Designing Motorola Mobile Radio for Shock and Vibration Requirements

Ben Nagaraj, Tony Kakiel, Ketul Patel and Dave Reiff

G&PS, Motorola, Plantation, Florida

Abstract: Mobile products developed by Motorola must meet internal vibration test requirements formulated from industry and military specifications which include shock, random and swept sine profiles. A key component is the mobile radio trunnion mounting system. Inadequate design margin of the trunnion can lead to mounting system failure as well as failure of internal components (displays, flex circuits, PCB components, etc.). The testing regimen requires approximately one week to complete and several passes to come up with the proper design. To reduce the test-based development time, simulation methods using ABAQUS have been developed.

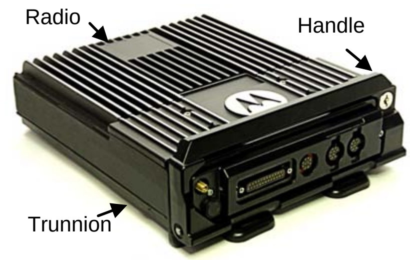
This paper presents the analyses techniques using ABAQUS to identify and prevent typical failure modes seen in shock and vibration testing. The analysis techniques are (i) Steady state response with base excitation for vibration requirement using implicit code, and (ii) Dynamic, explicit with impulse load at the base for shock using explicit code. These analysis methods are being used at Motorola to reduce overall development cycle time required to introduce a new product to market.

Keywords: Vibration, Shock, Motorola Mobile Radio.

1. Introduction

Two typical Motorola mobile products and their mounting systems (trunnion) are shown in Figure 1. In the first setup, the Radio is mounted to the trunnion using screws. While, in the second setup, there are no screws, but the trunnion has stops to avoid motion in each direction. There are mainly two types of mechanical tests the products must pass for qualification. One is for vibration and the other is for shock. Typical test time is between 1 to 2 weeks. An inadequate design can result in one or more of the following failures. (i) radio internal component failure, (2) mounting screw back-off, (3) mounting screw failure, and (4) trunnion cracking at the anchor screws.

In this study we are focusing on the qualification of the trunnion. An inadequate design can lead to trunnion failure as shown in Figure 2 i.e. cracking at the anchor screws for product shown in Figure 1a. Similar failure is observed for product shown in Figure 2a.



a) Product with mounting screws b) Product with self locking mechanism

Figure 1. Motorola mobile radios and the mounting systems.

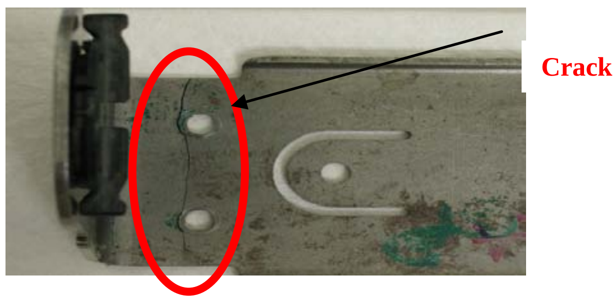


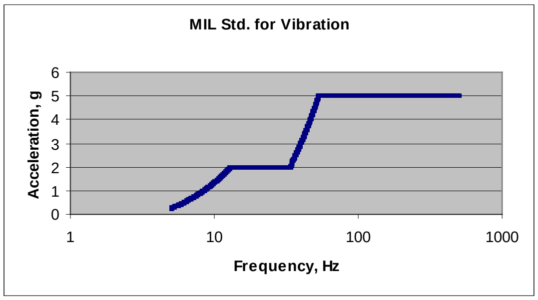
Figure 2. Typical failure mode of Trunnion in Figure 1a.

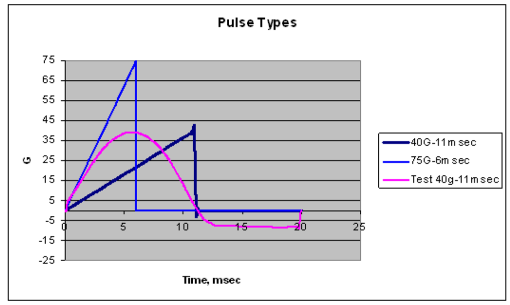
2. Motorola vibration & shock test profiles

There are mainly two test requirements for mechanical reliability of mobile radio products. One is swept sine vibration based on MIL Spec 810. The frequency sweep is from 5 to 500 Hz and back to 5 Hz, and the excitation load varies as a function of the frequency as shown in Figure 3a. The sweep duration is 1 hour. The test requirement is in all three axes. The second test requirement is for shock loading. There are three types of Shock profiles depending on the profile, and are shown in Figure 3b. The test requirement is in each direction, x+, x-, y+, y-, z+ and z-. In both the cases, the loading is applied at the anchor screws. After every sweep or shock, the trunnion is visually inspected for cracks and screw loosening.

3. Finite element model

The finite element model has progressed over the years as capabilities in ABAQUS, computer CPU and disk space have improved. Very early models included shell elements for the trunnion and the radio was replaced by a point mass at the radio CG. The screws are replaced by rigid beam MPC's, between the trunnion and the CG of the radio. The preload on the trunnion from the screw is ignored. In the later stages, the trunnion is modeled using continuum shell (SC8R) or Hex elements with incompatibility mode (C3D8I). The radio has been modeled as 3D rigid body





a) Vibration test profile **b) Shock test profiles**

Figure 3. Vibration and shock test profiles.

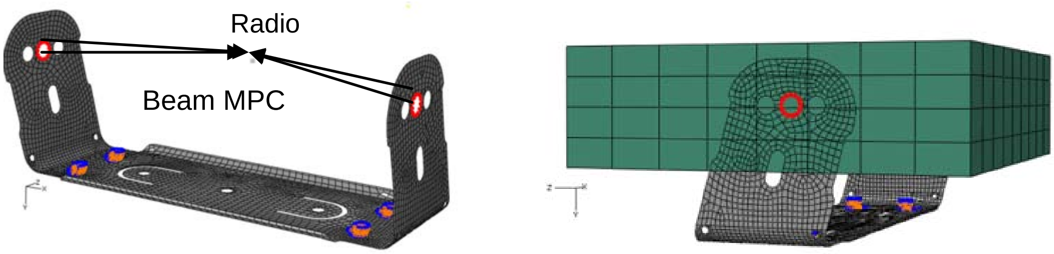
using Hex (C3D8) elements with uniformly distributed mass density. The screws are modeled without the threads and tied appropriately at the point of contacts. For products with a self locking mechanism, the point of contacts between the radio and the trunnion are tied for linear dynamic models. For nonlinear dynamic models, contact interactions between the radio and the trunnion are established.

4. Finite element analysis and results

Usually three types of analyses are performed during the design stage. They are (i) Natural frequency extraction, and their corresponding mode shapes within 1000 Hz. This is to identify possible modes of vibration within the sweep range and to identify the high stress locations. (ii) Steady state dynamics with base excitation along positive x, y & z directions. This analysis is carried out to identify the most severe sweep direction and then to identify the peak stress and its location. The design is iterated such that the peak stress is a fraction of yield strength of the material. (iii) Dynamic analysis for Shock loading at the base. For the product shown in Figure 1.1a, the ABAQUS/Standard code is used. For the product shown in Figure 1.1b, the ABAQUS/Explicit code is used since contact interaction plays a very important role. From this analysis, the high plastic strain regions are identified and the design is iterated such that the maximum plastic strain is a fraction of allowable elongation of the material.

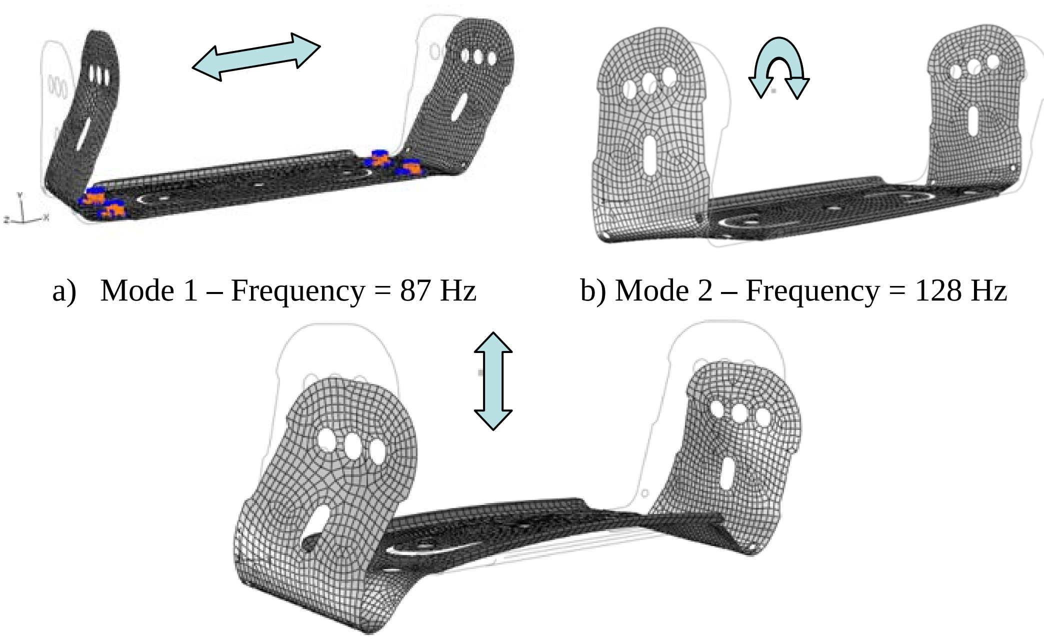
4.1 Natural Frequency Extraction

For the product shown in Figure 1a, the simplified FE model is shown in Figure 4. Figure 4a shows the model with a point mass at the Radio CG, while Figure 4b shows the model with a uniformly distributed mass. The predicted natural frequencies and their corresponding mode shapes within the sweep frequencies are shown in Figure 5. They are ‘side-side’ sway at 87.4 Hz (Figure 5a), ‘rotation about radio mounting axis’ at 128 Hz (Figure 5b), and ‘up-down’ translation at 372 Hz (Figure 5c). These results and along with the results from the distributed mass model are compared with the observed resonance frequencies during sweep tests in Table 1. Both models predict the frequencies for first and third modes with greater than 90% accuracy. For the second mode, the mode shape is predicted correctly but not the frequency. Further model refinements are planned for the future to predict this frequency more accurately.



a) Simplified model with point mass b) Simplified model with distributed mass

Figure 4. Simplified finite element model for product shown in Figure 1a.



a) Mode 1 – Frequency = 87 Hz b) Mode 2 – Frequency = 128 Hz

c) Mode 3 – Frequency = 372 Hz

Figure 5. Predicted natural frequencies and their mode shapes using point mass.

Table 1. Natural modes and their frequencies

Mode	FEA model Point mass	FEA model Distributed mass	Observed test data
1	87 (4% error)	77 (-8% error)	84 Hz
2	128 (-42% error)	101 (-54% error)	220 Hz
3	372 (6% error)	371 (6% error)	350 Hz

4.2 Steady state dynamics

For the above model after extracting the natural frequencies, mode-based steady state dynamic analyses were performed with the base excitation shown in Figure 3a along each axis separately. All the modes within 1000 Hz are used in the analysis. Based on previous experience, a modal damping between 5 to 10% is assumed for all the modes. A typical response of radio CG due to base excitation along X, Y and Z are shown in Figure 6. When excited along X-direction, there is one dominant peak at first mode and this peak is the maximum among all the three excitations. When excited along Y & Z -directions, there is peak response both at second and third modes. Figure 7 shows the radio CG response to base excitation along the X-direction for modal damping ratio ranging from 5 to 10%. These results are compared with the test response. The peak test response is about 17G. The simulation result with a damping ratio between 8 and 10% correlates with the test measurement. Figure 8 shows Mises stress contour at first mode (~ 87 Hz). The maximum principal stress contour plot is also similar to the Mises stress plot. The FEA predicts peak stress location at the anchor screws where base excitation is applied and matches well with the observed crack initiation location as shown in Figure 2. The maximum stress shown at the top bolt location for radio attachment is due to beam MPC assumption, which may not be accurately represented. Figure 9 shows the Mises stress at the peak stress location (at anchor screw) for all the three base excitations. It is clear from the plot that at the anchor screw, stress contribution from excitation along X and Z loading are significant.

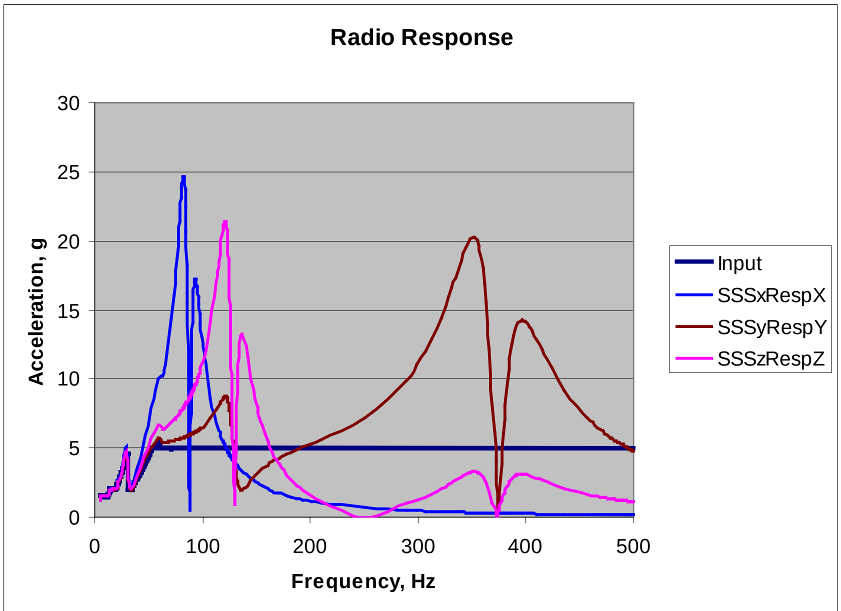


Figure 6. Radio CG response to base excitation along X, Y & Z directions

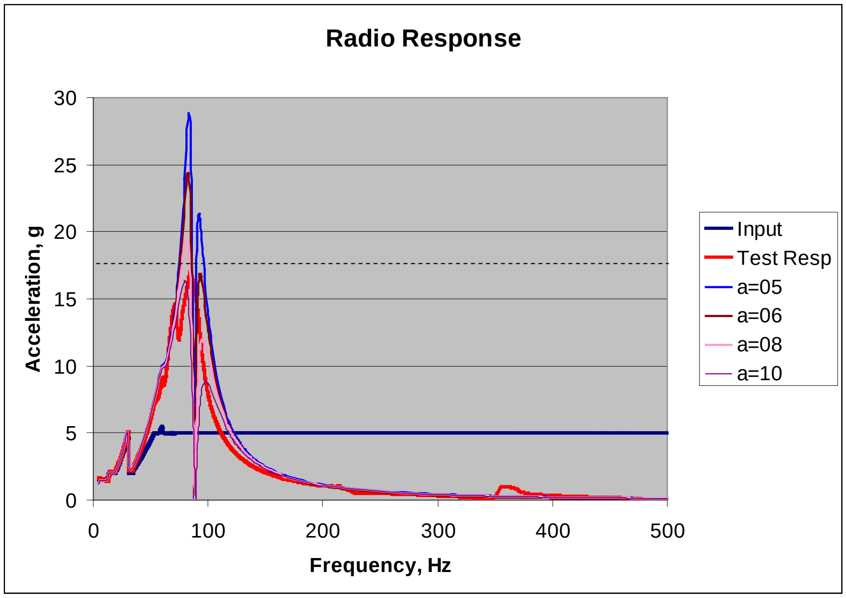


Figure 7. Radio CG response to base excitation along X with different damping

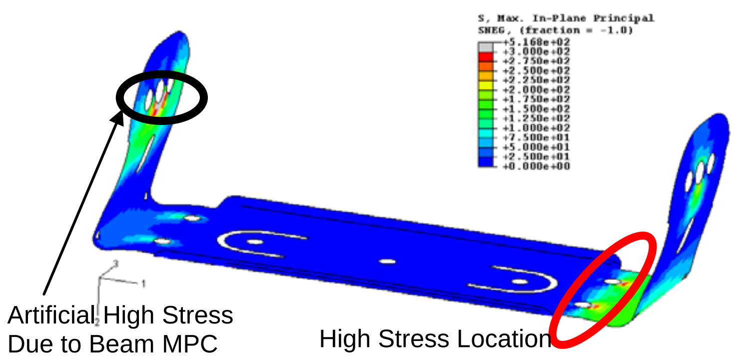


Figure 8. Mises stress response at 87 Hz for X-excitation.

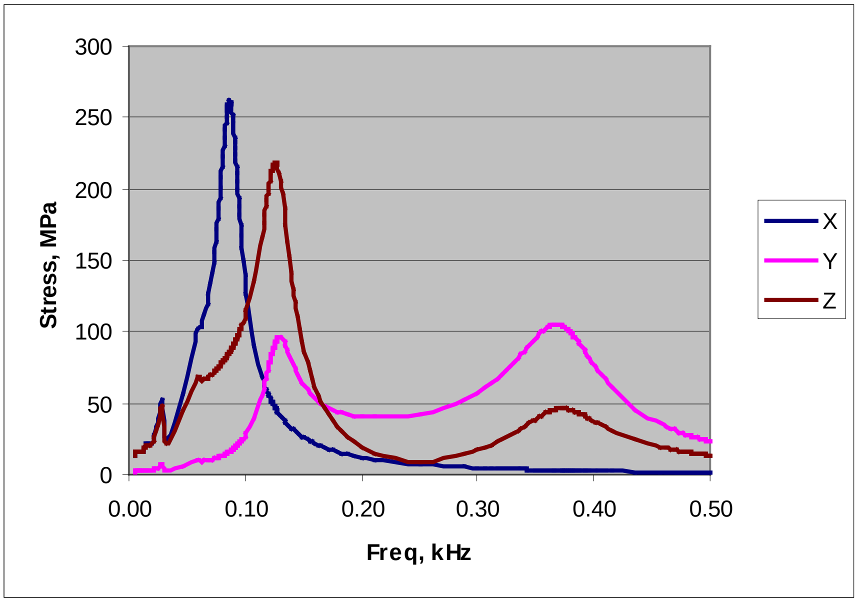


Figure 9. Mises stress response at peak location to all the three base excitations

4.3 Dynamic analysis for shock loading

For the product shown in Figure 1b, a simplified representation of the CAD model for FEA showing the mounting location is shown in Figure 11, and the corresponding further simplified FE model is shown in Figure 12. The trunnion, handle and the hinge are modeled in detail. Only the radio contacting surface is modeled using rigid elements with a reference point at the radio CG. The total mass of the radio is modeled using a point mass at the reference point. All contact interactions are modeled between the components. The base excitation shown in Figure 3b is applied at the mounting locations along each direction separately. Only the results from Z-direction excitation loading are discussed in this paper. The analyses are carried out in ABAQUS/Explicit.

The responses of the radio CG for positive and negative Z-excitations are shown in Figure 13 and 14, respectively. For any given pulse, the responses are quite different for positive and negative excitations. This is because different contact interactions come into play in each direction. For positive excitation, the 75G shock pulse is more severe than the other two. For negative excitation, the 40G, 11msec pulse is more severe. The FEA prediction is compared with the test data for 40G, 11 msec pulse and shown in Figure 15 and 16 for positive and negative excitations, respectively. For positive excitation, the predicted first peak value (125G) closely matches with the test data

(110G) and occur approximately between 4 to 4.5 msec. The second peak shows a similar response, except that the FEA prediction is slightly delayed. For the negative excitation, the predicted first peak value (75G) closely matches with the test data (85G) and occurs approximately between 9.5 to 10 msec. The response pulse and crossing point between negative to positive also match well with the test data. FEA predictions for both excitations have higher frequency content which needs to be damped out.

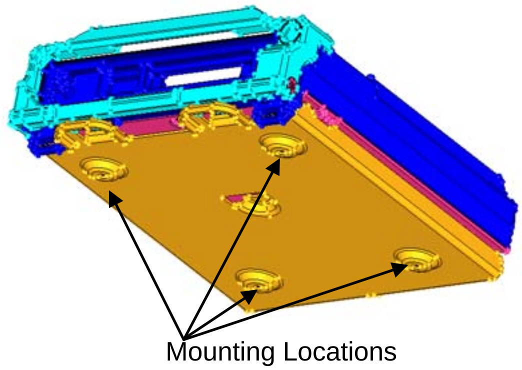


Figure 11. Simplified CAD model for FEA.

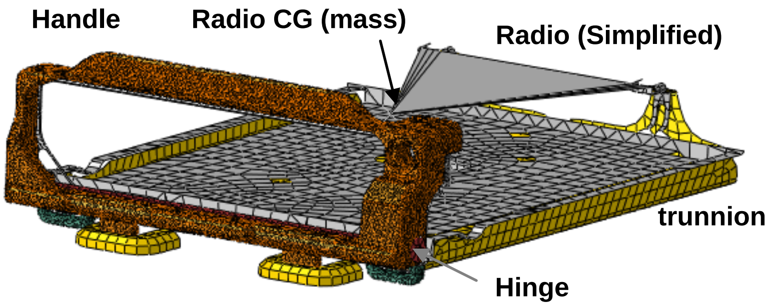


Figure 12. Simplified FE model for dynamic analysis with shock loading.

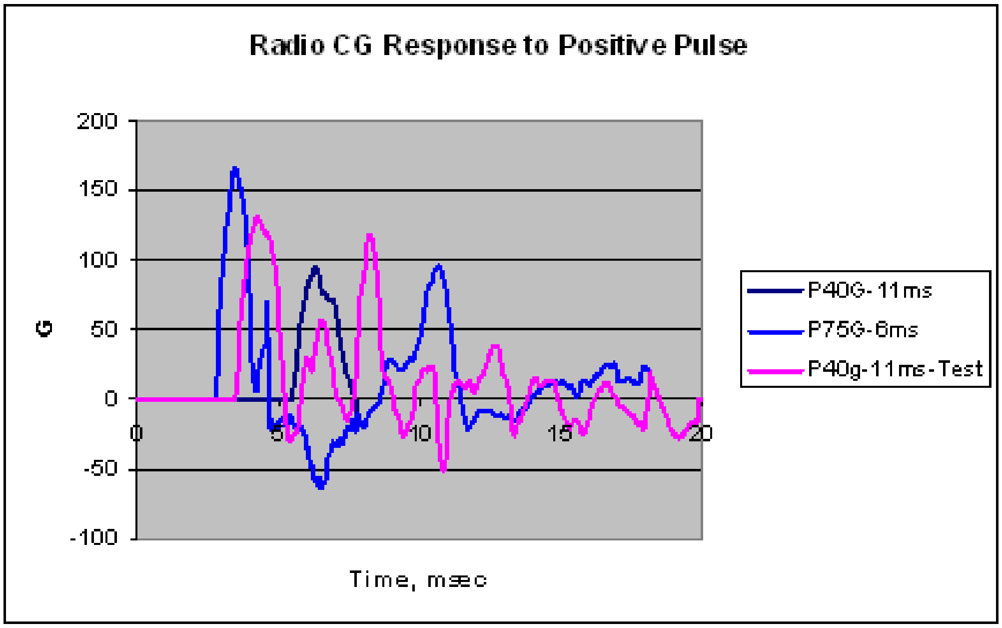


Figure 13. Radio CG response for positive Z-excitation.

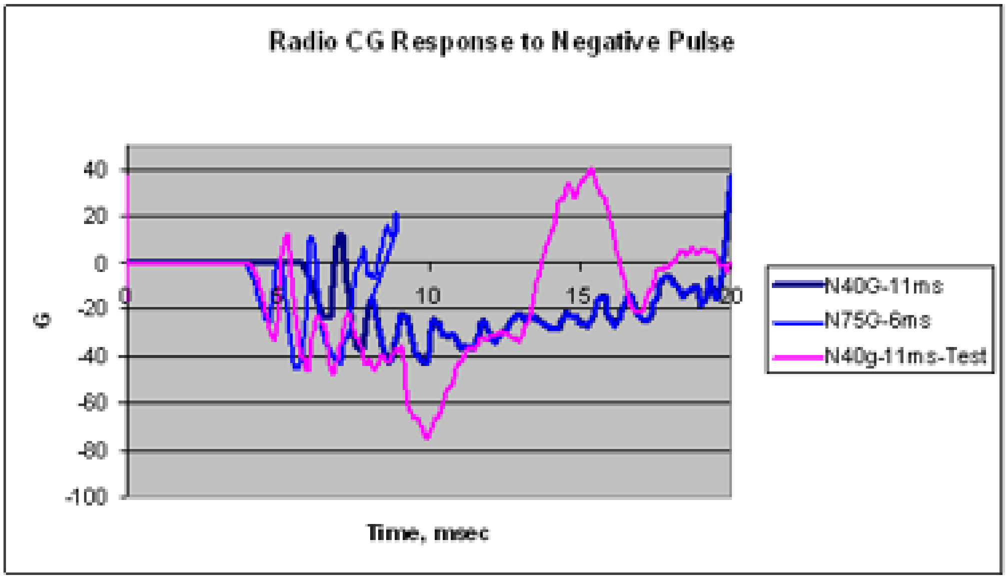


Figure 14. Radio CG response for negative Z-excitation.

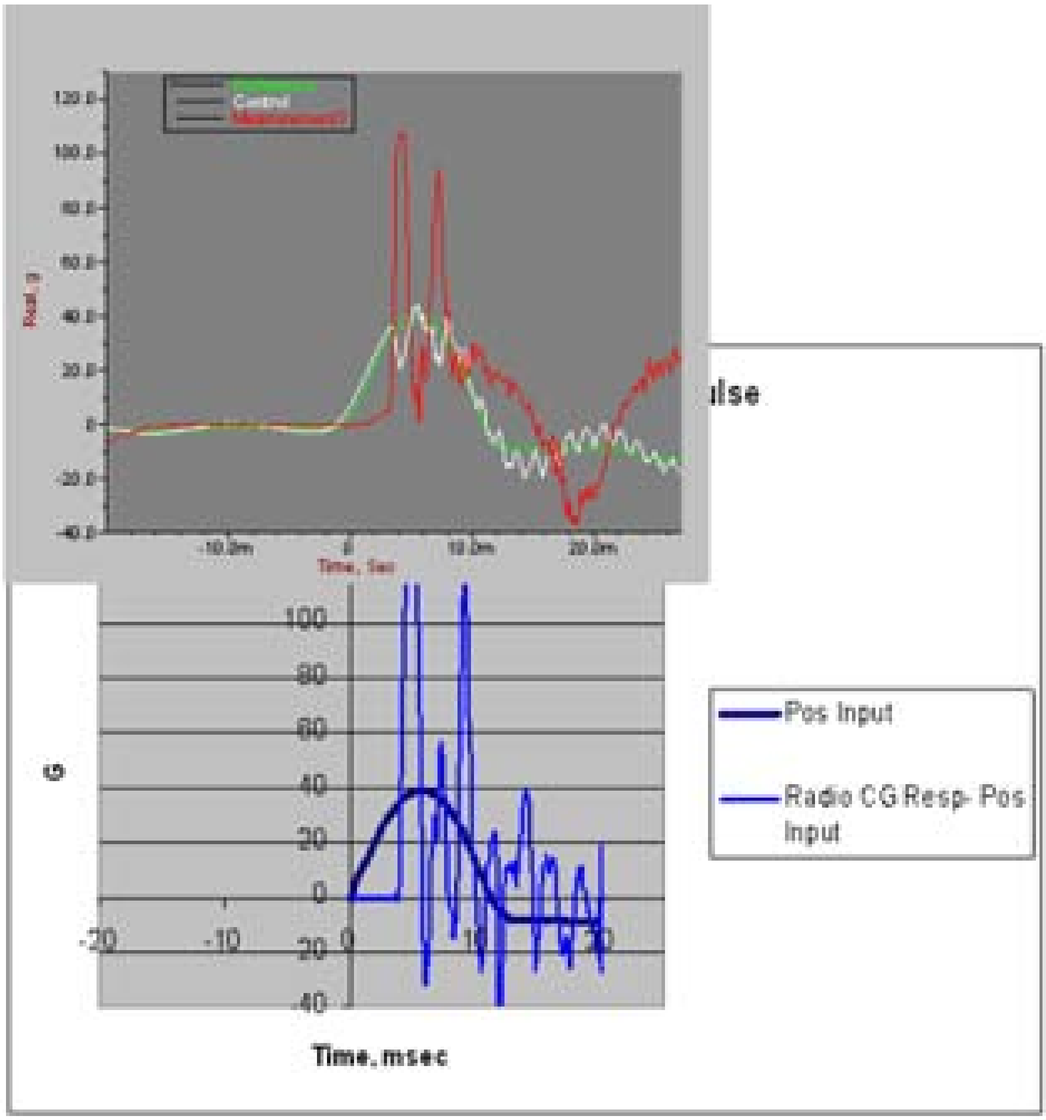


Figure 15. G Response comparison with test data for positive Z-excitation.

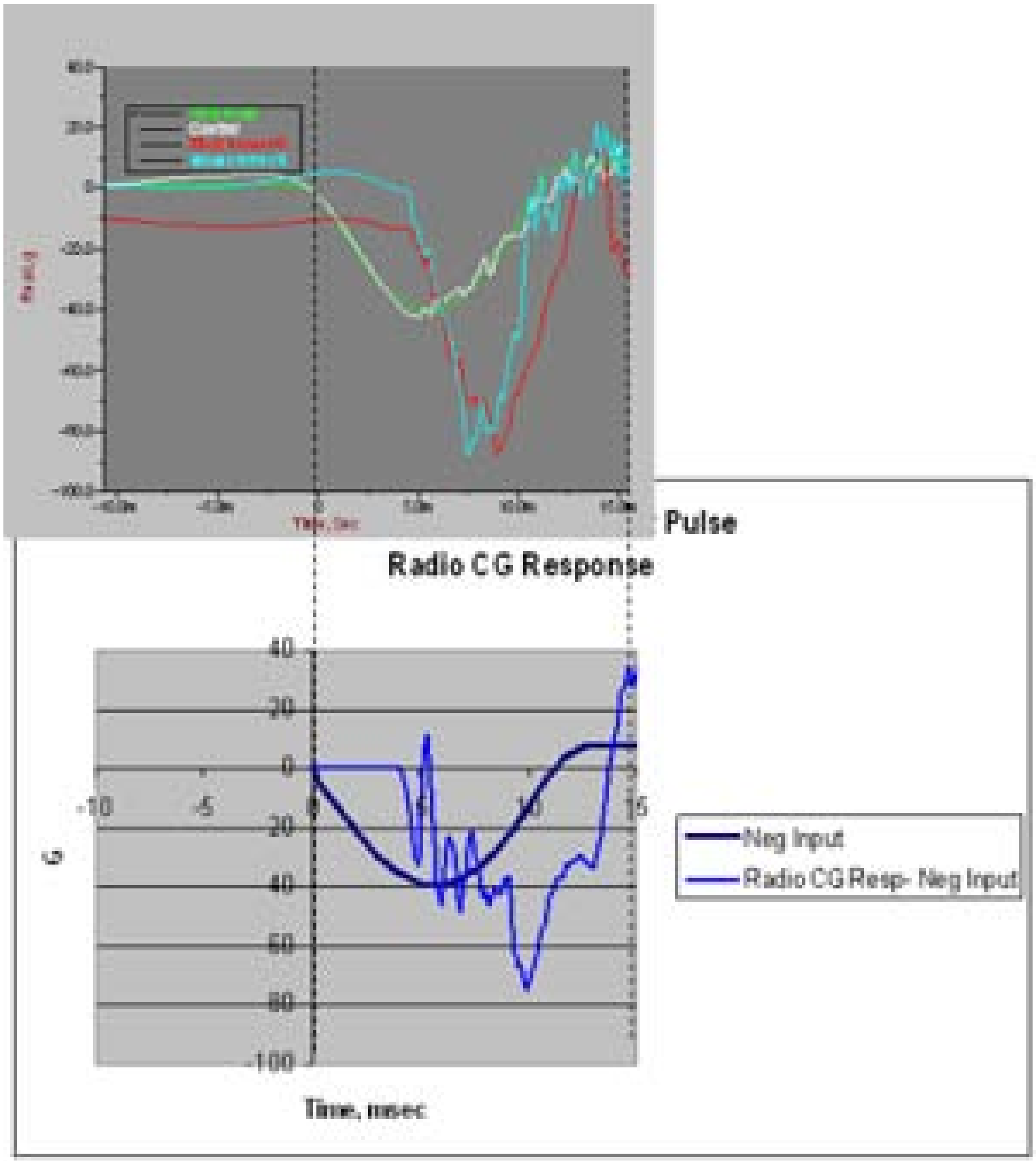


Figure 16. G Response comparison with test data for negative Z-excitation.

5. Conclusions

- Motorola internal vibration and shock test requirements for a mobile radio “trunnion” mounting system design are discussed. FEA simulation to guide design iterations leading to a viable product design is demonstrated. The simulation results are compared with test results.
- Natural frequency predictions are greater than 90% accurate for two translational modes. The natural frequency corresponding to a rotational mode was only about 50% accurate and is being investigated.
- The radio CG acceleration response prediction correlates well with test results for vibration analysis. The high stress location at the anchor screw location correlates with the cracks seen in the actual trunnion failures.
- For dynamic shock analysis, peak acceleration prediction at the radio CG was about 85% accurate for both positive and negative excitations. For negative excitation, the pulse shape and width are also predicted accurately. Some high frequency responses are observed in FEA, but are damped in the actual test.
- Overall, the FEA results compare well with the test results and are very helpful in designing Motorola products to meet internal specifications for mechanical reliability.