# Cryogenic Vertical Test Stand Dynamic Model and Experimental Validation for Vibration Suppression

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**Abstract.** High Q-factor superconducting radio-frequency (SRF) cavities are extremely sensitive to mechanical vibrations, which can cause a shift in the electromagnetic resonance frequency, degrading the cavity performance. To avoid this phenomenon (i.e. microphonics) specifically designed vibration suppression systems are needed. In this paper, the numerical model of the Vertical Test Stand (VTS) insert adopted at the Fermi National Accelerator Laboratory was studied, in the framework of the experimental characterization of Dark SRF cavities conducted by the SQMS (Superconducting Quantum Materials and Systems) center. Vibration testing was performed on the actual system to validate and tune the model, and a good correlation between numerical and experimental results was found. Additionally, two different setups were compared, with and without implementing a preliminary passive vibration suppression system, to assess its effectiveness in reducing the microphonics phenomenon. The obtained results allowed for the design of an improved isolation system, which will be the subject of future experimental testing (FERMILAB-CONF-24-0134-SQMS-TD).

**Keywords:** Vibration suppression, Numerical model, Superconducting cavities.

## 1 Introduction

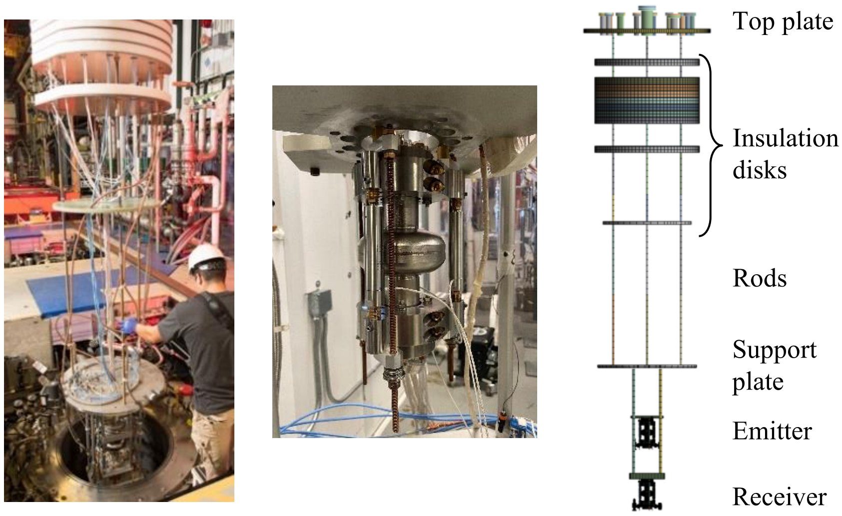
Developing and testing superconducting radio-frequency (SRF) cavities require extremely low temperatures, in the range of 2-4 K [1]. This ensures the material reaches superconducting properties, and drastically enhances the quality factor of the cavity itself (in the order of $10^{10}$ - $10^{11}$), thus leading to high electro-magnetic resonance responses. On the other hand, such high quality-factors result in extremely sharp resonance peaks of the cavities. Depending on the application, even small changes in the resonance frequency can cause dramatic performance degradation. For experiments such as Dark SRF [2], or the heterodyne axion dark matter search [3,4] in preparation at Fermilab, frequency control and stability are of utmost importance. For such experiments, the combination of tuner systems to actively control the cavity frequency, and passive methods to suppress the effect of external sources of vibration, should enable optimal sensitivity. It is worth noting that the actual resonance frequency of the cavities directly depends on the geometrical shape of the RF volume, i.e. the vacuum volume inside the cavity. Generally, vibrations represent an uncontrolled source of mechanical deformation in operation, due to ground motion or moving components (such as pumps or valves) [5]. This phenomenon, called microphonics, is generally mitigated with passive [6] and active [7] vibration suppression methods. While the former act as mechanical filters to reduce the vibration amplitude experienced by the cavity, the latter requires a dynamic actuator on the cavity that can compensate for the mechanical vibrations. Both approaches work in synergy since passive isolation is mandatory to enable active control to mitigate the vibrations effectively.

In this paper, the effect of microphonics is investigated in the case of the Vertical Test Stand (VTS) insert adopted at Fermi National Accelerator Laboratory for cold tests, in the framework of the Dark SRF 2.6GHz experiment, in cooperation with the SQMS (Superconducting Quantum Materials and Systems) center. In this research field, passive suspension systems are generally designed through highly simplified lumped models. On the other hand, this paper introduces an innovative methodology based on a complete FE model of the assembly. A numerical model of the VTS was developed to obtain a reliable simulation of the system response subject to ground excitation. Experimental vibration measurements were used to update and validate the model both in terms of mechanical response and cavity detuning. The model was then used to simulate different working conditions, with and without the use of a preliminary passive suspension system. The developed model was used to develop an enhanced suspension system, to improve the isolation performances. The testing of the designed solution will be the subject of future research activities.

## 2 Experimental dataset

The VTS setup is designed to hold the tested cavity in a liquid helium bath. To this extent, a main metal disk (top plate, 1.09 m diameter) is bolted to the laboratory ground, hosting three threaded rods which are screwed to the top plate. Across the rods, other discs are mounted to act as thermal isolators. At the end of the rods, another metal disc is connected, to be held right below the liquid helium level during testing. This support plate presents several holes to connect different testing setups for different cavities. In the case of the Dark SRF experiment [2], two cavities are needed, namely the emitter and the receiver. These are mounted onto two separate plates, both of them independently connected to the support plate with four rods each. While microphonics could impact both cavities' performance, in this study only the emitter behavior was considered since the receiver cavity input and output RF lines were not available for a portion of the cold tests. Both emitter and receiver have the same geometrical shape and frequency (2.6 GHz), and they are mounted into a frame structure that presents three threaded rods, and three piezoelectric actuators. Figure 1(a) shows the overall structure of the VTS while Fig. 1(b) shows a close view of the emitter sub-assembly.

The experimental activity was conducted at Fermilab during cold test operations of the Dark SRF 2.6 GHz setup. In particular, two sources of data were used: mechanical vibration of the top plate measured through a seismometer and the emitter cavity detuning measured through a phase noise analyzer.




(a) (b) (c)

**Fig. 1.** Dark SRF experiment installed on the VTS insert: (a) overall view of the test setup, (b) detail of the emitter assembly and (c) numerical model schematic.

The seismometer measures vibration velocity along the vertical direction, and is acquired with the National Instruments (NI) DAQ NI USB-9162 and ADC NI-9239. This data can be directly compared with the numerical model results. On the other hand, cavity detuning measures the instantaneous RF resonance frequency of the cavity and its oscillation around the mean value. Thus, it is not directly related to mechanical displacements for model validation. Nevertheless, a direct relation between the cavity's overall length (i.e. distance between the two cavity flanges) and the detuning can be assumed. Thus, it is possible to compare the measured detuning with the numerical cavity's length variation. It is worth noting that the compared quantities are correlated but physically different, thus only qualitative considerations can be drawn in terms of peaks in the frequency domain. Two different loading sources were considered in the tests. In the case of seismometers, an impulse hammer excitation was conducted by hitting the center of the top plate with an impulse hammer (PCB Piezotronics Model 086D20). On the other hand, ground excitation was sufficient to drive microphonics, thus no additional mechanical excitation was needed to measure a relevant detuning. For detuning measurements, the cavity was excited to a low accelerating field so that its resonant frequency could be tracked.

# 3 Numerical model

The VTS is a complex assembly composed of hundreds of components. To obtain a manageable numerical model, geometrical simplifications were mandatory. To this extent, minor geometrical details such as small holes, shoulders, fillets and chamfers were removed from the CAD model. Additionally, most of the assembly is composed of disks and rods, which were modeled with shell and beam elements respectively. This drastically reduced the overall number of nodes, thus decreasing the computational time. Additionally, all the bolted connections were assumed to be rigid. The complete model was divided into sub-assemblies, which were separately modeled to simplify the model setup. All the subassemblies were then imported into the final model of the full VTS assembly. A schematic view of the model organization is reported in Fig. 1(c). The same model was used to perform modal analysis and harmonic response analysis, by exploiting the mode superposition method. The same boundary conditions were used throughout the analysis. The excitation of the harmonic response analysis was imposed by a nodal force applied to the top plate, to mimic the hammer excitation, and by the base excitation option applied to the top plate constraints to represent ground vibration. The main unknown parameter in the model is represented by the damping ratio, thus the test results were used to infer a realistic value which was then applied as a constant structural damping to the whole model.

The final aim of the model is to evaluate the performances of different suspension systems on microphonics. The damping ratio of the model was set by exploiting the experimental data measured with the seismometers. Subsequently, the same damping ratio was used for the detuning evaluation, since the physical system remains the same for all the testing conditions. Even if the damping ratio is influenced by material properties and, mainly, connection conditions, characterizing all the local damping sources would not be feasible due to the assembly complexity. Thus, a global damping value was estimated starting from experimental data through the half-power bandwidth method [8]. The hammer test was performed on the VTS once mounted in its pit. The Frequency Response Function (FRF) between hammer input and seismometer output was computed, averaging ten repeated measurements. The results are reported in Fig. 2(a), where the obtained FRF is plotted in the frequency domain. The peak at 344 Hz was selected for the computation, as shown in the zoomed view in Fig. 2(a). The obtained damping ratio was 0.02, which was kept constant in all the simulations. The harmonic response analysis simulating the hammer test was then performed by imposing different possible boundary conditions on the top flange. Three different configurations were investigated: fixed support to the external area of the flange (namely FEM-1), fixed support to the line corresponding to the bolted connection (namely FEM-2) and fixed support to the external line (namely FEM-3). The constrained regions are shown in Fig. 2(c), (d) and (e) respectively. FEM-1 relies on the hypothesis that the whole region in contact with the support is rigidly constrained. FEM-2 is based on the hypothesis that only the mounting hole diameter is constrained, but still allows for local plate rotation. FEM-3 is based on the hypothesis that the external edge of the plate is constrained. This latter scenario was investigated because the top plate is not directly bolted to the ground, but it is bolted to a flange of unknown stiffness instead. The obtained results were compared with the experimental response of the top flange, to assess which boundary condition better represents the experimental scenario.

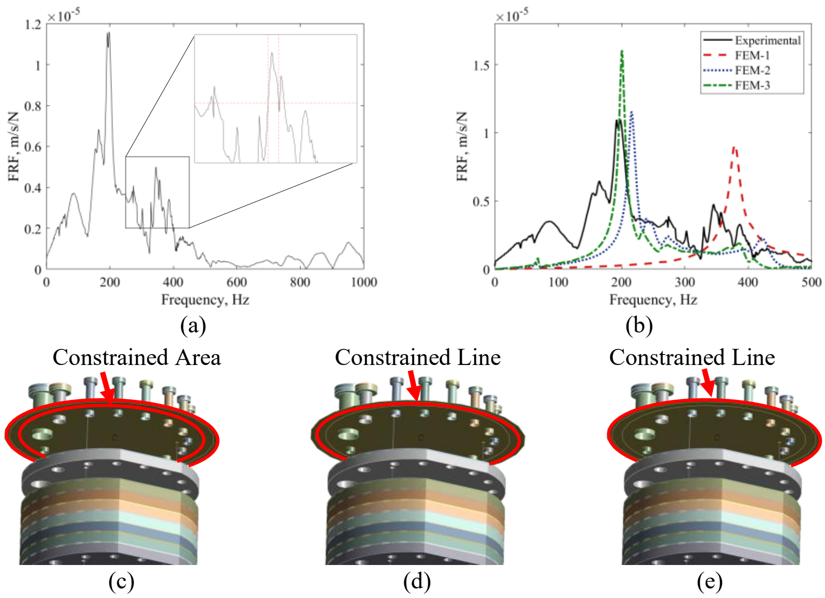



**Fig. 2.** Top flange FRFs: (a) experimental and (b) numerical results. FEM model boundary conditions: (c) flange area (FEM-1), (d) bolted connection (FEM-2) and (e) external line (FEM-3).

As shown in Fig. 2(b), FEM-1 (red dashed line) is unrealistic, since the FRF only shows one peak in the high-frequency range, demonstrating that this boundary condition is too stiff with respect to experimental results (black solid line). FEM-2 (blue dotted line) moves the main peak close to 220 Hz, but the model is still too stiff. Finally, FEM-3 (green dash-dotted line) shows the best correlation with experimental results in terms of resonance frequency, which is 192 Hz and 202 Hz for experimental and numerical data, respectively. It is worth noting that the model still results in being slightly stiffer than the physical assembly. This can be ascribed to the fact that several simplifications were needed to obtain a reliable model, in terms of geometry removal (details, holes, slots, shoulders) and components removal (screws, dice, hose, electronics). This simplification results in an overall mass underestimation, which can justify the higher resonance frequencies.

# 4 Vibration suppression performances

## 4.1 Preliminary vibration suppression system

Experimental testing was performed in the case of a rigid connection between the support plate and the emitter, thus representing a baseline of microphonics (i.e., no isolation at all is provided). The microphonics due to ground vibration in operation were recorded. The results are presented in terms of the detuning spectrogram in Fig. 3: the horizontal axis represents time, the vertical axis represents the frequency and the color is proportional to the detuning amplitude (in logarithmic scale).

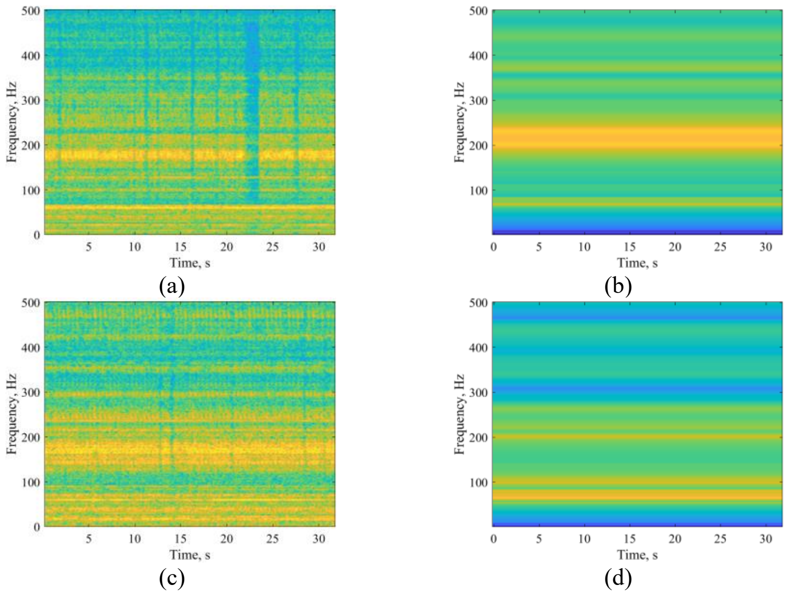



**Fig. 3.** Microphonics in operation. Without vibration isolation: (a) experimental and (b) numerical results. With the preliminary suppression system: (c) experimental and (d) numerical results.

The color scale was omitted because, as stated, numerical and experimental data do not refer to the same physical quantity, since experimental results refer to frequency detuning (Hz), while numerical results refer to the distance between cavity flanges (mm). A coherent color scale was used for all the experimental data (i.e. Fig. 3(a) and Fig. 3(c)) and the numerical data (i.e. Fig. 3(b), Fig. 3(d) and Fig. 4(b)), for comparison purposes. Lower values are represented by blue colors, while higher response levels are depicted with yellow colors. Figures 3(a) and (b) show that the numerical results are well aligned with the experimental data, showing a mode family close to 70-80 Hz, and a higher frequency mode family close to 200-220 Hz (about 10% higher than experimental data). To mitigate the microphonics, a preliminary vibration suppression system was tested by introducing compliant wire mesh washers [2] between the support plate and the emitter rods. These components are compact and cheap, but their highly non-linear behavior and lack of information from the manufacturer result in unreliable modeling. It is well known that the dynamic equivalent stiffness of the isolator is much higher than the static stiffness [9]. In this particular case, the isolators were chosen based on the static stiffness declared by the manufacturer but, as shown in Fig. 3(c), they resulted in being ineffective. Indeed, the detuning intensity in Fig. 3(c) is comparable with the results in Fig. 3(a), demonstrating that the filtering effect of the washers is not significant. To reproduce this behavior, the numerical model was updated by adding a bushing joint between the support plate and the emitter rods. The stiffness of the bushing was set equal to the static stiffness of the washers multiplied by a factor of four. This factor was chosen based on the previous experience in [9]. The results are shown in Fig. 3(d), demonstrating a reasonable correlation between numerical and experimental data, thus confirming that the actual washer stiffness is too high to provide an effective suspension of the emitter and, thus, relevant vibration suppression.

### 4.2 Enhanced vibration suppression system

Since the main drawback of the preliminary suspension system was determined by the high dynamic stiffness of the wire mesh washers, an improved version of the suspension system was designed by selecting linear helical springs. In particular, the stiffness of the springs was chosen to determine the first natural mode of the spring-emitter system of 20 Hz. The system requires two springs per rod, one above the support plate and one below, to avoid any direct contact between the support plate and the rods. The detail of this connection is reported in Fig. 4(a), while Fig. 4(b) reports the results of the corresponding numerical model (the springs were modeled through beam elements).

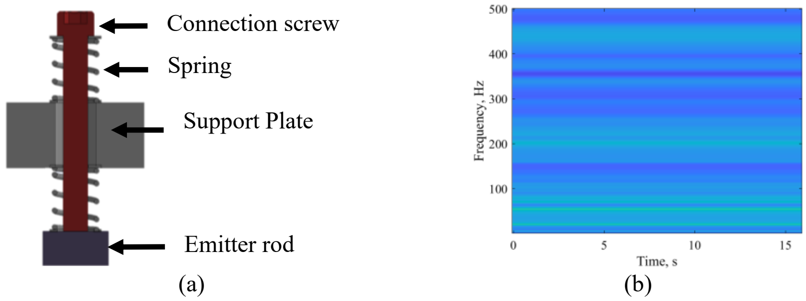



**Fig. 4.** Enhanced vibration suppression system: (a) CAD model and (b) numerical results.

Figure 4(b) highlights that the expected response amplitude of the cavity is much lower than the results shown in Fig. 3(b) and (d), demonstrating that the proposed suppression system should provide a relevant filtering effect, thus significantly reducing the microphonics phenomenon. This enhanced system is under development at Fermilab and will be tested in future activities to further validate the numerical model.

## 5 Discussion and Conclusions

In this paper the Vertical Test Stand (VTS) insert adopted at Fermi National Accelerator Laboratory for the Dark SRF cavities cold tests was analyzed. A Finite Element model was developed to predict the dynamic behavior of the assembly, aiming at microphonics mitigation. To this extent, the main components of the full assembly were modeled, and the main parameters (damping ratio and boundary conditions) were set through comparison with experimental hammer test data. This approach represents a novel methodology in this field, since generally simplified lumped models are used to design the vibration suppression system. The updated model was then used to simulate the effect of different connections between the support plate and the emitter rods, i.e. rigid connection (no isolation) and preliminary suppression system based on wire mesh washers. The results demonstrated a good correlation between numerical and experimental data, both under the effect of the hammer test and ground vibration during operation. Given the good performance of the model in predicting the cavity response, it was used to design an improved suppression system based on linear springs, which should drastically reduce the microphonics in operation. The experimental validation of the improved system will be the subject of future research activities.

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