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DEVELOPMENT OF A SUSPENSION SYSTEM FOR THE ROAD TRANSPORTATION OF CRYOMODULE SSR1 THORUGH A MULTILEVEL FINITE ELEMENT-MULTIBODY APPROACH

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Abstract

The on-road transportation of cryomodules (CM) is a critical phase during which the structure may be subject to relevant dynamic loading. Thus, an accurate design of Transportation Tool (TT), equipped with a proper suspension system, is mandatory. In this paper the TT design for the PIP-II proto SSR1 CM is presented. A finite element (FE) model was developed considering the main CM parts. However, the full model was not suited for the design of the suspension system because of its computational time. Thus, it was exported as a Modal Neutral File to a multibody (MB) software, where minor components were modelled as rigid bodies or lumped stiffnesses. The reduced MB model considerably shortened the computational time and it was exploited for the design of the TT, which includes helical isolators (HI) acting as a mechanical filter. A real 3D acceleration profile, acquired during the transportation of a LCLS-II CM from Fermilab to SLAC, was used to validate the TT effectiveness in reducing the vibrational loading. In addition, the results of the MB analysis were used to perform FE analysis of critical components, such as bellows.

INTRODUCTION

The PIP-II project at Fermilab is a proton driver superconducting linac that consists of five different SRF cavity types: half wave resonator (HWR), 325 MHz single spoke resonators (SSR1, SSR2), and 650 MHz multicell cavities (LB650, HB650) [1, 2]. An important milestone in the design of each of those CMs is the development of a fixtures and procedures to mitigate potential failures during the transportation phases.

The prototype SSR1 is currently being assembled at Fermilab and a transportation tooling was designed since the CM will be subjected to several interfacility road transportations by trucks. The speed of the truck will be limited to about 10 m/s. The activity started studying and understanding the CM dynamic response to vibrational loading through a comprehensive modelling of the CM response exploiting both Finite Element Analysis (FE) and Multi Body analysis (MB), which proven to be effective in complex assembly analysis [3]. Then, it was possible to design a suspension system that acts as a mechanical filter to protect CM components from vibrational loadings generated by the truck and roads. The main deformable components were modelled through FE, then they were imported in a comprehensive MB model which also accounted for minor components, modelled as lamped masses and stiffnesses. This lean MB model could then be used to design the transportation tool, by considering various types of dynamic loading. The effectiveness of the transportation tooling design can be qualitatively performed by MB in terms of main components displacements. The, loading determined through the MB model could also be imported in a detailed FE model of critical components, to assess the expected stress levels during transportation.

FINITE ELEMENT ANALYSIS

An estimation of dynamic behaviour of the CM was needed to design the TT. Since the TT should mainly act as a mechanical filter with respect to the road dynamic loading, it was crucial to assess the main CM natural frequencies in order to set the needed cut-off frequency. This was achieved through a combined Finite Element and Multi-Body approach. Thus, the FE model was firstly developed. Since the full assembly comprises thousands of parts, the main sub-assemblies were firstly selected, by considering their impact on the total mass of the assembly and their potential critical behaviour during transportation.

FE Model Implementation

Most of the geometries have a small thickness, thus they were modelled as surface bodies exploiting shell elements. Shapes, material properties and shell thicknesses for all the components were obtained from the 3D model and drawings of the assembly. The selected parts (which are shown in Fig. 1) were the external vessel, the thermal shield, the strong back, the cavities, the support posts and the solenoids.

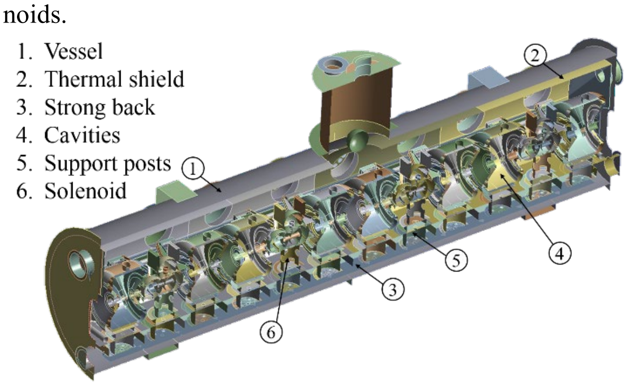

Figure 1: Main subassembly of the FE model.

The bellows and the two-phase pipe were excluded from this preliminary FE model because their geometry would require a huge number of elements to achieve reliable results, which would lead to extremely long computational time. These parts, which indeed may be critical during transportation, will be considered in later analysis through MB modelling. Each of the six cited sub-assemblies was firstly modelled separately for simplicity, and all the sub-

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assemblies were finally imported in the full assembly to produce the complete FE model. All the connections between parts were setup through node merging and fixed joints, to completely avoid contact regions. The final model had a total mass of about 6200 kg (i.e. about 75% of the total estimated mass of the CM), having a total number of 1.1 e6 nodes and 1.2 e6 elements. The modal analysis was setup, requiring about 5 hours to compute the first fifty modes of the CM. A section view of the full model and an example of a computed (high frequency) mode are reported in Fig. 2. A list of the main modes is reported in the following sections.

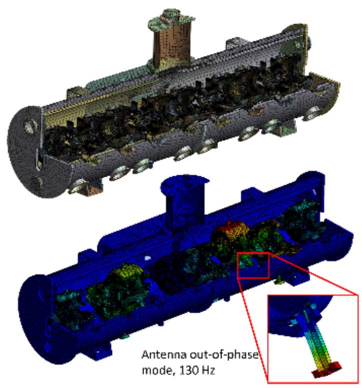

Figure 2: Example result of the full assembly.

Bellows FE Modelling

As stated, the presented FE model does not account for potential critical parts, such as the bellows which allows for the thermal deflection compensation during operation. On the other hand, they were considered in the MB model (see below) through six degrees-of-freedom lumped stiffnesses, i.e. through a six-by-six stiffness matrix. In order to compute the values of this matrix, a dedicated FE model was setup for each bellow. The model was meshed with shell elements, since the thickness of all the bellows is much lower than their radius. Figure 3 shows the load scheme applied to the bellows.

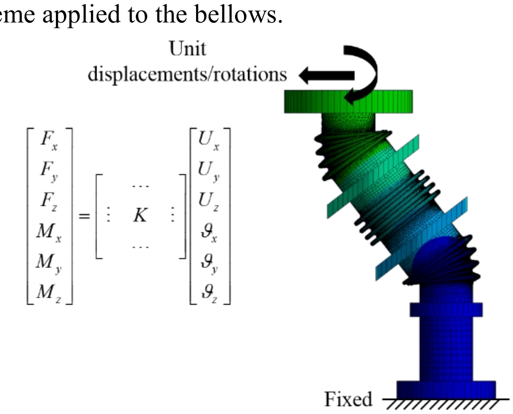

Figure 3: Equivalent stiffness matrix computation.

One end of the bellow was fixed, while a unit displacement along the six degrees-of-freedom (i.e. 3 translations and 3 rotations) was sequentially applied to the other end. The equivalent stiffness matrix of the bellow could then be computed by measuring the reaction forces and moments corresponding to each loading condition: a column of the matrix was obtained by considering each of the six loading conditions. It is worth noting that this model was also exploited to assess the natural frequencies of the bellows themselves, to check that they were significantly higher than the mechanical loading they will be subject to during transportation. This allowed to neglect their inertia effects.

MULTIBODY ANALYSIS

A multibody model aimed at computing the natural frequencies of the whole CM and, subsequently, to assess the TT filtering capability was developed. The model, shown in Fig. 4, was made up of different parts, modelled through different implementation strategies:

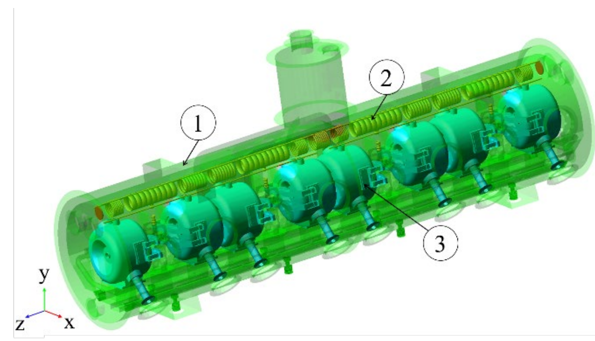

Figure 4: Multibody model of the CM.

- The CM vessel, no. 1, together with the thermal shield, the strongback, the support posts and the solenoids were imported as a modal neutral file from FE software, which allows to consider the deformability of the whole structures through the Craig-Bampton modal reduction technique [4]. By using this technique, the degrees of freedom of the model were reduced from about 800000 to about 330.
- The two-phase pipe, no. 2, was modelled splitting the pipe in several beam elements whose interaction was determined through the Timoshenko beam theory.
- The cavities, no. 3, were imported as rigid bodies, considering their actual mass and inertia properties. The assumption of considering them as rigid bodies is justified by the FE modal analysis of the cavities themselves, whose natural frequency were found to be much higher than the ones which are of interest during the transportation (up to 20-30 Hz).
- The bellows, which are not represented in Fig. 4, were modelled as lumped stiffness, whose characteristics (equivalent stiffness matrix) were derived from FE analysis.

MB Model Validation

To check the accuracy of the imported Craig-Bampton reduced model, a first modal analysis was performed on the sole CM Vessel and the Thermal Shield (this configuration in the following is referred as *V+TS*). The computed natural frequencies were compared with the ones obtained through a full model analysis in the FE software. Subsequently, the two-phase pipe, the cavities and the bellows were added, and the natural frequencies of the whole CM

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were computed in MB environment. Table 1 shows the results of modal analysis for FE and MB models. Frequencies up to about 20 Hz are considered because, due to the low truck speed on the road, the input amplitude due to the road roughness related to higher frequencies is considered negligible.

Table 1: Natural Frequencies of the FE and MB Models

Mode no.	f^{FE}_{V+TS} (Hz)	f^{MB}_{V+TS} (Hz)	f^{MB}_{CM} (Hz)
1	9.5	9.5	9.6
2	15.4	15.3	15.2
3	17.2	17.1	17.9
4	19.1	18.8	21.6
5	21.2	20.8	23.1

Concerning the modal reduction procedure, the comparison between f^{FE}_{V+TS} and f^{MB}_{V+TS} confirms the accuracy of the reduction and importation procedure, because the differences in terms of natural frequency of the first five natural modes never exceed 2%. In addition, a check of the mode shapes was also performed, confirming the matching between the two models.

The natural frequencies of the complete model (f^{MB}_{CM}) do not substantially move away from the ones related to the sole vessel and thermal shield, being the difference always below 15% (for the first five natural modes). Considering the modal shapes, they slightly change, with respect to the *V+TS* analysis, due to the contribution of the mass and inertia of the rigid cavities. However, adding the two-phase pipe, the bellows and the cavities does not introduce new low frequency natural modes which may be dangerous during the transportation.

Figures 5 and 6 shows the modal shape of the first and second natural modes respectively. The first mode involves the oscillation along the longitudinal (z) axis of the thermal shield, being the other parts of the CM stationary. This mode is not considered particularly critical since it does not encompass relevant deformation of the bellows, which are the most critical parts of the CM. On the contrary, the second natural mode is related to the lateral bending of the two-phase pipe and of the cavities string and strongly involves bellows deformation. For this reason, this natural mode, among the low-frequency ones, is assumed to be the most critical.

SHOCK ABSORBER DESIGN

A first estimate of the Shock Absorber (SA) stiffness was performed considering only the vertical motion on the CM (heave mode). The SA stiffness influences both the vertical travel of the CM and the mechanical filtering effect; in particular, the higher the stiffness is, the lower the CM travel and the filtering effect result. Given as constraint a reference vertical travel $\delta_{st} = 25$ mm, related to layout constraints, the natural frequency of the heave mode of the CM together with the TT was computed as follows

$$f_{Heave} = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_{st}}} = 3.1 \text{ Hz}$$

where g is the gravity acceleration. Being f_{Heave} sufficiently lower than the natural frequency of the first and second natural modes computed in the previous section, the mechanical filtering effect of this first attempt configuration should be appreciable. The SA total vertical stiffness k_v^{tot} necessary to obtain this frequency was obtained basing on

$$k_v^{tot} = 4m\pi^2 f_{Heave}^2 = 3600 \text{ N/mm}$$

where m is the sum of the masses of the CM and the TT.

Basing on the first attempt sizing, a draft of the TT was created (Fig. 7).

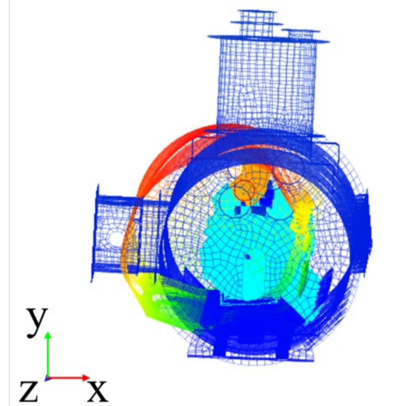

Figure 5: Modal shape of the first natural mode.

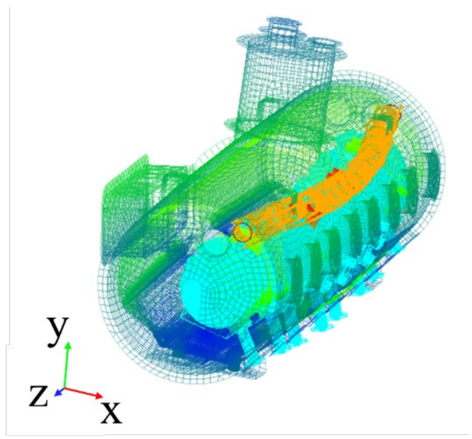

Figure 6: Modal shape of the second natural mode.

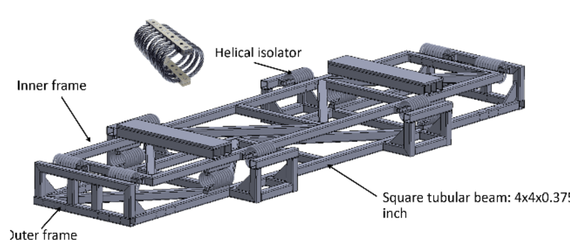

Figure 7: Drawing of the TT with helical isolators.

The TT is made up of an outer frame, rigidly fixed to the loading bed of the truck, and of an inner frame, flanged to the CM supports. The inner and the outer frame are con-

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nected through 12 helical isolators (HI), whose characteristics were selected starting from the first attempt choice of the vertical stiffness [5]. The linearized characteristics of the helical isolators were implemented in MB environment through bushing element which exploit three-axial force (F_x, F_y, F_z) due to three-axial displacement (δ_x, δ_y, δ_z)

$$\begin{pmatrix} F_x \\ F_y \\ F_z \end{pmatrix} = \begin{pmatrix} k_x & 0 & 0 \\ 0 & k_y & 0 \\ 0 & 0 & k_z \end{pmatrix} \begin{pmatrix} \delta_x \\ \delta_y \\ \delta_z \end{pmatrix}$$

where k_x, k_y and k_z is the spring stiffness along three directions.

The natural modes of the CM together with TT were computed in MB environment and are listed in Table 2.

Table 2: Natural Frequencies of the CM+TT

Mode no.	Description	*f* (Hz)
1	Roll	1.9
2	Pitch	3.0
3	Heave	3.3
4	Yaw	4.1
5	Comb. 1	4.5
6	Comb. 2	5.3

The first 4 mode shapes are simpler and can be directly related to the conventional mode shapes of vehicles. On the other hand, the fifth and sixth modes showed more complex shapes, which were a combination of the previous. The natural frequencies are included in the range 1.9-5.3 Hz, sufficiently lower with respect to the first and second natural modes of the CM. It was also verified that the first and second natural modes of the CM itself do not appreciably change due to the introduction of the TT and helical isolators.

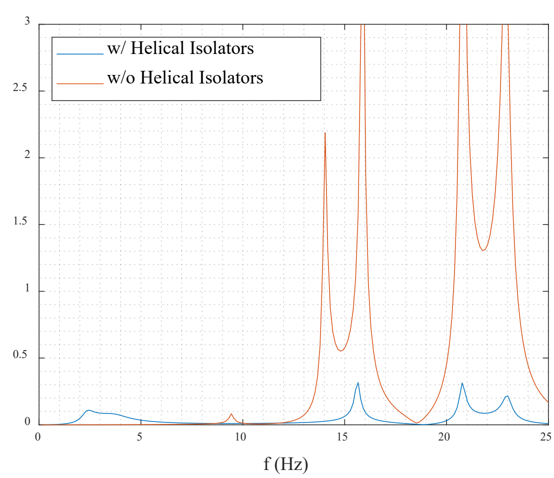

Figure 8: FRF of the vertical displacement of the reference cavity – With and without helical isolators.

In order to evaluate quantitatively the mechanical filtering effect of the TT with helical isolator, a harmonic analysis was performed, imposing three different inputs to the loading bed of the truck: heave, roll and pitch. For each input, the vertical (y-axis) and lateral (x-axis) displacement of the centre of mass of a reference cavity were measured and plotted as frequency response function (FRF), normalized with respect to the input magnitude.

For the sake of readability, only the results related to the heave input are presented. Figures 8 and 9 show the FRF of the vertical and lateral displacement, respectively, of the centre of mass of the reference cavity normalized with respect to the heave input amplitude, both obtained with and without helical isolators.

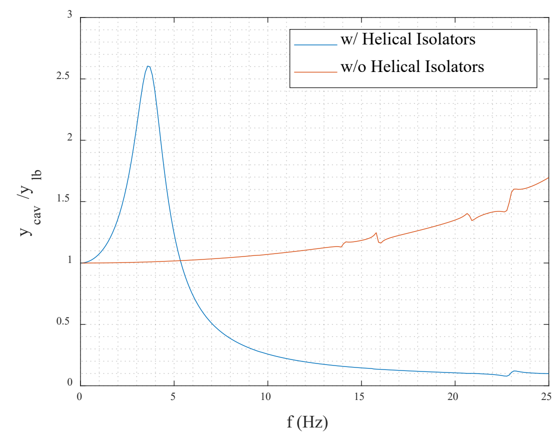

Figure 9: FRF of the lateral displacement of the reference cavity – With and without helical isolators.

Considering the vertical (y) displacement of the cavity, due to the introduction of the HI, it is magnified at low frequency of about 2.5 times (for $f \approx 3$ Hz), due to the resonance of the heave mode. However, this is not critical because no deformable body modes are interested around this frequency. This means that the response is characterized by a rigid body motion of the CM, and all the displacement is absorbed by the HI. For higher frequency values (i.e. $f >$ 5 Hz) the vertical input is reduced (magnitude < 1 in Figure 7) due to the mechanical filter effect. In particular, considering the frequency values corresponding to the first and second CM natural mode frequencies, the vertical input are reduced by a factor of 4 and 5 respectively.

Considering the lateral (x) displacement of the reference cavity, in the case of no-spring TT configuration, this is very high due to the resonance of the second natural mode. This effect could be very critical for the bellows from a structural point of view. However, if the HI are used (solid lines), the lateral displacement is strongly reduced. Similar consideration can be inferred for higher frequency modes and for pitch and roll inputs.

ROAD LOADING SIMULATION

In order to assess the loads borne by the CM components during transportation with or without the helical isolators, a dynamic simulation was performed. The truck loading bed displacement (heave, roll and pitch) were imported in the model from the data which were recorded during the transportation of a similar CM. A reference record of one

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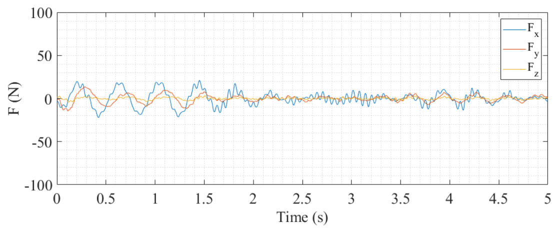
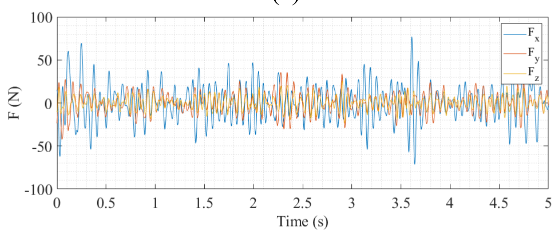
minute, including the largest measured displacement, was considered.

The forces acting on each component were then recorded during the simulations and a particular attention was devoted to the forces acting on the bellows. Figure 10 shows the x, y and z components of the force acting on a reference bellow considering the TT equipped with helical isolators (Fig. 10a) and without helical isolators (Fig. 10b).



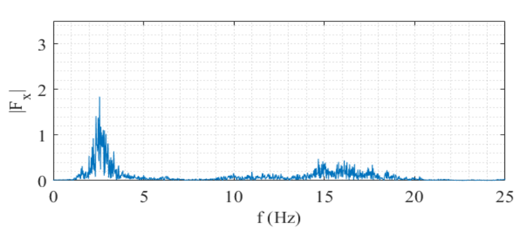
(a)



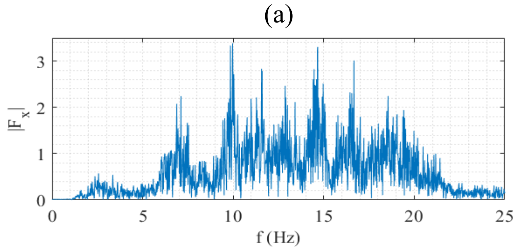
(b)

Figure 10: Force on a reference bellows – a) TT equipped with helical isolators; b) rigid TT.

A strong reduction of the oscillation amplitude due the mechanical filtering effect is appreciable using the helical isolators. In particular, if the spectral analysis of the force acting on the reference bellow is considered (only the x-component is presented for the sake of conciseness, since the results along the other directions are similar), the filtering effect can be directly appreciated. Indeed, as shown in Fig. 11, if rigid TT layout is considered (Fig. 11b), the force contribution in all directions due to 10-20 Hz frequency is much higher than the case where HI are used (Fig. 11a).



(a)



(b)

Figure 11: x-component of the force on a reference bellows. a) TT equipped with helical isolators; b) rigid TT.

CONCLUSION

The described activity allowed to design a reliable Transportation Tool for the Cryomodule SSR1. The design workflow was structured using a multi-level approach. A Finite Element modelling of the main sub-assemblies of the Cryomodule was setup to assess the dominant natural frequencies. This preliminary analysis provided a rough estimation of the needed cut-off frequency of the transportation tool. The FE model was then imported into a Multi-Body model to provide a preliminary design of the suspension system, in terms of spring placing and stiffness. These guidelines were exploited to select commercial isolators and to define the frame geometry. A time series of displacement profiles recorded during previous transportation of a similar CM was then imported in the MB model. This allowed to assess the TT performances and, also, to estimate the loading acting on the critical components (e.g. bellows, support post).

However, an experimental campaign will be essential to fully validate the numerical models to improve their reliability for future uses. The experimental validation of the design procedure will be obtained in future developments through some trial transportation of the Cryomodule onsite at Fermilab. Several accelerometers will be placed at critical locations of the structure to check the reliability of the models' predictions.

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