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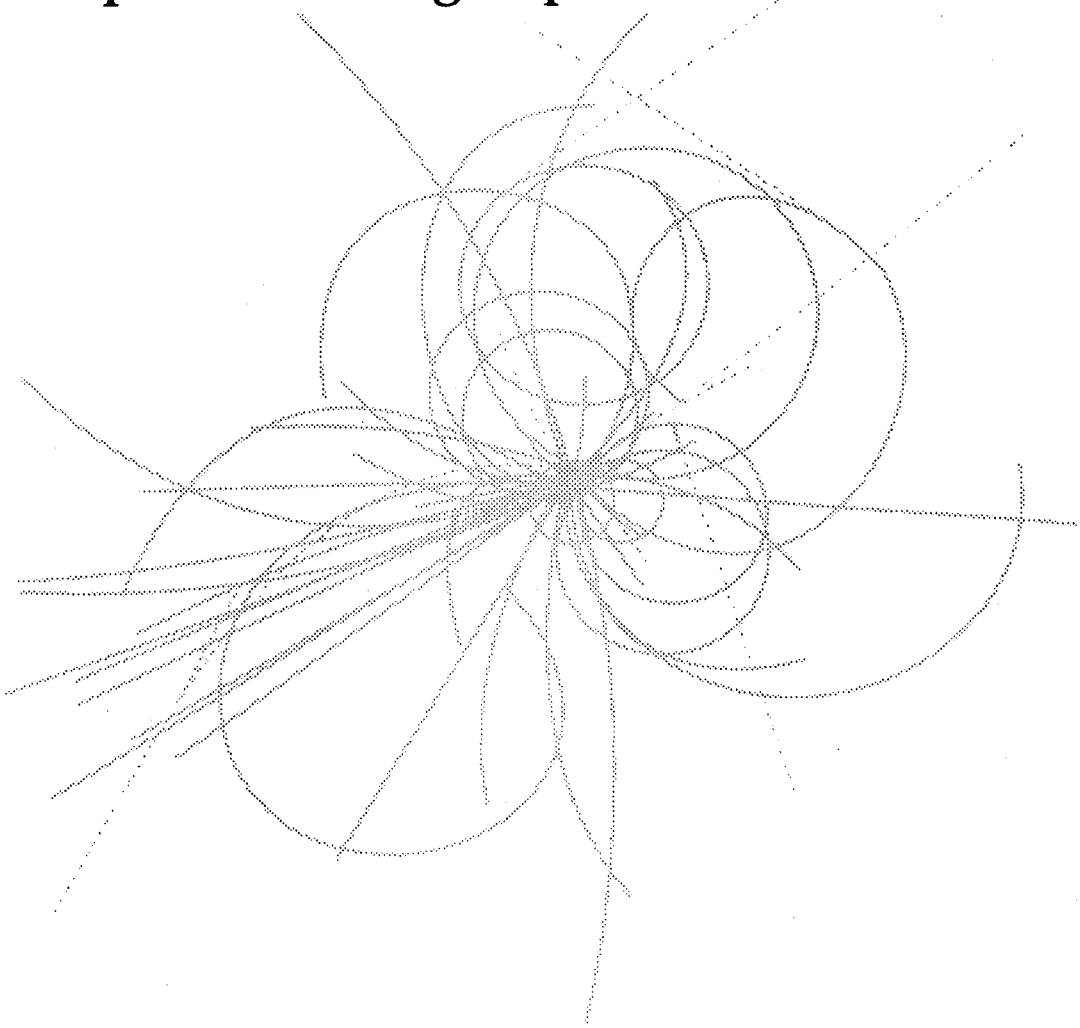

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A. R. Jalloh, E. Daly, and R. Viola

Superconducting Super Collider Laboratory†
2550 Beckleymeade Ave.
Dallas, TX 75237

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STRUCTURAL DYNAMIC ANALYSIS OF THE SSC 40 MM COLLIDER DIPOLE MAGNET UNDER TRANSPORTATION LOADS

A. R. Jalloh, E. Daly, and R. Viola

Magnet Systems Division
Superconducting Super Collider Laboratory*
2550 Beckleymeade Avenue
Dallas TX 75237-3946

ABSTRACT

The Superconducting Super Collider (SSC) magnets will be transported to the tunnel site from various manufacturing locations in the United States. Dynamic loads during transportation could damage the magnets. As part of the overall magnet development program, the effects of transportation loads on the magnets should be investigated to ensure that design limits are not exceeded. A validated structural finite element model of the SSC 40-mm collider dipole magnet (CDM) was excited with actual road input data measured during transport. The test data was in the form of power spectral density (PSD). A random vibration analysis was carried out to determine system response at selected points on the structure. The results showed very good correlation with measured data. Dynamic stresses were calculated to try to understand the mechanism that caused the magnets to yield during shipment. The stresses observed were very small compared to the design limits of the vacuum vessel, where permanent deformation was observed after shipment. It was concluded that the yielding was not due to the transportation loads but to another mechanism that was not included in the model.

INTRODUCTION

The SSC magnets will be transported by road to the tunnel site in Waxahachie, TX, from various manufacturing locations in the United States. The magnets will have to be designed to survive transportation intact, otherwise they will not be acceptable for installation in the SSC tunnel. Thus, a thorough understanding of the impact of the transportation loads on SSC magnets is an essential component of the overall magnet development program.

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An on-going research and development effort in the the Magnet Systems Division of the SSC Laboratory is investigating transportation loads and their effects on magnet design. As part of this effort, the design B 40-mm collider dipole magnet was instrumented with accelerometers, loaded on a flat bed trailer, and driven from the Fermi National Accelerator Laboratory (FNAL) in Batavia, IL, to Brookhaven National Laboratory (BNL) in Upton, NY, over roads and highways typical of those on which the tunnel-bound magnets will be transported. The magnet was mounted on inflatable rubber cushions. Accelerometers were placed at 14 points on the cold mass, vacuum vessel, and trailer bed. The accelerometer locations on the magnet correspond to the locations where large dynamic amplitudes are expected. Displacement time histories at the accelerometer locations were recorded during transit.[1] These were subsequently fed into an HP3567A Spectrum Analyzer and converted to PSDs. A typical PSD is shown in Figure 1. The PSD defines an upper envelope of the distribution of vibrational energy incident at each accelerometer location.

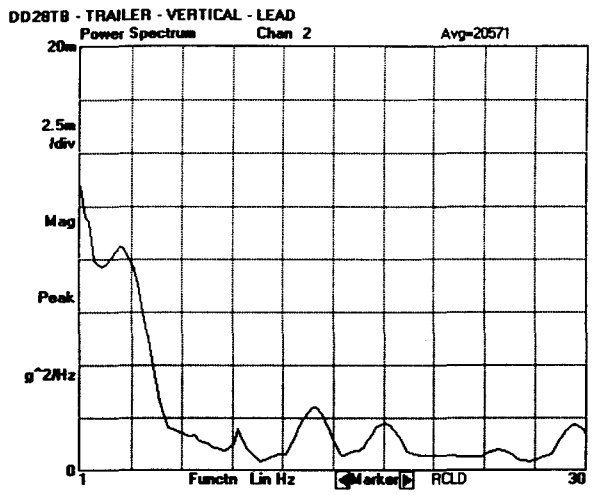

Figure 1. Representative PSD Curve for Trailer Bed.

It should be noted that the range of interest for transportation loads is approximately 0–50 Hz. Dynamic displacement is inversely proportional to the square of the frequency of interest, as shown in Eq. (1). Dynamic amplification for sinusoidal motion can be expressed as:

$$d = \frac{A\, g_c}{(2\pi f)^2}, \tag{1}$$

where d is dynamic displacement, A is acceleration amplitude normalized to 1 g, g_c is consistent units for acceleration due to gravity, and f is the frequency of interest. Consequently, small amplitudes of acceleration resident at frequencies above 50 or 60 Hz yield very small dynamic displacements. An inspection of the test data showed a drop-off and leveling out of the PSDs after 15–20 Hz. As a result, the range of interest considered in the present study is 0–30 Hz.

A finite-element model of the CDM has been developed and validated with test data by Jalloh et al.[2] and was used to predict beam tube displacements due to ground motion excitation. Good correlations were obtained between natural frequencies and corresponding mode shapes predicted by the model and those obtained experimentally. This model is a very useful tool for all types of dynamic studies, ranging from further ground motion studies to handling and transportation studies. Benda et al.[3] investigated the stresses in the SSC

dipole magnet that resulted from transportation loadings. However, that work was not validated with test data. The validated dynamic model for the 40-mm CDM developed by Jalloh et al.[2] was the basis of the present work.

For the present study, the inflatable rubber cushions were added to the model to represent the structural dynamic system as it is mounted on the trailer bed. Natural frequencies predicted by the augmented model were compared with those obtained experimentally. This was done to establish that the model adequately represented the dynamic system. The output PSDs of the trailer bed were then input into the model as forcing functions, and a random vibration analysis was carried out. Mean square displacements and their 1-sigma deviations were obtained at points on the vacuum vessel and cold mass corresponding to accelerometer locations. Also, response PSDs were obtained at these points. These results were compared with test data.

After shipment, the vacuum vessel was observed to have deformed permanently. A dynamic stress analysis was carried out to try to understand the mechanism that caused the yield. However, because of software restrictions and limitations, the stress field could not be computed directly in the random vibration analysis. Instead, a response spectrum analysis was carried out with input data modified from the trailer bed PSD to determine the stress field. The dynamic stresses predicted by the model were very low compared with the yield strength of steel, the vacuum vessel material.

MODEL DEVELOPMENT AND ANALYSIS

The complete 40-mm CDM and inflatable rubber cushions, as they were mounted on the trailer bed, were modeled for this study. The cold mass model accounted for the structural and dynamic characteristics of the magnet only. Thermal and magnetic effects were neglected because they were not considered relevant to the objectives of this work. The inflatable rubber cushions were modeled as three-dimensional helical springs. The model also included the trailer bed as the source of dynamic excitation. However, it was dynamically decoupled from the magnet and rubber cushion system.

The model was developed using the finite-element method and ANSYS, a commercial finite-element software package.[4] The magnet was modeled with thin-shell and thick-plate finite elements using ANSYS elements Stif63 and Stif43, respectively.[3] The inflatable rubber cushions were modeled by ANSYS element Stif14, the three-dimensional longitudinal spring-damper element without torsion. The springs were anchored at the top to the external feet of the magnet and at the bottom to a large mass that simulates the trailer bed. The trailer bed was modeled with ANSYS element Stif21, a mass element with three translational degrees of freedom. The trailer bed was given an excessively high mass value to ensure that its dynamic characteristics were decoupled from that of the magnet and rubber cushion system. The mass of the trailer bed was made 1×10^8 times larger than the combined mass of the magnet and rubber cushion, as recommended in the ANSYS software users' guidelines. With this large mass anchor, the magnet was not restrained during the dynamic analysis, as is required in ANSYS for the random vibration analysis that was ultimately carried out. This, of course, resulted in rigid body modes. However, the non-rigid body modes were not affected by the dynamics of the truck bed. The decoupling of the modes was verified by carrying out two separate free-vibration analyses of the system. The first analysis was with the large unconstrained masses attached to the springs. The second analysis was with the large masses removed and the bottom ends of the springs constrained. The natural frequencies and mode shapes of the non-rigid body modes from the first analysis were the same as those from the second analysis.

After the free vibration analysis was completed, a harmonic analysis was carried out to compute a transfer function for use in the random vibration analysis. The PSD of the trailer

bed in the vertical direction, shown in Figure 1, was discretized and input as the forcing function in the random vibration analysis. This PSD is for the vertical direction only. It should be noted that the test data showed that only the vertical direction excited significantly during transportation. Structural damping of 2% for the magnet components and 15% for the springs was assumed. The value of 15% for the springs was obtained from the literature.[5]

Output displacement PSDs were generated from the analysis for points on the magnet where accelerometers were located. The results of the random vibration analysis were consistent with test data.

In an attempt to understand the mechanism that caused the magnets to yield during shipment, a dynamic stress analysis was carried out. The current version of ANSYS does not lend itself to the direct computation of the stress field in random vibrations analysis. Therefore, a response spectrum analysis was carried out using the trailer bed PSD shown in Figure 1 as the input spectrum. However, because of software restrictions and limitations, the input had to be modified to a constant amplitude PSD. Also, the displacement results showed that the effects of the transportation loads were significant only in the low frequency range. The modified input spectrum was 11×10^{-3} g^2/Hz over a frequency range of 1–10 Hz.

RESULTS

The natural frequencies of the vertical modes predicted by the magnet and rubber cushion system model and those obtained by test were compared to ensure that the model adequately represented the dynamic characteristics of the structure. This comparison, shown in Table 1, reveals very good correlation between the analytical and experimental results. The fundamental frequency showed complete agreement. In general, there was agreement within 8% for all modes.

Table 1. Comparison of Analytical vs. Experimental Frequencies for the 40-mm Collider Dipole Magnet in the Vertical Direction.

Vertical Mode Number	Analytical Results Natural Freq (Hz)	Experimental Data Natural Freq (Hz)
1	1.498	1.5
2	5.93	5.5
3	6.93	6.5
4	9.85	10.0

The mean square displacements and the 1-sigma standard deviation for the lead and return ends of the vacuum vessel and cold mass are shown in Table 2. Experimental data for the mean square displacements for the cold mass lead and return ends were not available. Again, there is good correlation between model results and experimental data.

Table 2. Comparison of Analytical vs. Experimental Mean Square Displacements for the 40-mm Collider Dipole Magnet.

Location	Analytical Mean Square Disp (in.^{2})	Expt. Mean Square Displacements (in.^{2})
Vac Vessel Lead End	2.58 (1.61)**	2.86
Vac Vessel Return End	2.54 (1.59)	2.31
Cold Mass Lead End	3.09 (1.76)	NA
Cold Mass Return End	1.83 (1.35)	NA

**The numbers in parentheses represent the 1-sigma standard deviation of the analytical mean square displacements.

The output displacement PSDs for the four locations on the structure are shown in Figures 2–9. There is good correlation between the predicted PSD and test data for each location on the structure. Also, the maximum dynamic displacements occur in the low frequency ranges. The response spectrum analysis showed low dynamic stresses in the magnet. The maximum values were about 1000 psi, in the area where the center support post attaches to the vacuum vessel. This value is extremely low compared with the yield strength of steel, the vacuum vessel material. Of course, it should be noted that the stress field computation was carried out with a modified PSD because of software restrictions. Thus, the stress field was not computed accurately. However, the stress values obtained indicate that even if the proper analysis had been carried out, the stresses would not have approached the yield threshold.

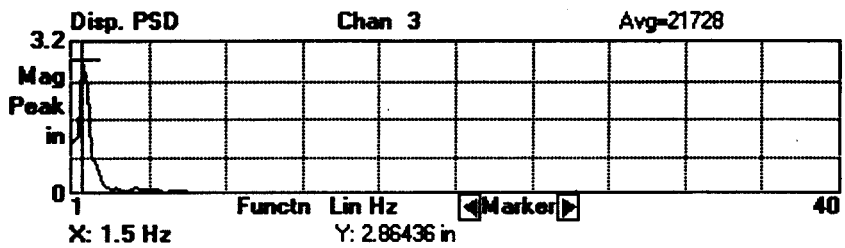

Figure 2. Displacement PSD for Vacuum Vessel Lead End/Test Data.

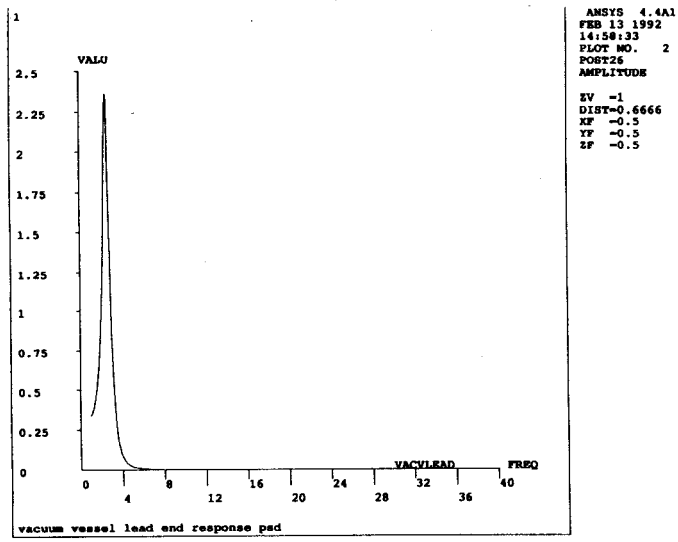

Figure 3. Displacement PSD for Vacuum Vessel Lead End/Analytical Results.

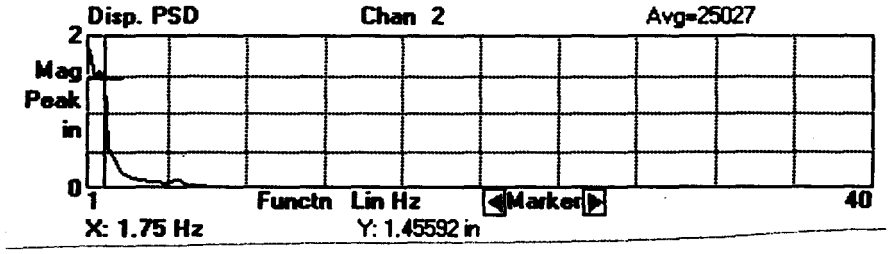

Figure 4. Displacement PSD for Vacuum Vessel Return End/Test Data.

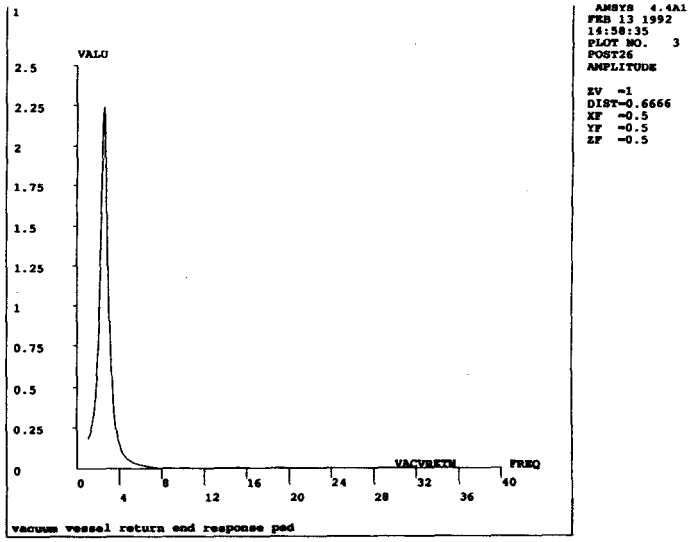

Figure 5. Displacement PSD for Vacuum Vessel Return End/Analytical Results.

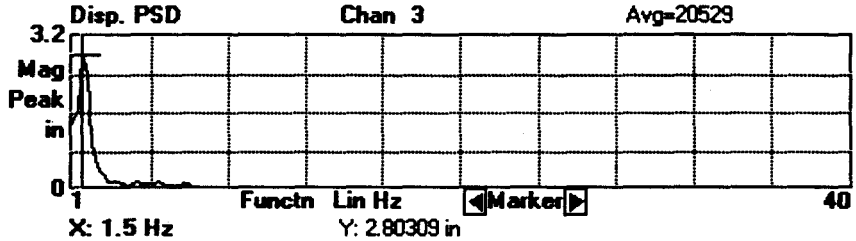

Figure 6. Displacement PSD For Cold Mass Lead End/Test Data.

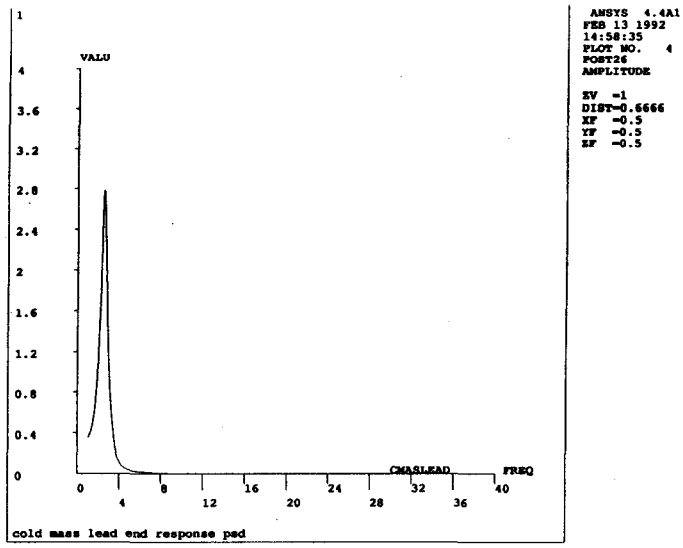

Figure 7. Displacement PSD for Cold Mass Lead End/Analytical Results.

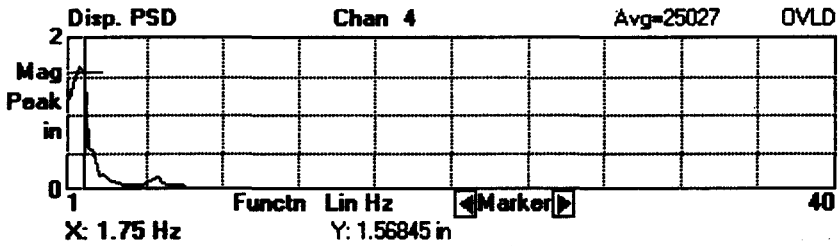

Figure 8. Displacement PSD for Cold Mass Return End/Test Data.

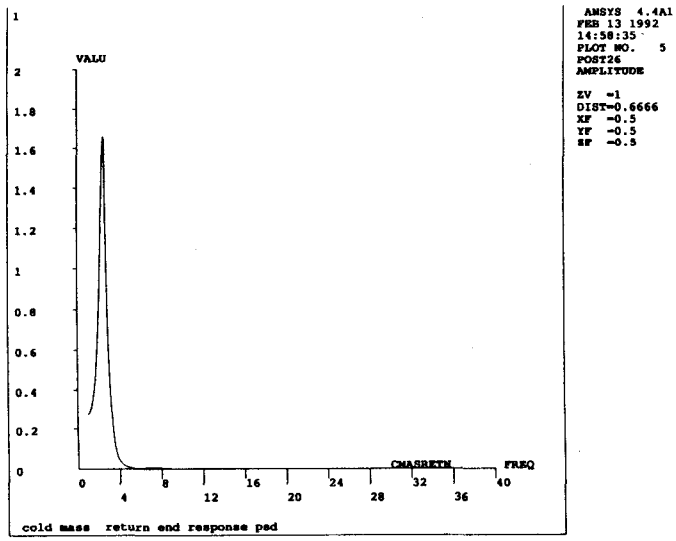

Figure 9. Displacement PSD for Cold Mass Return End/Analytical Results.

CONCLUSIONS

The model accurately represented the transportation environment experienced by the 40-mm CDM, confirming that it is a useful tool for these types of studies. Computer predictions of dynamic displacements compared well with experimental data. However, initial computer predictions of dynamic stresses were very low compared with the yield strength of the vacuum vessel steel. The software calculates dynamic stresses using the dynamic displacements in combination with the stiffness matrix. This suggests that another mechanism that was not included in the model caused the deformations seen in the vacuum vessel. A possible candidate is stress relief of the welds that join sections of the vacuum vessel pipe to the reinforcing rings. If these welds were already pre-stressed, lower dynamic stresses could possibly drive the material in the weld to yield conditions.

FUTURE WORK

This work was investigative in nature, and it proved that a validated model is useful. A validated model of a 50-mm CDM as well as other types of magnets should be developed using this approach to complement research and development efforts on all magnet development programs. Also, the stress field should be determined for the entire magnet system under random transportation loading. As was pointed out, the current version of the ANSYS software does not have this capability. It is expected that the next version will lend itself to this type of analysis. If that proves correct, then an elaborate stress field computation should be pursued.

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