

STRUCTURAL DYNAMIC MODELLING AND MEASUREMENT OF SWISSFEL BUNCH COMPRESSOR

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Abstract

Magnetic chicanes are used in accelerator facilities to longitudinally compress the accelerated particle bunches. The second compression chicane of SwissFEL consists of four dipole magnets bending the beam on the horizontal plane along a C-shaped orbit and has a total length of 17 m. The position of the two central dipoles can be continuously adjusted to achieve the required transverse offset in order to realize a wide range of compression schemes. To ensure the required mechanical stability of the accelerator components sitting on the long and movable steel girder (7.7 m), it is essential to design a stiff support structure with high eigenfrequencies. In the design stage, displacement frequency responses are calculated in a modal based linear dynamic analysis using finite element method to ensure vibration amplitude below 1 micrometer. Special considerations are given to the modelling of linear guide systems, as they introduce non-linear support conditions and need to be adequately simplified in the calculation. After completing the second bunch compressor (BC2) assembly, vibration measurements were performed. Finally, the validation of the numerical model by measurement results will be presented.

INTRODUCTION

BC2 consists of two welded steel girder structures (Figure 1), which are supported on granite tables. Each granite table is fixed to floor by three or four rigid jacks with built-in commercial levelling wedge elements.

From the middle table girders can be moved together with dipole 2 and 3 transversally 505 mm. With a girder length of 7.7 m (dipole separation), the bending angle is 3.8 degree. Girder is supported on pivot bearing for rotation. A linear stage is used for transversal movement. A second linear stage for longitudinal movement is rotatable against the first stage (Figure 2).

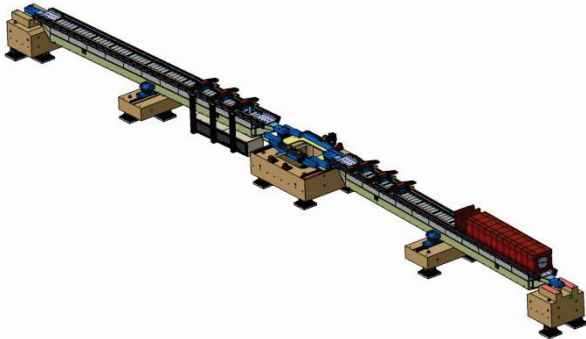


Figure 1: SwissFEL BC2.

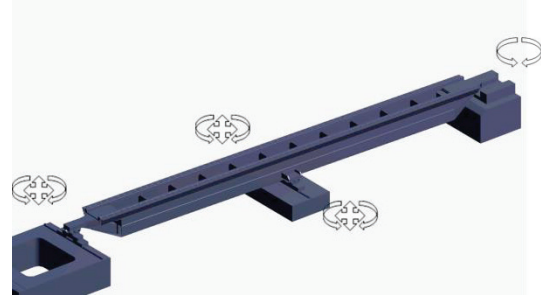


Figure 2: Girder kinematics.

The first steel girder in the beam direction has a total weight of 3'200 kg, including 1'900 kg lead and polyethylene (PE) material for radiation shielding. To ensure the proper operation of accelerator components (BPM and magnets) on the girder, a mechanical vibrational stability below 1 μm is required.

In order to fulfil the stringent design goal of a long and movable steel girder which is in the meantime vibrational stable, finite element (FE) analysis has been performed in the design phase. The modelling is finally verified by vibration measurement.

FINITE ELEMENT MODELLING

The girder structure is welded from rectangular hollow steel sections and steel plates with thickness from 5 mm to 10 mm. Shell elements with 6 degrees of freedom (DOF), three rotational and three translational, have been employed for efficient presentation of in-plane and bending stiffness. Point mass elements have been used to represent shielding masses in 3D space. The voluminous support consoles have been modelled with 3D solid elements (Figure 3). The linear contact connecting shell and solid elements is defined by Multi-Point-Constraint (MPC) formulation with the coupling of rotational DOF to translational DOF.

The FE model has following parameters:

Number of nodes:	329'000
Number of elements:	216'000
Incl. shell elements:	122'000
Degrees of freedom:	1'326'000

Linear static and dynamic calculations have been performed with finite element program ANSYS. FE Modell was prepared from CAD geometry with SpaceClaim Direct modeler.

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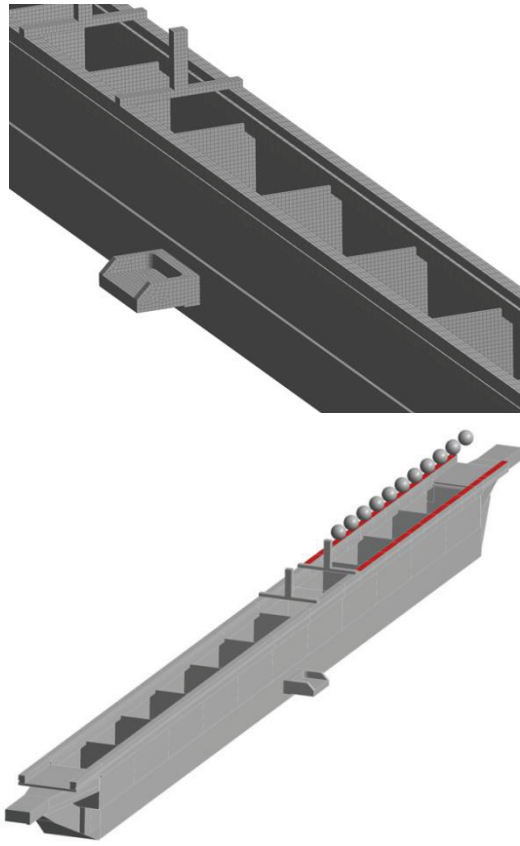


Figure 3: Finite element model.

Support conditions involving linear bearings introduce nonlinearity to the model. In a rigid dynamic analysis dedicated to kinematic simulation, it is simulated to be free a DOF. In a linear steady-state dynamic analysis, the linear guide can be represented by spring elements with equivalent stiffness. The determination of the stiffness requires detailed analysis and measurement [1]. Moreover, the stiffness depending on the preloading on the stage may not be linear. In the early design phase, not all required detail information was available and hence approximation was necessary.

Provided that friction forces in the joint are high enough to withstand excitation forces, joint is well defined and contact is closed. On the other hand, if the excitation force is higher than friction force, the contact becomes open, and we have a non-linearity in the model. We started with the stable situation of closed joint, and finally checked if the condition is fulfilled in a random vibration analysis. It is helpful to model linear bearing constraints with different stiffness and to investigate the sensitivity of structural response. Very stiff constraint stiffness leads to a fixed boundary condition, while extreme soft stiffness leads to a free boundary condition.

STATIC AND MODAL ANALYSIS

We started with a static analysis under gravity load, followed with an undamped modal analysis. Boundary conditions for linear guides are considered as:

1. Free

2. Constraint stiffness $k=2e4$ N/mm
3. Constraint stiffness $k=2e7$ N/mm
4. Fixed

Maximum vertical displacements due to gravity load are between 0.152 mm with fixed boundary conditions and 0.157 mm with free boundary conditions (Figure 4). The influence of the linear guide stiffness is small.



Figure 4: Deflection under gravity load

From vertical deformation, bending frequency can be estimated analytically considering relationship between natural frequency f in Hz and maximum static deflection d in mm for pinned-pinned prismatic beam [2]:

$$f = 17.8/\sqrt{d}$$

The first analytical bending frequencies are between 45.7 Hz with fixed boundaries and 45.1 Hz with free boundaries, they are comparable with results from FE analysis in Table 1 between 41.3 and 42 Hz (Figure 6). Except for free boundary conditions, other three models have the same first eigenmode as a transversal torsional mode (Figure 5). The natural frequencies for all three models between 20.3 to 22.8 Hz are quite close to each other.

Table 1: Summary of Natural Frequencies [Hz] with Mode Description

Free Hz	Soft Hz	Stiff Hz	Fixed Hz	Mode description
1.6				<i>Rotational mode</i>
16.4	20.3	22.7	22.8	<i>first torsional mode</i>
31.2	34.6			<i>first transversal bending</i>
41.3	41.7	42.0	41.8	<i>first vertical bending</i>
47.8	45.5	48.6	49.0	<i>second torsional mode</i>
		69.6	70.1	<i>first transversal bending</i>



Figure 5: Mode shape of first torsional mode.

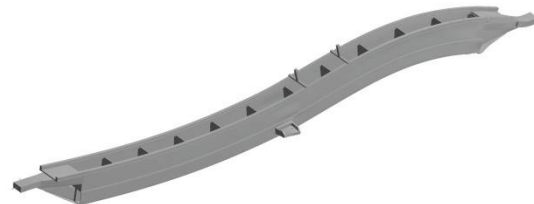


Figure 6: Mode shape of first vertical bending mode.

HARMONIC RESPONSE ANALYSIS

In a harmonic response analysis, system response to steady-state sinusoidal loading at a given frequency is solved. The general equation of motion is

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}$$

With mass matrix $[M]$, damping $[C]$, and stiffness $[K]$ Both excitation $\{F\}$ and response $\{x\}$ are assumed to be harmonic.

$$\begin{aligned}\{F\} &= \{F_{\max}\} e^{j(\Omega t + \psi)} \\ \{x\} &= \{x_{\max}\} e^{j(\Omega t + \phi)}\end{aligned}$$

Ω : imposed circular frequency, ψ , ϕ : phase shift. The mode superposition method solves the harmonic equation in modal coordinates, with displacements expressed as a linear combination of mode shapes.

Ground vibration has in general broadband distribution rather than single harmonic. Before facility has been built, and excitation spectrum is not known, harmonic response analysis provides helpful information on critical frequency and on displacement at important locations. Random vibrations measured on ground at different sites at PSI are below 1 mm/s^2 . In the harmonic response analysis, a base excitation of 1 mm/s^2 up to 100 Hz is applied in phase in all directions and at all supports in order to give a conser-

vation estimation of structural response. Damping ratio is assumed to be 2%. Frequency responses are evaluated at positions of BPM and magnets. Calculation has been performed with soft and stiff constraint stiffness as well as with fixed boundary conditions. The response in vertical and longitudinal directions is lower than in transversal direction due to the fact that the first mode shape is transversal.

Figure 7 shows transversal displacement response at the position of magnet with fixed boundary conditions. At a frequency of 22.8 Hz , the maximum response was $0.9 \text{ }\mu\text{m}$ for both fixed and stiff linear bearings. With soft spring constraints, the maximum response was $1.7 \text{ }\mu\text{m}$. Stiff constraint is important for stable response. To ensure the connection is firm and stable, linear guides are equipped with brakes.

Figure 8 shows transversal acceleration response at the position of magnet with fixed boundary conditions. As the input excitation is defined as 1 mm/s^2 , the value of response acceleration in mm/s^2 depicts the transmissibility function. The maximum transmissibility (amplification) for fixed and stiff bearing is 18.8 and 19.1 , respectively. And with soft springs it is as high as 28.1 . Because the real excitation is rather distributed in a broadband than single harmonic, the amplification will be lower. Stiff support is important to reduce vibration amplitude.

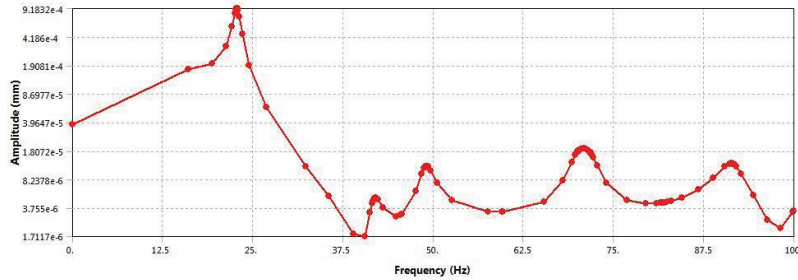


Figure 7: Frequency response of displacement in transversal direction at the position of magnet.

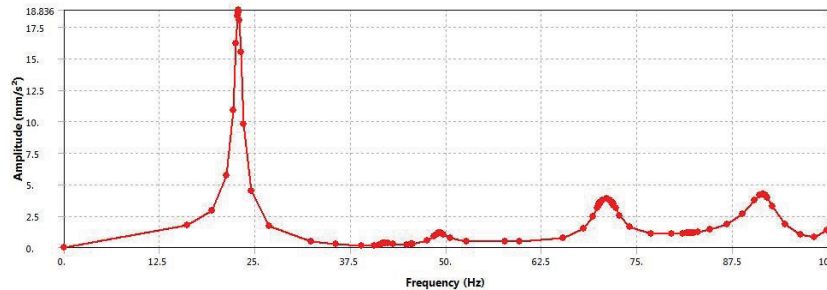


Figure 8: Frequency response of acceleration in transversal direction at the position of magnet.

RANDOM VIBRATION MEASUREMENT

Vibration measurement was performed in May 2016 after BCII is fully equipped and before SwissFEL commissioning. Seismic accelerometers PCB393B31 were placed on the floor, on all three granite tables and on the steel girder both in vertical direction and in horizontal direction

transversal to beam. Beam direction is in Figure 9 from right to left. Three measurement positions on the girder, girder1, girder2 and girder3 are placed equal spaced in beam direction between two granite tables. During the measurement brakes were inactive.

A summary of RMS displacement from 5 to 200 Hz is given in Table 2. The amplification ratio girder to ground



Figure 9 Vibration measurement on SwissFEL BC2

is below 1.35 in transversal direction, and below 1.14 in vertical direction. BC2 structure shows excellent mechanical stability.

Table 2: RMS Displacement from 5 to 200 Hz

Position	displacement
Floor Transversal	37nm
Floor Vertical	29nm
Table transversal	40 nm
Table vertical	31 nm
Girder1 transversal	41 nm
Girder1 vertical	31 nm
Girder2 transversal	49 nm
Girder2 vertical	33 nm
Girder3 transversal	50 nm
Girder3 vertical	32 nm

Transversal spectral displacements from measurement show peaks at 21.4 Hz and 50 Hz (Figure 10). The first and second torsional modes calculated with fixed boundary conditions are 22.8 Hz and 49 Hz, respectively. The modelling of linear bearing either with fixed conditions or with stiff constraint stiffness yields a good prediction of transversal modes. From the vertical deformation, it is however not clear if there is a mode around 42 Hz, which is predicted to be the first bending mode from FE analysis (Figure 11). Vibration level in vertical direction is very low with amplification ratio girder to floor of only 1.13. It is therefore difficult to detect all eigenmodes in this measurement without external excitation.

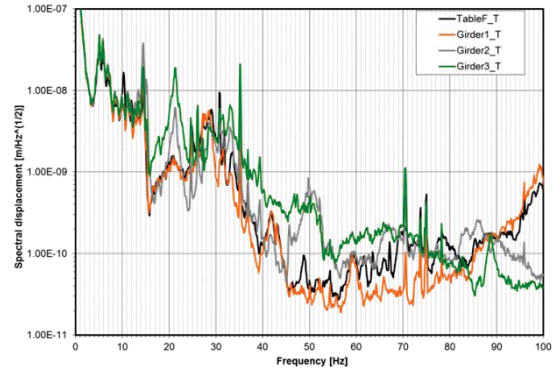


Figure 10: Spectral displacement in horizontal direction transversal to beam.

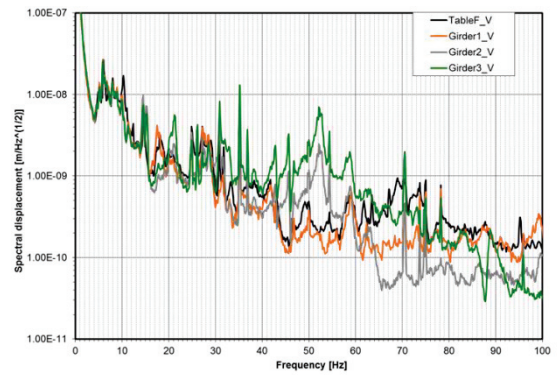


Figure 11: Spectral displacement in vertical direction.

RANDOM VIBRATION ANALYSIS

Random vibration analysis is another spectral method to determine structural response in modal space. The power spectral density (PSD) of acceleration measured on granite table in all three directions is applied as base excitation to all support conditions. For simplicity PSD acceleration from 5 to 100 Hz are applied, a constant damping

ratio of 2% is assumed. PSD response in the same frequency range is calculated. In Figure 12 response displacement in transversal direction with fixed boundary is depicted. As random vibration analysis considers on time history, both excitation and response are based on statistical representation.

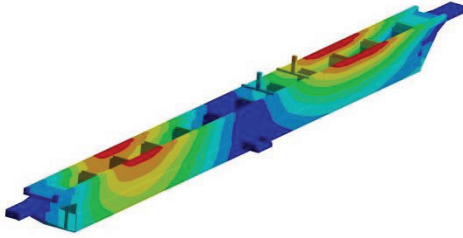


Figure 12: Response PSD displacement in transversal direction on girder.

Table 3: Comparison of RMS Displacement from Measurement and FE Calculation

Position	Measurement	Calculation
Girder1 transversal	41 nm	44 nm
Girder1 vertical	31 nm	31 nm
Girder2 transversal	49 nm	109 nm
Girder2 vertical	33 nm	36 nm
Girder3 transversal	50 nm	44 nm
Girder3 vertical	32 nm	29 nm
Magnet transversal	119 nm	
Magnet vertical	30 nm	

For the girder measurement position 1 and 3 RMS displacements from measurement and from calculation coincide well with each other (Table 3). In the measurement displacement from 100 to 200 Hz is also considered. It makes a contribution of maximum 2 nm. In the middle position between two supports, calculated transversal displacement of 109 nm is twice as much as measured value. A close examination of displacement spectrum depicts high content at high frequency range around 70 Hz. The constant damping ratio of 2% is rather conservative for high frequency vibration.

Reaction forces for all supports have been checked. The maximum reaction force is only 3.3 N in transversal direction and 3.1 N in vertical direction. With a vertical preloading of 9'500 N on the joint estimated from static

calculation, a minimum friction coefficient 0.00034 is required. The available friction force is much higher, and the condition for linear modelling for the joint stiffness is fulfilled. In addition, test measurement with brakes also confirms this conclusion.

SUMMARY AND CONCLUSIONS

BCII shows excellent mechanical stability behavior. The amplification ratio girder to floor is below 1.35 in transversal direction, and below 1.14 in vertical direction. A maximum RMS displacement of 50 nm has been measured. The RMS displacement at magnet position is calculated to be 109 nm.

From vibration measurement, the first two transversal modes from finite element analysis can be confirmed. The modelling of linear bearing with fixed conditions or with stiff constraint stiffness provides a good prediction. The vertical bending mode cannot be identified in this measurement as vertical displacement amplification is very low.

For a linear representation of linear guide systems, dynamic excitation forces are required to be very small compared to sliding friction forces. The verification of excitation forces is given in a random vibration analysis. But this procedure is not straight ahead, because PSD analyses is based on modal representation, and finally on correct modelling of boundary conditions. The solution procedure is iterative. The final confirmation of the linear representation of linear guide system is given by vibration measurement.

ACKNOWLEDGEMENT

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