MODAL SURVEY OF MEDIUM ENERGY SUPERCONDUCTING RADIO FREQUENCY CAVITY FOR ACCELERATOR PRODUCTION OF TRITIUM PROJECT *

G.W. Ellis#, B.G. Smith, LANL, ESA-DE, MS H821, Los Alamos, NM 87545, USA

Abstract

We conducted a modal survey of a medium energy (β=0.64) SCRF (Superconducting Radio Frequency) 5-Cell cavity to experimentally determine its natural frequencies and mode shapes. We compared the results of this testing to finite element analysis performed using Cosmos [1]. The experimental results were in good agreement with the numerical results, with the worst case mode differing by approximately 5%. This work was part of a larger effort to accurately predict axial and transverse cavity deformations, and therefore RF frequency shifts, of these medium energy cavities due to facility ambient noise (mechanical vibration) under final operating conditions.

1 INTRODUCTION

A 64% β 5-Cell niobium superconducting cavity being developed at Los Alamos as part of the APT (Accelerator Production of Tritium) program will be used to accelerate a proton beam to ~1050 MeV. The cavities are housed in liquid helium vessels, which keep them at an operating temperature of 2 K. The liquid helium vessels are mounted inside vacuum vessels, keeping them isolated from ambient thermal conditions, which are mounted to the facility floor. See Figures 1 and 2. The cavities are designed to resonate electromagnetically at 700 MHz, but this frequency can be shifted by stretching or compressing the cavity, due to its bellows-like geometry, thereby changing the cell shape slightly.

Although the ability to adjust the frequency is necessary and desirable, cavity shape change as a result of dynamic motion from mechanical vibratory noise (microphonics) can result in unwanted and undesirable shifts in the RF resonant frequency and associated electromagnetic fields, which can affect the acceleration and particle beam dynamics. Preliminary estimation indicates that the RF frequency shift due to microphonics cannot be more than +/- 100 Hz [2]. This estimation was derived assuming a 1% power margin.

Because of the complexity of the cavity assembly (the cavity, helium vessel and vacuum vessel), it is difficult to determine with a high degree of confidence how the cavity will respond to ambient noise sources. Significant accuracy is lost in modeling boundary conditions, choosing structural damping values and making other such approximations that are necessary when dealing with large, complex FE models. For structural analysis these uncertainties can be mitigated by assigning large safety factors, which cannot be done in vibration analysis.

Two things can reduce the possibility of large amplitude resonance of a structure: designing away from problem input frequencies, and isolation or damping of the structure. Both of these require accurate knowledge of a structure's natural frequencies and mode shapes to

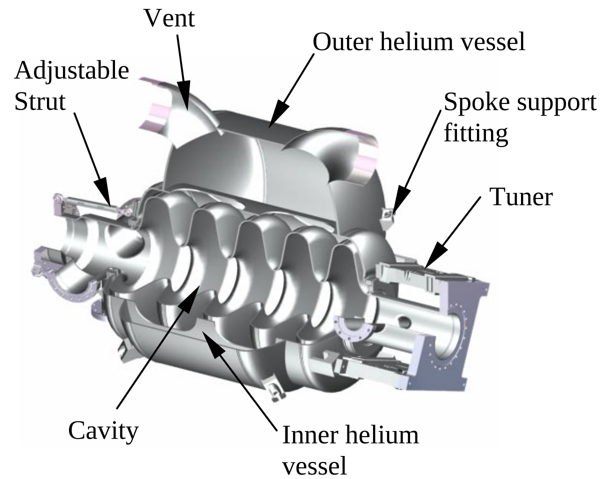

Figure 1: Cut-away of cavity installed in helium vessel

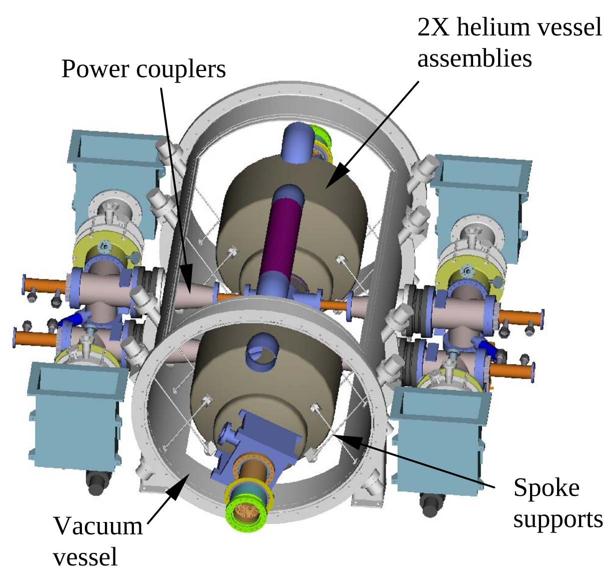

Figure 2: Complete cryomodule assembly

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# E-mail: gellis@lanl.gov

determine what must be done to prevent unacceptable resonance.

To inspire more confidence in dynamic modeling results, a modal survey can be performed to validate numerical results. In a modal survey, all frequencies over a bandwidth of interest are excited in a structure and resonances are identified using spectrum analysis software. Frequency response functions, corresponding to excitation and response points, are used by modal analysis software to animate the mode shapes of the structure.

In this effort, we developed a three-stage plan to be able to accurately predict the mechanical response of the cavity, installed in the helium vessel and cryomodule, due to the input of a given facility power spectral density curve. The three steps of the plan are as follows: 1.) Identify mechanical natural frequencies and associated mode shapes of a prototype niobium cavity under free-free boundary conditions. Compare numerical results using Cosmos FEA and experimental results using SigLab dynamic signal analyzer and StarModal modal analysis software. 2.) Increase the complexity of the structure. Repeat step 1. using a CERCA 5-Cell 64% β cavity with bulkheads, struts, and dummy tuners installed. Compare numerical and experimental results. 3.) Perform modal survey of a cryomodule mockup complete with spoke supports, power couplers installed, and cavity and helium vessel assembly installed in vacuum vessel.

Completion of step three will give a high degree of confidence in the accuracy of the finite element model at that point. That model can then be used with the damping values calculated from performing the modal survey and specific facility input spectra to determine maximum cavity deflections and associated shapes. The maximum RF frequency shifts resulting from the cavity deformation can then be calculated.

Stages 1 and 2 of the above plan have been completed and the results of that work are presented in this report.

2 MODAL ANALYSIS

Modal analysis requires calibrated measurements of input force and output motion (displacement, velocity, or acceleration) at points on a structure. The force input could be a calibrated hammer or a dynamic shaker, and the output can be measured by an accelerometer. The number of measurement points required on the structure is determined by the complexity of the mode shape that is desired. For example, in Figure 3a, a single measurement point on the structure is adequate to capture the fundamental bending mode, but three points are required to capture the third harmonic in Figure 3b.

There are primarily two ways to perform a modal survey. One way is to have a single excitation point and multiple response points, and the other is to have a single response point and multiple excitation points. A complex frequency response function is obtained between the driving point degree of freedom (the single excitation or response point) and every other point on the structure. The frequency response function (FRF) is the ratio of

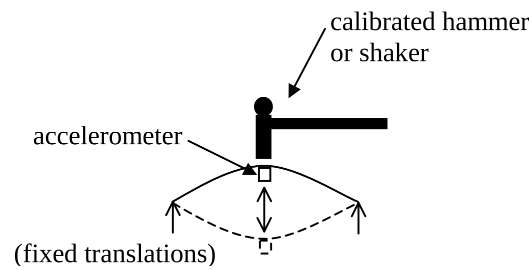


Figure 3a: Single measurement point necessary to capture mode shape

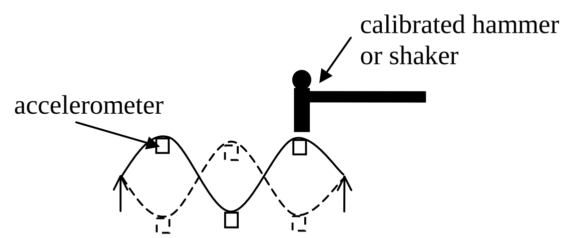


Figure 3b: Multiple measurement points necessary to capture mode shapes

the output at point x to the input at point y as a function of frequency, defined as follows [3]:

$$H_{xy}(f) = \frac{F_y(f)}{F_x(f)}$$

where

$H_{xy}(f)$ is the FRF from x to y,
$F_y(f)$ is the Fourier transform of the output signal $f_y(t)$ measured at point y, and
$F_x(f)$ is the Fourier transform of the input signal $f_x(t)$ measured at point x.

The FRF is complex, meaning it contains phase information as well as amplitude information.

Theoretically, one set (i.e., one driving point and multiple response points) of frequency response functions is sufficient to determine the natural frequencies and mode shapes of a structure, given that there are enough points on the structure to capture the most complex mode of interest. However, when dealing with structures that have mode shapes in three-dimensional space, the participation of a mode with respect to the direction of input becomes an issue. Generally, if a structure is excited in the x direction, modes that move in the x direction will have a greater participation than modes that move in the y and z directions. This means that the x direction modes will be excited more efficiently than the y and z direction modes. Therefore, it is advantageous to use multiple sets of frequency response functions when doing a modal survey so that each mode has a strong participation with respect to at least one of the excitation directions. This minimizes the chance of missing a mode.

For modal testing, a frequency domain model of the structure is used in conjunction with FRF measurements to determine the modal parameters (mass, stiffness, damping, mode shape) of the structure [4]. Figure 4 shows a frequency response function in StarModal.

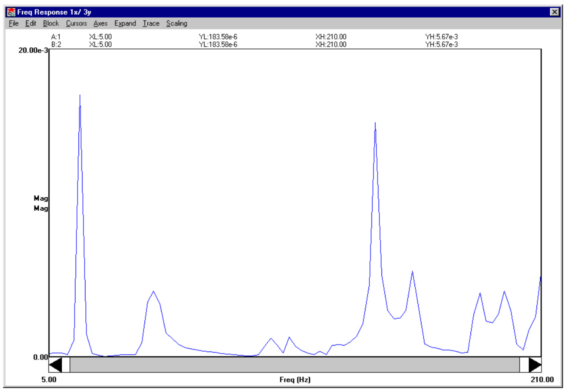


Figure 4: Frequency response function in StarModal

Figure 5: Test Setup for Stage 1

3 STAGE 1 TEST SET-UP AND RESULTS

For Stage 1 testing, a prototype niobium 5-cell cavity was suspended from a test stand using bungee cords to simulate free-free boundary conditions. The weight of this cavity was ~130 lb. A 25-lb Bruel & Kjaer shaker was used to excite the cavity in the axial and transverse directions at a point on the endflange (see Figure 5). Frequency response functions were measured using an Endevco ISOTRON tri-axial accelerometer at 28 points on the cavity, which was determined to be sufficient to capture the mode shapes in the 5-200 Hz range.

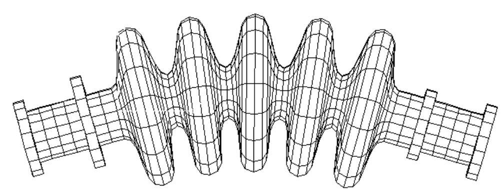
A three-dimensional Cosmos model of the cavity was built using ~1400 second order shell elements. The RF coupler ports and their endflanges were modeled as ring masses around the beam tube extensions at the proper longitudinal positions. Figures 6 and 7 show the first bending mode of the cavity calculated by Cosmos and measured experimentally using SigLab and StarModal. Table 1 shows the results of this stage of testing.

Modal damping in Table 1 is expressed as a percent of critical damping, according to the following equation [5],

$$\%\text{critical} = 100 * \frac{\sigma_k}{\sqrt{(\sigma_k^2 + \omega_k^2)}}$$

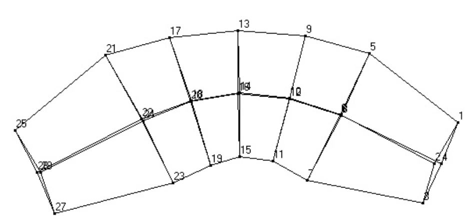
where

σ_k = damping value in Hz for mode k,
ω_k = damped natural frequency in Hz for mode k.



Mode 1 = 25.45 Hz

Figure 6: First bending mode, Cosmos model



Mode 1 = 24.94 Hz
Damping = 0.605%

Figure 7: First bending mode, StarModal animation

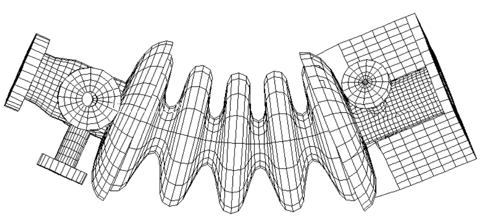
Table 1: Stage 1 Results

Mode	Cosmos Natural Frequency	Exerimental Natural frequency	% Difference	Damping
1	25.45 Hz	24.94 Hz	2.0%	0.605%
2	67.17 Hz	66.54 Hz	0.94%	0.219%
3	124.47 Hz	121.38 Hz	2.48%	0.439%
4	140.19 Hz	135.54 Hz	3.32%	0.049%
5	182.03 Hz	175.72 Hz	3.61%	0.161%

4 STAGE 2 RESULTS

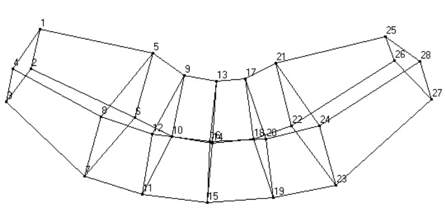
For Stage 2 testing, a CERCA 5-Cell niobium cavity with bulkheads, struts, and dummy tuners installed, was suspended using soft springs from the test stand. The weight of this cavity was approximately 264 lb. The shaker was used to excite the cavity in the axial and transverse directions.

A three dimensional Cosmos model of the cavity was built using ~4600 second order shell elements. Figures 8 and 9 show the first bending mode of the cavity calculated and measured. Figures 10 and 11 show the first axial mode. Table 2 shows all of the results of this stage of testing.



Mode 1 = 17.84 Hz

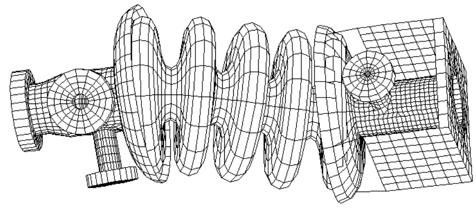
Figure 8: First bending mode, Cosmos model.



Mode 1 = 17.26 Hz
Damping = 0.765%

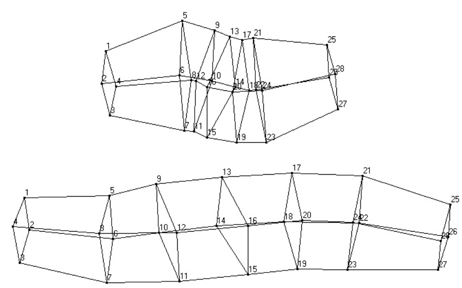
Figure 9: First bending mode, StarModal animation.

The damping values in Tables 1 and 2 are probably affected somewhat by the experimental suspension system suspending the cavity. However, at this stage the goal is to validate the numerical results with testing. Damping



Mode 2 = 44.51 Hz

Figure 10: First axial mode, Cosmos model



Mode 2 = 43.37 Hz
Damping = 3.40%

Figure 11: Two frames of first axial mode, StarModal animation

Table 2: Stage 2 Results

Mode	Cosmos Natural Frequency	Exerimental Natural frequency	% Difference	Damping
1	17.84 Hz	17.26 Hz	3.25%	0.765%
2	44.51 Hz	43.37 Hz	2.56%	3.40%
3	47.87 Hz	46.87 Hz	2.09%	2.22%
4	94.62 Hz	97.79 Hz	3.35%	1.51%
5	115.90 Hz	116.82 Hz	0.79%	0.814%
6	128.75 Hz	123.98 Hz	3.70%	0.970%
7	133.57 Hz	140.50 Hz	5.19%	0.833%
8	152.69 Hz	157.27 Hz	2.99%	0.489%
9	163.78 Hz	163.17 Hz	0.37%	0.961%
10	199.42 Hz	196.41 Hz	1.51%	0.312%
11	201.43 Hz	196.12 Hz	2.64%	0.659%
12	203.97 Hz	205.74 Hz	0.87%	0.331%

values are shown here to illustrate that they are calculated by StarModal. Modal damping values obtained from future, more complex testing will be valuable information for final FE analysis.

5 FUTURE WORK: STAGE 3 TESTING

The final stage of testing will be a modal survey of the helium vessel assembly installed in the vacuum vessel. As much of the cryomodule assembly as possible, such as the power couplers, spoke supports, etc, will be included, while still allowing a feasible test scenario (access to the cavity). A priliminary proposed test setup is shown in Figure 12. The complete cryomodule assembly was shown in Figure 2. Testing at this stage will yield several important factors for final analysis: confidence in the FE model, accurate modal damping values, and measured gain values from input at specific points to output at cavity.

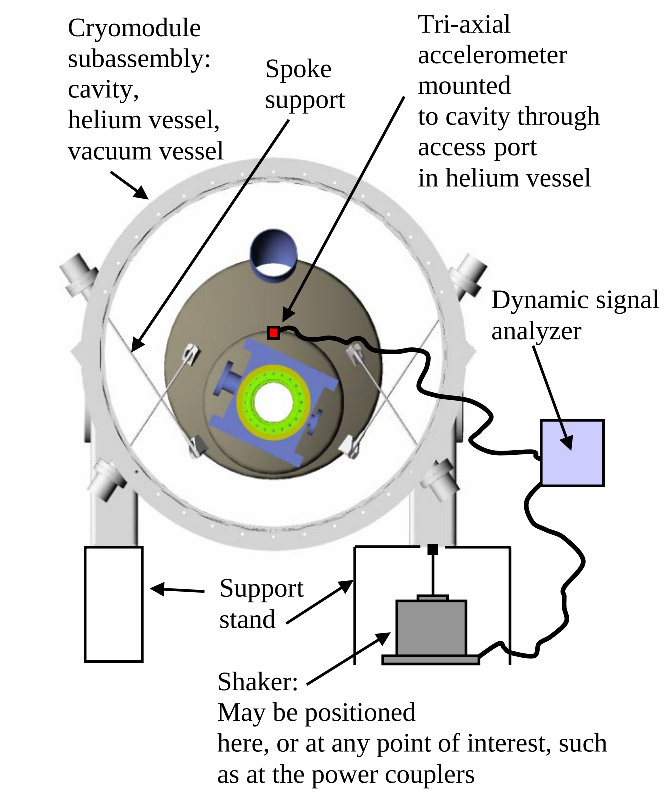


Figure 12: Proposed Stage 3 Test Setup

6 CONCLUSION

A premilinary maximum allowable RF frequency shift of +/- 100 Hz has been given as a frequency budget for the cavity. This number translates to only +/- 0.31 microns axial displacement, given that the frequency sensitivity of the cavity is 319 Hz/micron. This means extremely small cavity motions will cause unacceptable frequency shifts. More study is necessary to determine the cavity's frequency sensitivity to transverse displacements, but a frequency shift was observed when the cavity was deformed in a first bending mode shape. Because the displacements necessary to cause unacceptable frequency shift are so small, we do not feel that simply stiffening the cavity so that its fundamental frequency is above some arbitrary value will necessarily eliminate the problem.

We have developed a three stage plan which we feel will give us a high level of confidence in our finite element model, and therefore in the prediction of resonance displacements due to ambient facility noise. With this knowledge, we can take the necessary steps to ensure that any unacceptable motions are eliminated.

With the completion of the second stage of our test plan, we have obtained good correlation between our numerical and experimental results.

7 REFERENCES

[1] COSMOS/M Version 2.0

[2] Chan, K. C. D., private conversation, 1999

[3] Randall, R.B., and Tech, B., Application of B&K Equipment to FREQUENCY ANALYSIS, 2nd edition, Naerum Offset Tryk, Naerum, Denmark

[4] Spectral Dynamics, The STAR System Manuels, User Manuel, 3405-0113

[5] Spectral Dynamics, The STAR System Manuels, Reference Manuel, 3405-0114

7.1 *Equipment*

[1] Bruel & Kjaer Vibration Exciter Type 4808

[2] B&K Force Transducer Type 8200

[3] B&K Charge Amplifier Type 2635

[4] Endevco Model 63B-10 ISOTRON Triaxial Accelerometer.

[5] Endevco Model 133 3-Channel Signal Analyzer.

[6] Siglab Model 20-42 Hardware Measurement Module

[7] Siglab Version 3.0 Spectrum Analyzer

[8] StarModal Version 5.23.32 Modal Analysis Software