hz
  - The in-phase excitation produces a gain greater than 1 until frequencies >150Hz, with a peak around 100Hz
    - Expected behaviour for a hard mount system
  - The out-of-phase excitation produces a gain greater than 1 for frequencies between 70Hz & 150Hz, with peaks around 100Hz
  - The peaks at 80Hz and 100Hz align with the harmonic frequencies which produced the largest vertical displacements
- At low frequencies the vertical ground noise is broadly coherent over 2m (the length of the module)
  - Above 40Hz this coherence decreases
  - The large peak at 100Hz could be significant
- More studies needed



(a) Z Axis SAS frequency response.



(a) Z Axis SAS frequency response.

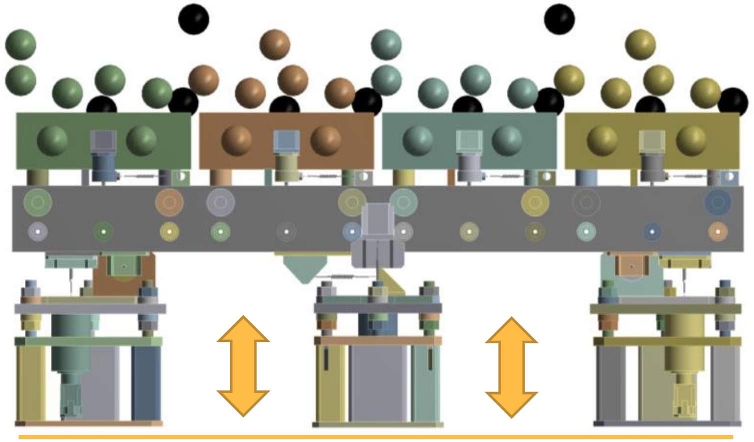
Above: The in-phase (left) and out-of-phase (right) transfer functions averaged across the four SAS for the Z-Axis (vertical)

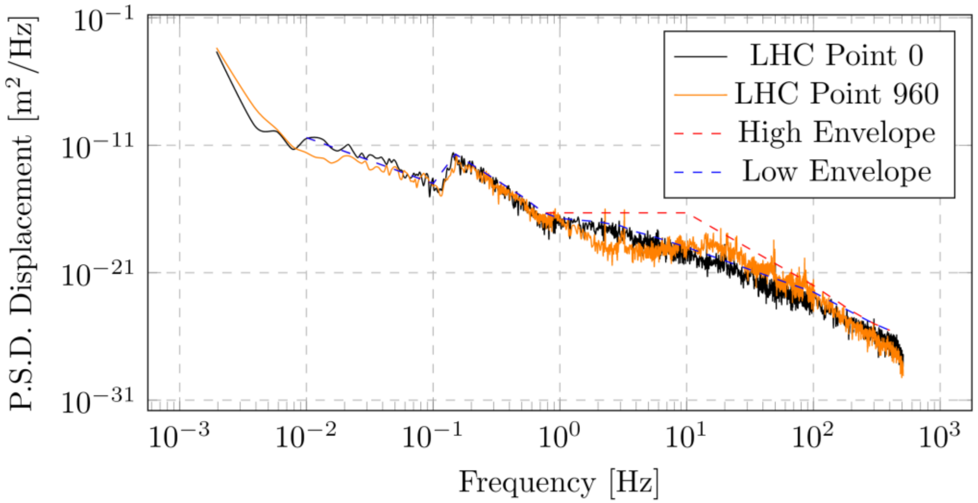
Ground Noise Analysis

- Random vibration analysis:
  - A spectrum analysis technique which calculates the probability distribution of a result due to some random excitation, using the combined effects from each harmonic mode.
    - Commonly used for jitter in alignment of optical equipment.
  - Assumes a Gaussian distribution of results.
  - Takes a Power Spectral Density function as the input: e.g. ground noise data.

- Using this method it is possible to statistically quantify the displacement of the module due to the Ground Noise
  - Gives the standard deviation of the displacements, which can be compared to the 1.4µm RMS value from the CDR

Right: The FEA mode used for the random noise analysis. Showing the base plates which are excited.

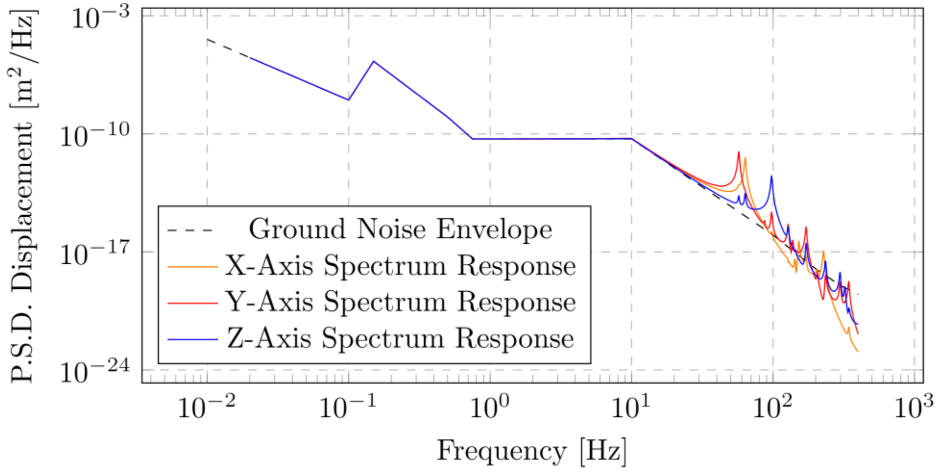




Top: LHC Ground Noise data from Points 0 & 960, and envelope curves

Ground Noise Analysis

- The ground noise data is typically characterised by its Power Spectral Density
- It is possible to plot the output Power Spectral Density on top of the input
  - The peaks align with the modal frequencies, agreeing with the modal and harmonic analysis
- For ground noise greater than 0.1Hz, the 1σ vertical displacement of the structures is <0.05μm, well within tolerance
- An important consideration when quantifying the RMS displacements of the module is the frequency range
  - We have ground noise data from <0.01Hz up to >500Hz
  - Including the very low data significantly skews the data
- Frequency ranges considered:
  - Below 1Hz vibrations are well suppressed by the CLIC beam trajectory feedback system.
  - Between 0.1Hz and 1Hz the ground noise is coherent over lengths around 1km



Left: The (input) ground noise PSD and the (output) X, Y, and Z response PSD of the module

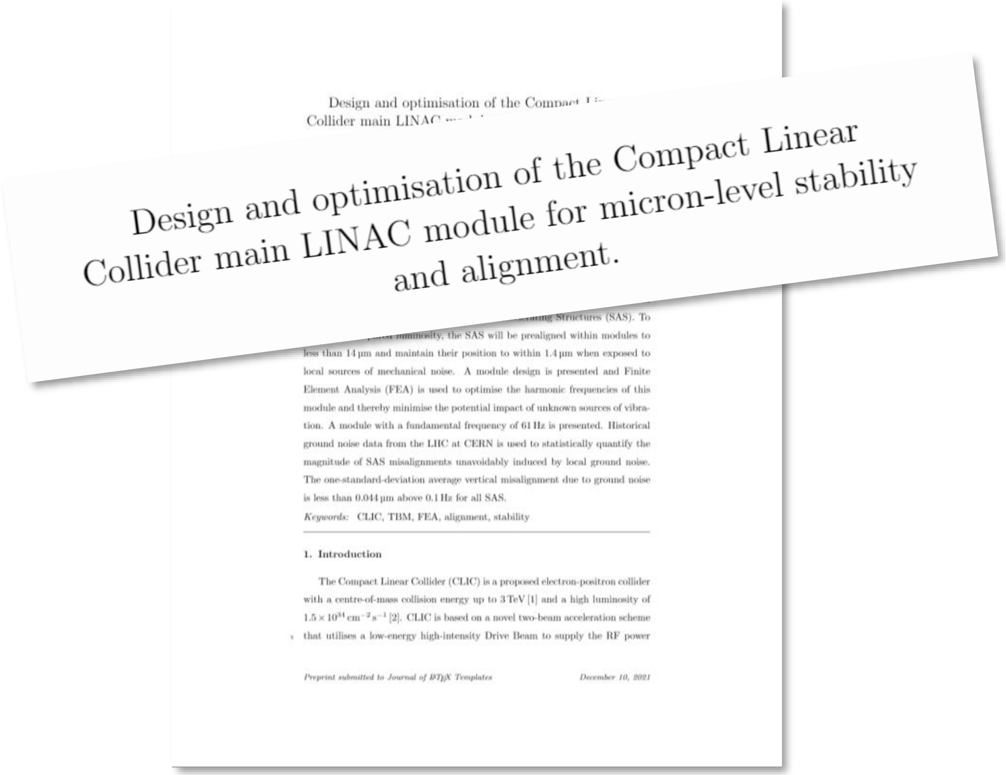
	Frequency Range		
Axis	**0.08Hz+**	**0.1Hz+**	**1Hz+**
X	*0.445*	*0.014*	*0.003*
Y	*1.375*	*0.028*	*0.003*
Z	*0.905*	*0.042*	*0.001*

Above: 1-sigma displacement of the module [μm]

Alignment & Stability Analysis Summary

- **Modal analysis**
  - Initial optimisation goals me, fundamental modes around 60Hz
  - Exact frequencies will depend on currently unknown factors
    - Structure design
    - Waveguide and vacuum network designs; height & mass
  - Of limited use when comparing directly against the PIP and CDR requirements
- **Harmonic analysis**
  - Extracted transfer functions for individual SAS and module average
  - Agrees with the other analysis, and expected behaviour
  - Could potentially be used in further stability and emittance growth studies
- **Ground noise analysis**
  - Real input data allows comparison to the PIP specification
  - Highlighted the importance of the frequency range when considering the ground noise
  - Further work is needed to fully understand and quantify the impact

We have written a paper covering the analysis & optimisation in more detail:


Design and optimisation of the Compact Linear Collider main LINAC module for micron-level stability and alignment.

... Structures (SAS). To ... luminosity, the SAS will be prealigned within modules to less than 14 µm and maintain their position to within 1.4 µm when exposed to local sources of mechanical noise. A module design is presented and Finite Element Analysis (FEA) is used to optimise the harmonic frequencies of this module and thereby minimise the potential impact of unknown sources of vibration. A module with a fundamental frequency of 61 Hz is presented. Historical ground noise data from the LHC at CERN is used to statistically quantify the magnitude of SAS misalignments unavoidably induced by local ground noise. The one-standard-deviation average vertical misalignment due to ground noise is less than 0.044 µm above 0.1 Hz for all SAS.

Keywords: CLIC, TBM, FEA, alignment, stability

1. Introduction

The Compact Linear Collider (CLIC) is a proposed electron-positron collider with a centre-of-mass collision energy up to 3 TeV [1] and a high luminosity of $1.5 \times 10^{34}\ \mathrm{cm^{-2}\,s^{-1}}$ [2]. CLIC is based on a novel two-beam acceleration scheme that utilises a low-energy high-intensity Drive Beam to supply the RF power

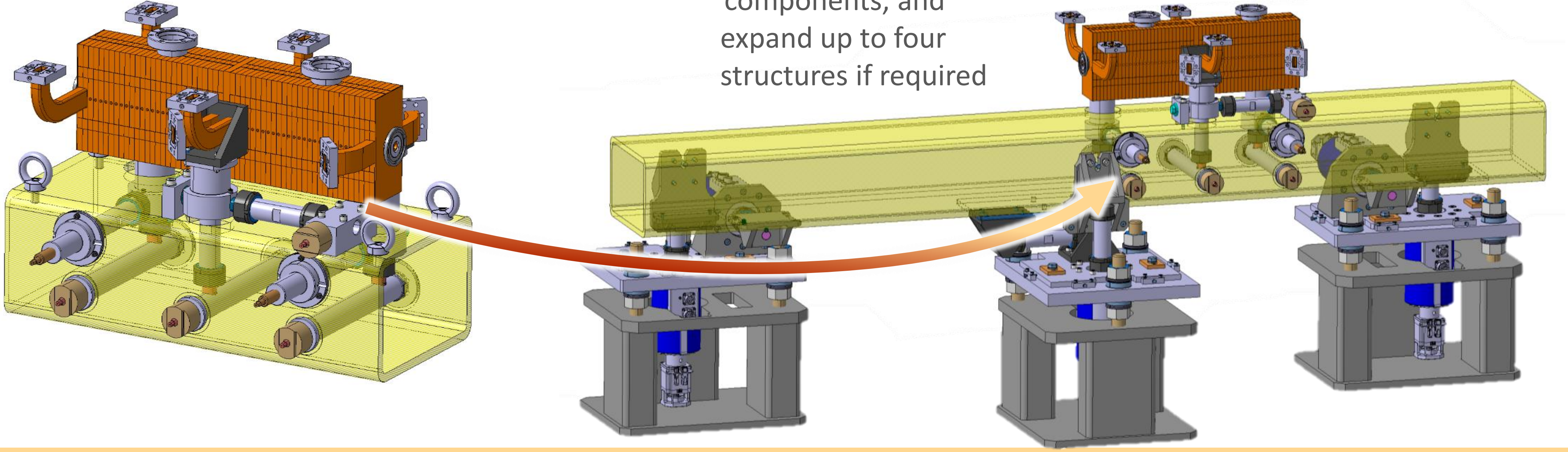
Preprint submitted to Journal of LaTeX Templates *December 10, 2021*

Future work

The current and immediate work of the module team. Prototypes currently on manufacture or final design work. Future aims.

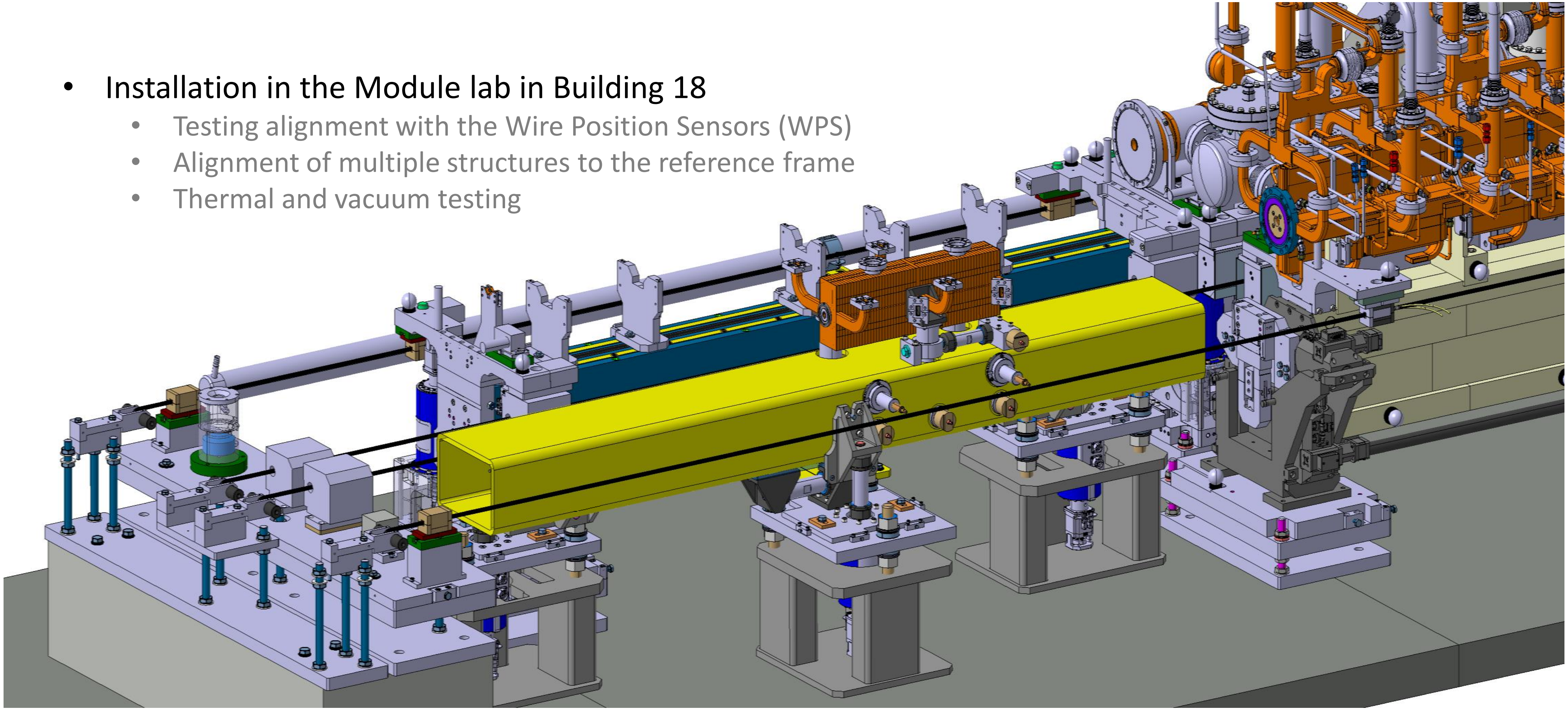
Future Module Work

- V4 SAS Adjustment platform prototype
  - Six universal joints with 22mm diameter (commercially available) bearings
  - 260mm x 160mm landscape orientation girder
- Girder positioning system prototype seeing
  - Based around six universal joints with 35mm diameter spherical bearings.
  - Five linear actuators to provide the active alignment capacity.
    - Longitudinal position defined but not actively adjusted
  - Capacity to integrate the V4 SAS adjustment platform components, and expand up to four structures if required



Future Module Work

- Installation in the Module lab in Building 18
    - Testing alignment with the Wire Position Sensors (WPS)
    - Alignment of multiple structures to the reference frame
    - Thermal and vacuum testing



Thank you for listening

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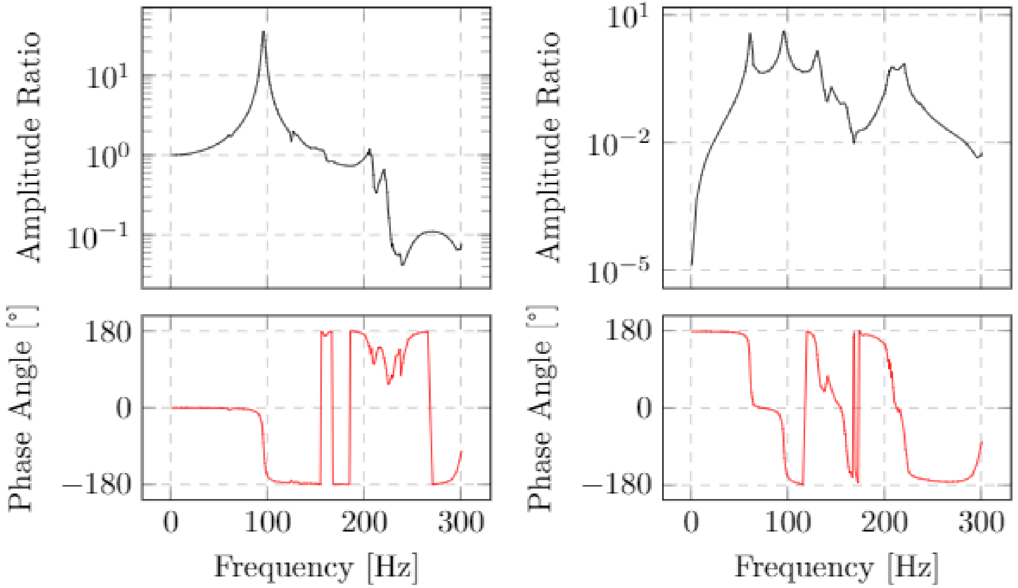
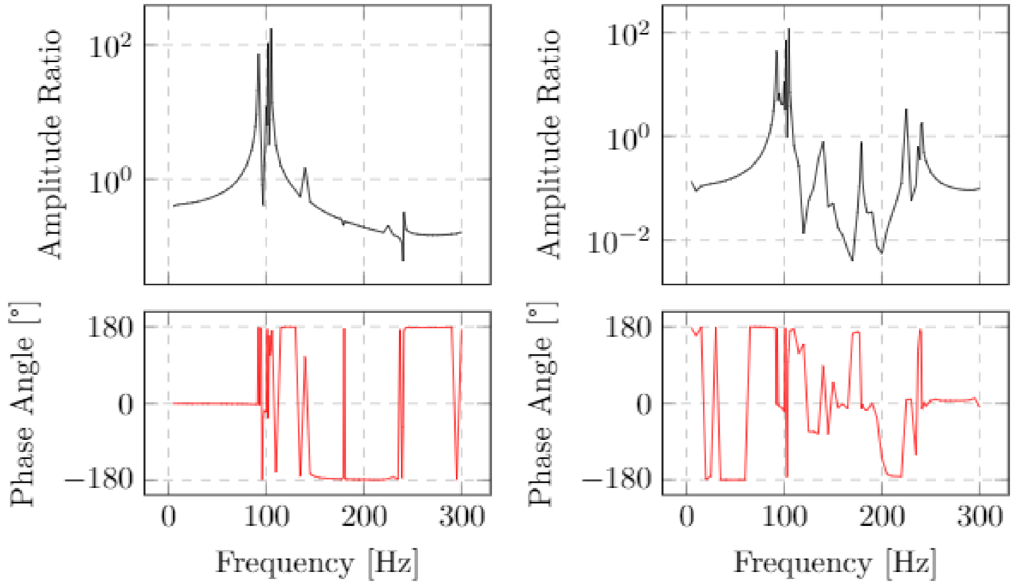
Bonus Slide: Frequency Response



(a) Z Axis SAS frequency response.　(b) Y axis SAS frequency response.

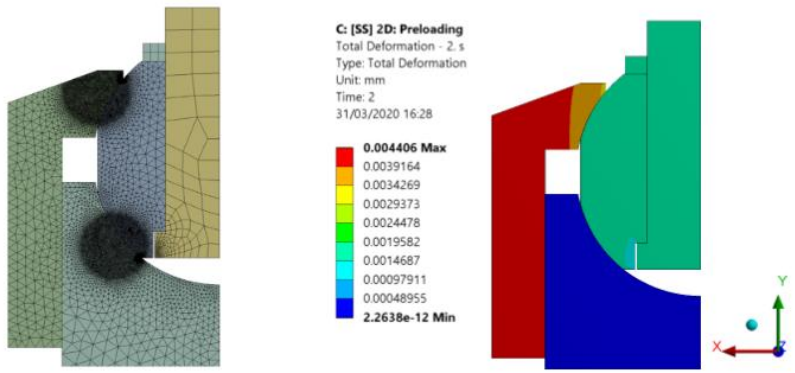
Figure 14: The frequency response of the module due to in-phase base excitation in the z axis and measured at the beam axis.



(a) Z Axis SAS frequency response.　(b) Y axis SAS frequency response.

Figure 15: The frequency response of the module due to out-of-phase base excitation in the z axis and measured at the beam axis.

Bonus Slide: Joint Axisymmetric Analysis



(a) The mesh used for analysis.

(b) The deformation of the spherical bearing under preload.

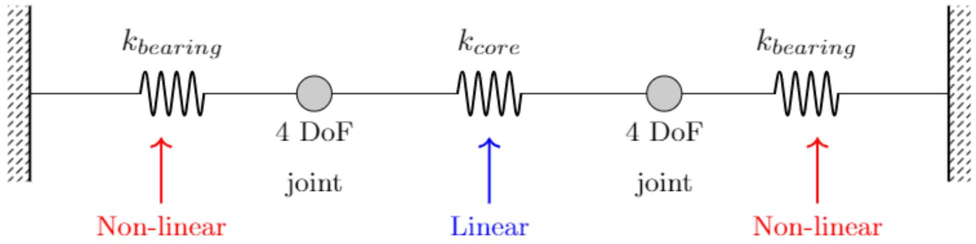
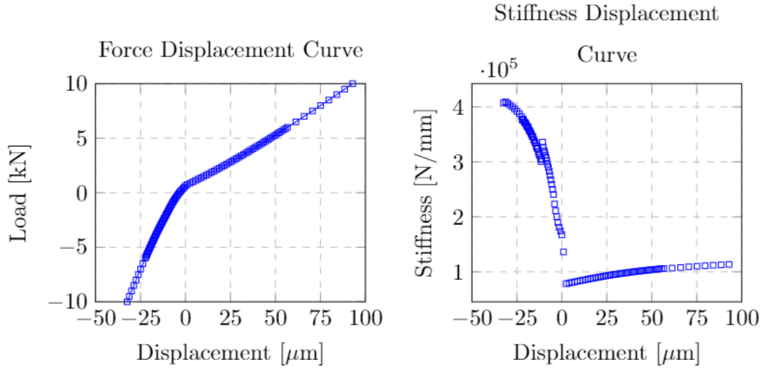


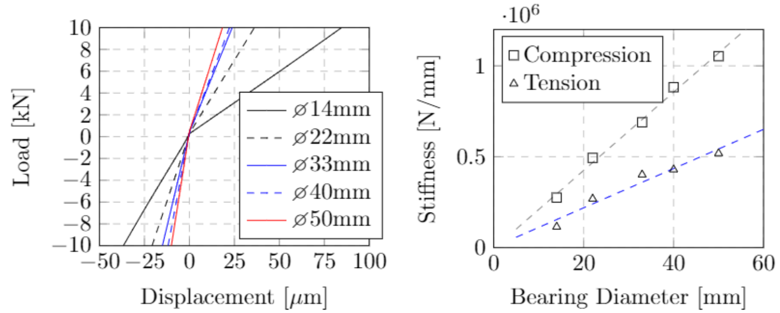
Figure 11: The spring model of a universal joint.



(a) Load-displacement results

(b) Stiffness-displacement results

Figure 8: The measured response of a 14 mm diameter bearing assembly under load, assumed to have an initial line contact.

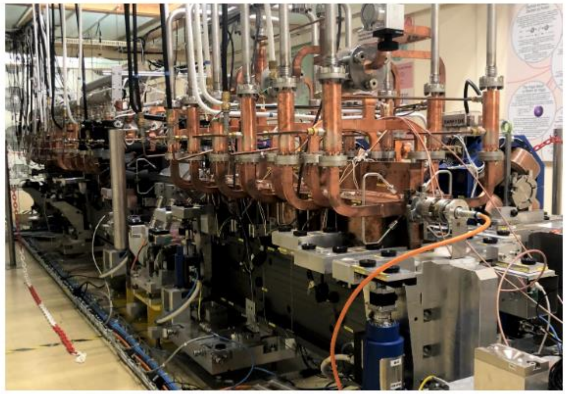


(a) The force-displacement curves for optimised joints with various bearing diameters.

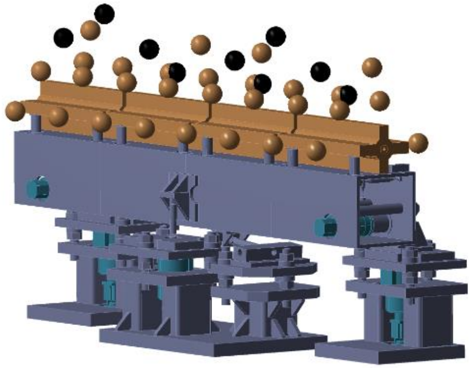
(b) Joint bearing diameter compared to compressive and tensile joint stiffness.

Figure 9: The results of analysis of the optimised joint analyses.

Bonus Slide: Waveguide Network

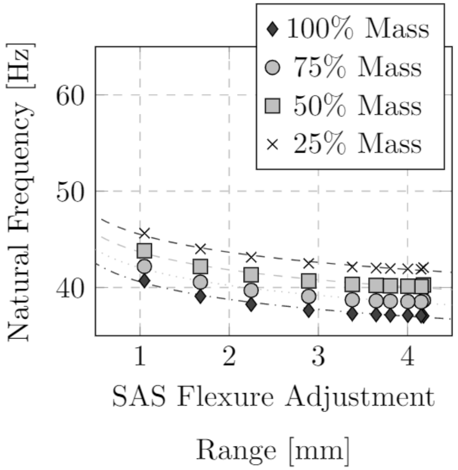


(a) A CLIC module prototype including all waveguide-network components.

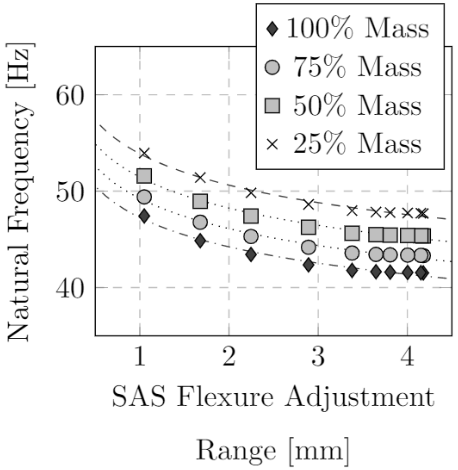


(b) A CLIC module model including waveguide-network point masses.

Figure 6: An image of a prototype CLIC module assembly (a) and a similar module modelled within FEA (b).



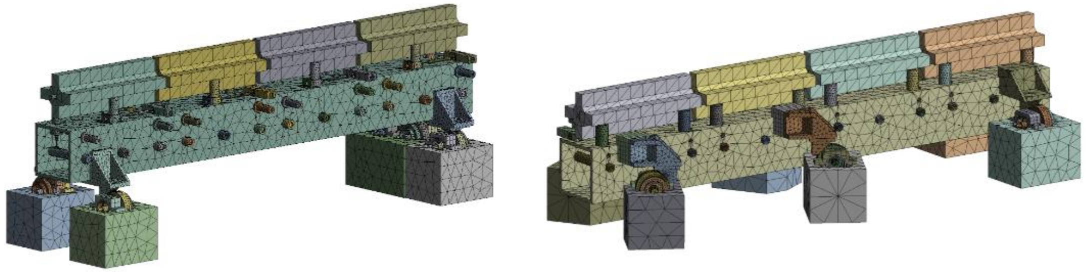
(a) Flexure supported girder.



(b) Cam mover supported girder.

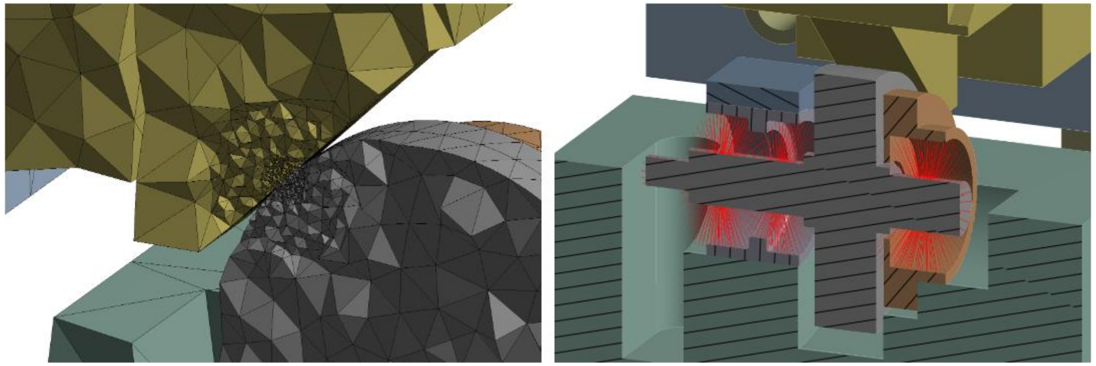
Figure 12: The natural frequency [Hz] of the flexure supported Main Beam girder compared to the adjustment range of the SAS flexures [mm] for the three waveguide system masses considered.

Bonus Slide: Cam Mover



(a) (b)

Figure 4: An image of a 5DoF cam system (a) and a 6DoF cams system.



(a) (b)

Figure 5: Specifics of the cam support system analyses, including the contact point refinement (a) and the bearing contact settings (b).

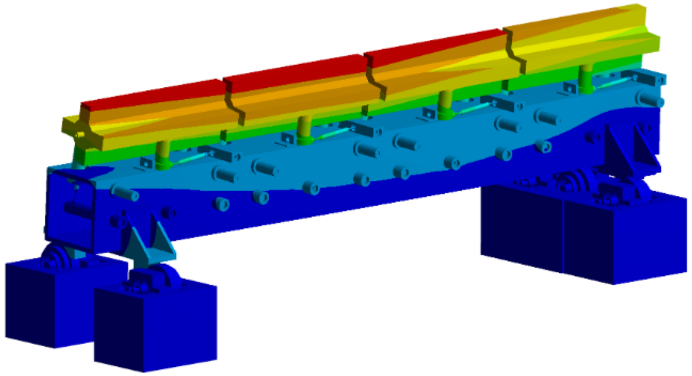


Figure 8: The primary mode of oscillation for the 5 degree of freedom cam mover system.

Bonus Slide: Flexure Range

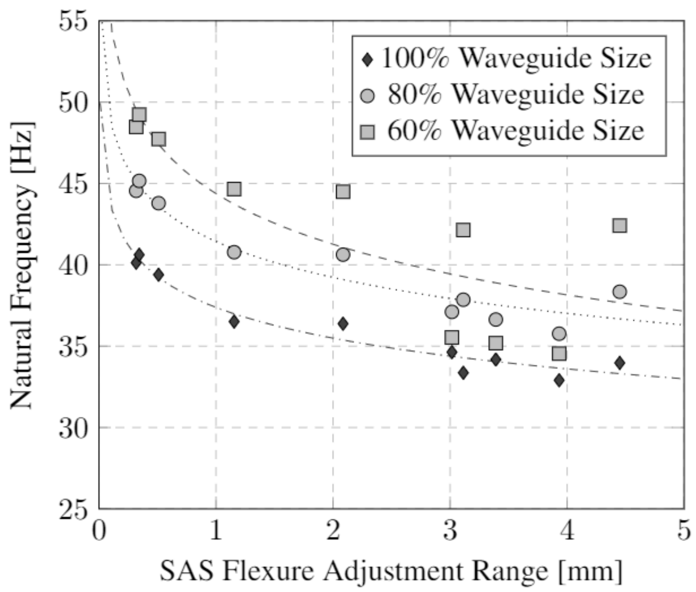


Figure 17: The natural frequency [Hz] of the flexure supported Main Beam girder compared to the adjustment range of the SAS flexures [±mm] for the three waveguide system masses considered.

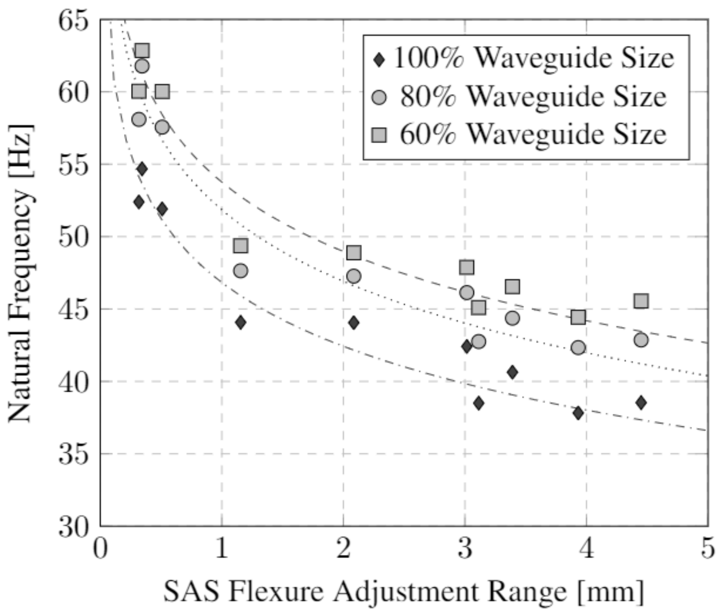


Figure 18: The natural frequency [Hz] of the cam supported Main Beam girder compared to the adjustment range of the SAS flexures [±mm] for the three Waveguide masses considered.