Random Vibration Analysis of Dual 101 EM Assembly

Carl Cheng Ph.D., Jeff Crull
Bjorksten | bit 7, Madison, Wisconsin

Wes Marcus, Pete Kjeer, Ron Hansen
Minnkota Motors Division, Johnson Outdoor, INC. Mankato, Minnesota

Abstract

A computer random vibration simulation of a Dual 101 electronic motor assembly are performed by using ANSYS [1] finite element analysis software. A testing procedure is performed to check the acceleration spectrum density. Then the standard acceleration spectrum density curve is used to apply to the structure and then the stress and fatigue life of the structure are analyzed.

Keywords: FEA, Random Vibration, Stress, Motor, Spectrum Density, Damping Ratio, Gaussian distribution, Fatigue

1. Introduction

The purpose of Dual 101 EM project is to design a motor mounting plastic bracket which can hold two motors at the same time. The motors are mounted on the Anti Cavitation plate of the boat's engine. The full assembly of the bracket and motors need to pass 30 G impact test and 8 hours standard random vibration test. The tooling for bracket alone will cost about one million dollars. Therefore, it is necessary and cost effective to simulate the tests by using FEA method. The impact analysis is simplified to a static analysis with the static load equal to 30G applied to the plastic bracket directly. After several iterations of the design and analysis, the structure successfully passed the 30G static load condition. A testing procedure is performed to record the acceleration vs. time curve of a boat then a MathCAD [2] program with FFT (Fast Fourier Transfer) was performed to transfer the curve from time domain to frequency domain. Several testing acceleration spectral density curves were compared with MIL-STD-810F, Figure 514.5C-17 General Minimum Integrity Exposure curve (Fig.1) and a conclusion is found that the MIL-STD-810F basically cover all the testing curves. Therefore, standard MIL-STD-810F is used during overall random vibration analysis. The final product based on the analysis successfully passes the test.

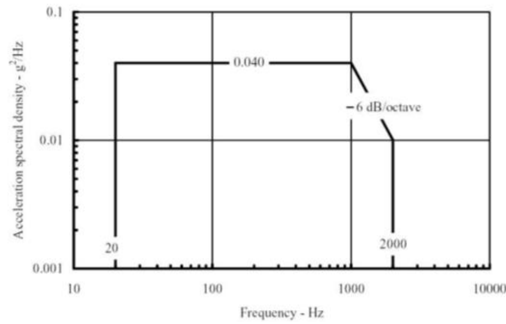

Fig.1 MIL-STD-810F, Figure 514.5C-17 General Minimum Integrity Exposure

2. Approach

2.1 Acceleration spectral density curve

MathCAD program is used to read the acceleration vs. time test data and perform Fast Fourier Transfer to generate the spectral density curve. The following curves show the procedure of this transformation. There are more than twenty tests were performed. Only one testing data is used here for demonstration purpose.

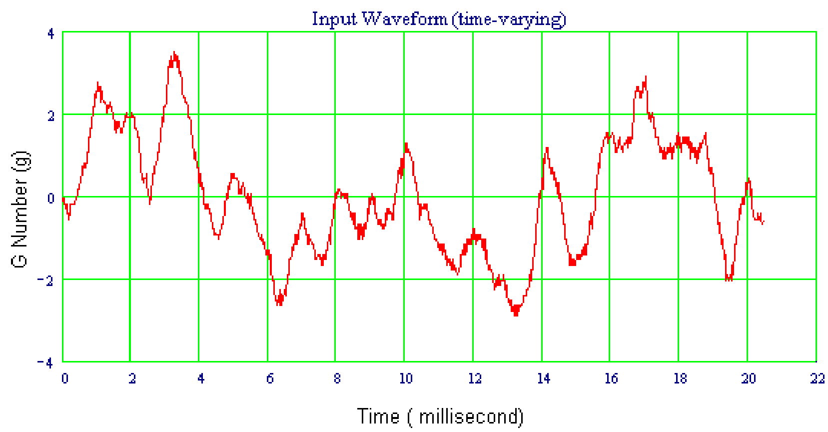

Fig.2 Acceleration vs. Time Testing Curve

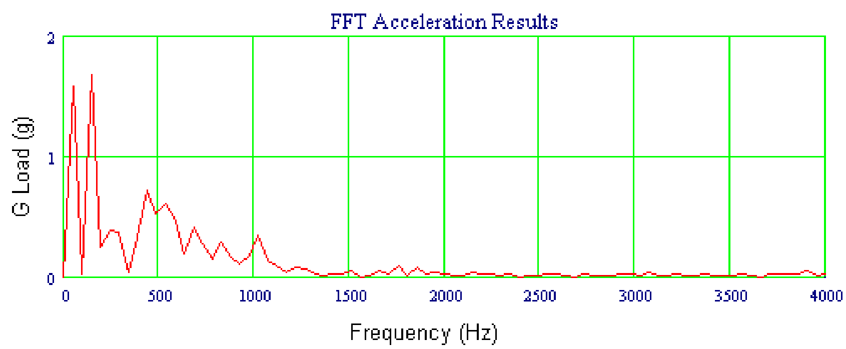

Fig.3 Acceleration vs. Frequency

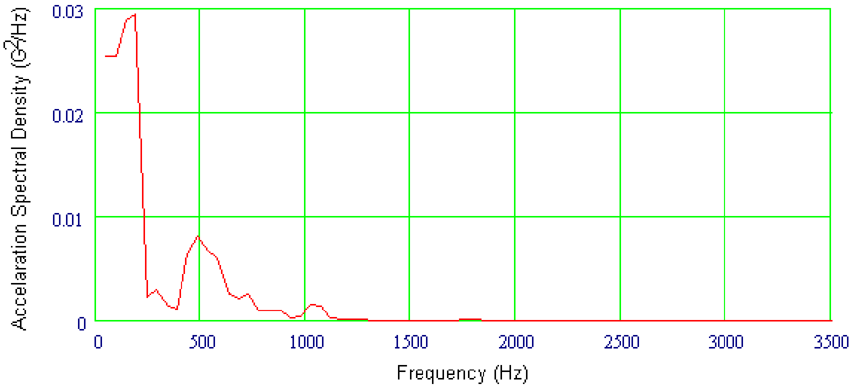

Fig.4 Acceleration Spectral Density vs. Frequency

2.2 Random vibration analysis

The material used for the structure is Solvay IXEF 1025/9008. The Young's modulus is 2,175,566 psi and the Material tensile strength is 25,381 psi. The damping ratio of the material given by Solvay is 0.2% at 250 Hz. By performing mode analysis of the full dual 101 EM assembly, the first five frequencies of the structure locate in the frequency range of 0 to105 Hz. The structure includes several contacting surfaces at the bolted joints, the damping of the material itself only accounts 10% of the system's damping ratio [3]. Therefore, the overall damping ratio of the system is about 2%. To find out how the damping ratio affects the stress level of the overall system, 1% damping ratio was also used for comparing standpoint.

FEA Model and Boundary Conditions

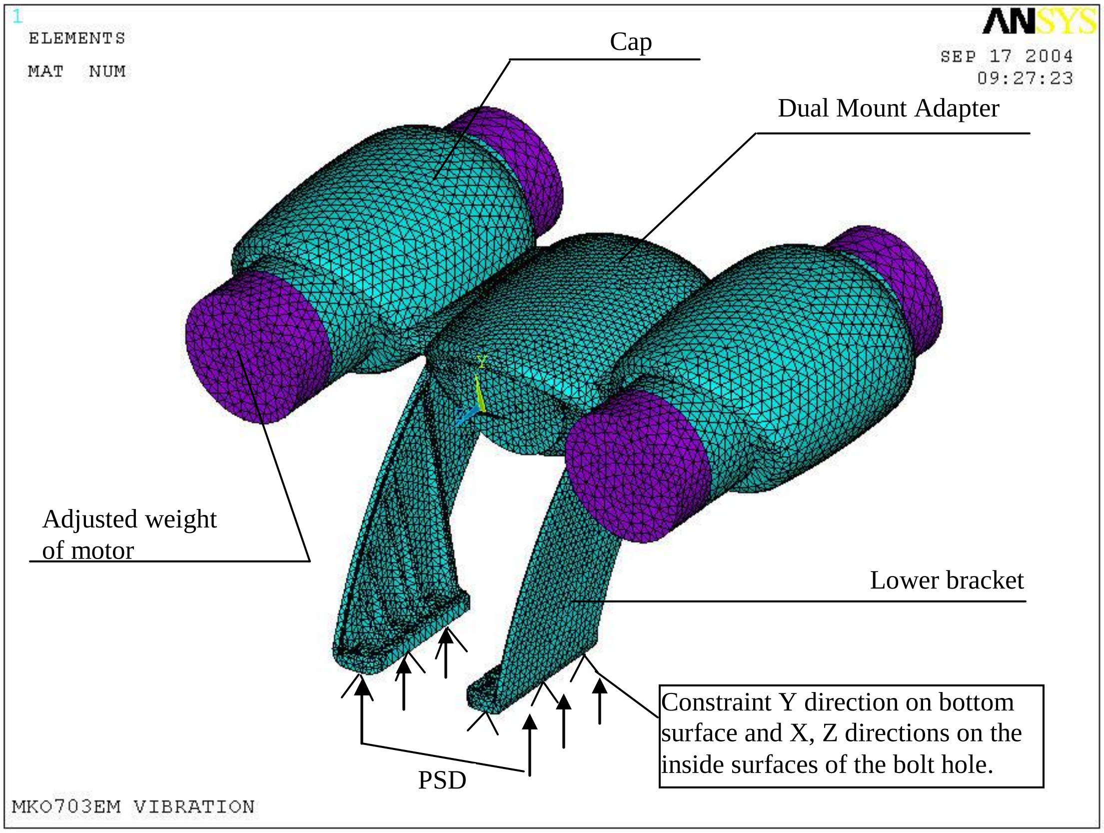

Fig.5 FEA model and boundary conditions

The cap and dual mount adapter are bolted to the lower bracket. The cap, dual mount adapter and lower bracket are made of Solvay material. To simplify the analysis, the actual weight and geometry of the motor is replaced by two adjusted weight of motors. The Young's Modulus of the adjusted weight of motor is 29,000ksi. The density of the adjusted weight of motor is calculated based on the weight of the actual motor and the volume of the adjusted weight.

FEA results (von-Mises stress)

Case 1: The overall damping ratio of the structure is 1%

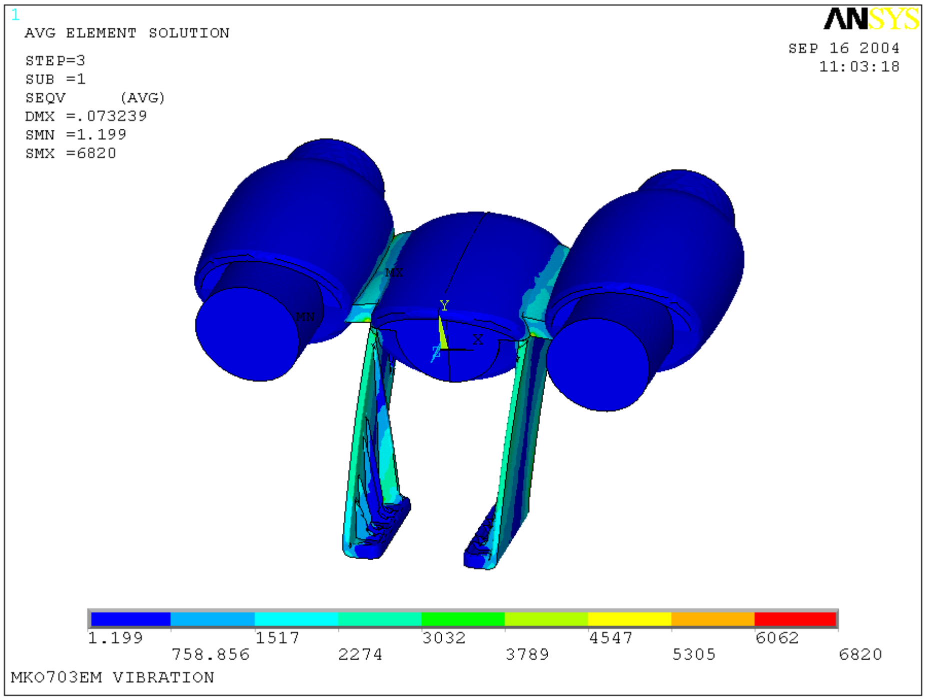

Fig.6 1σ stress distribution (case1, front view)

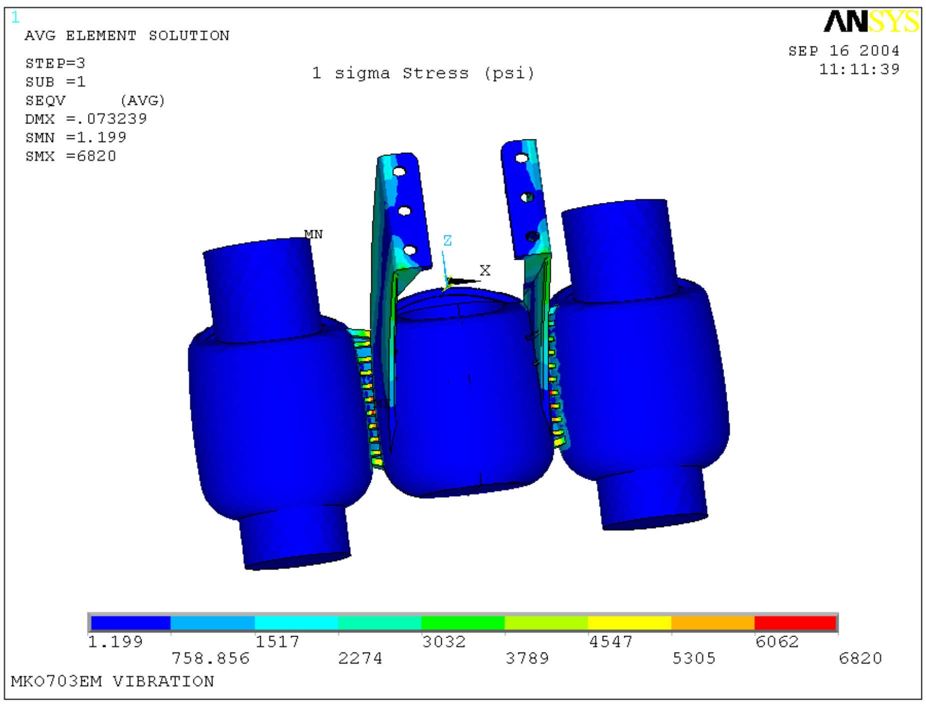

Fig.7 1σ stress distribution (case1, bottom view)

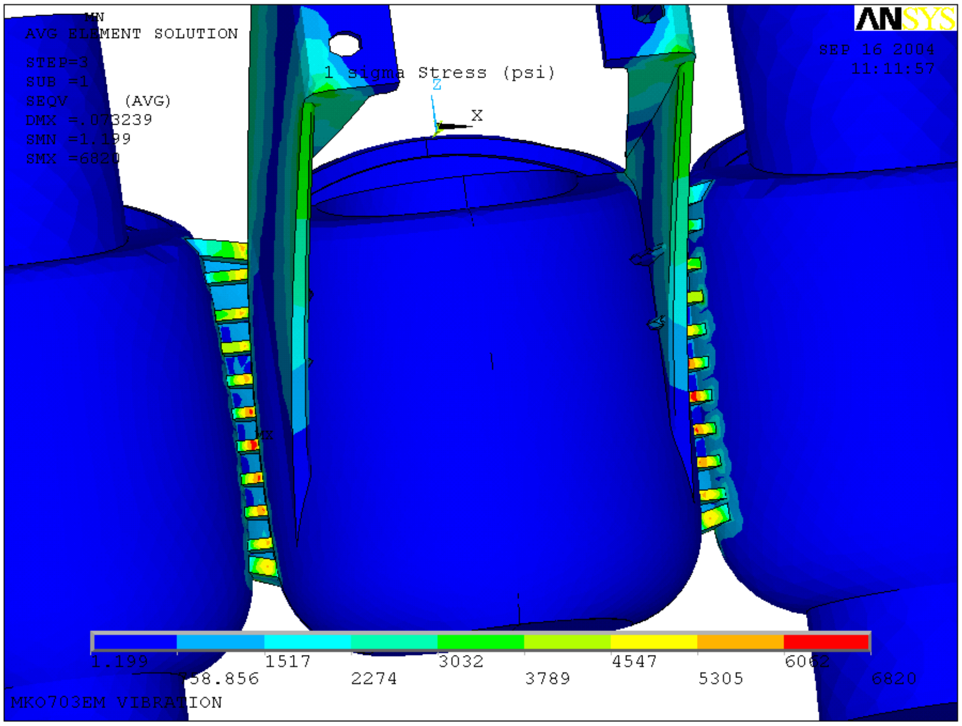

Fig.8 1σ stress distribution (case1, zoomed view, high stress area)

Case 2: The overall damping ratio of the structure is 2%

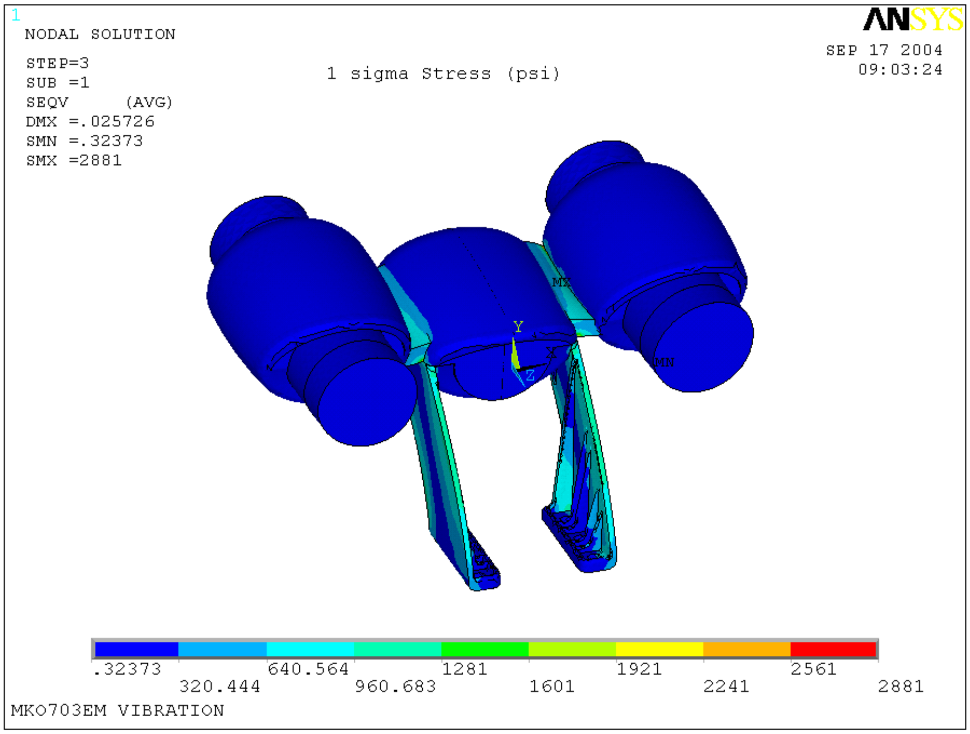

Fig.9 1σ stress distribution (case2, front view)

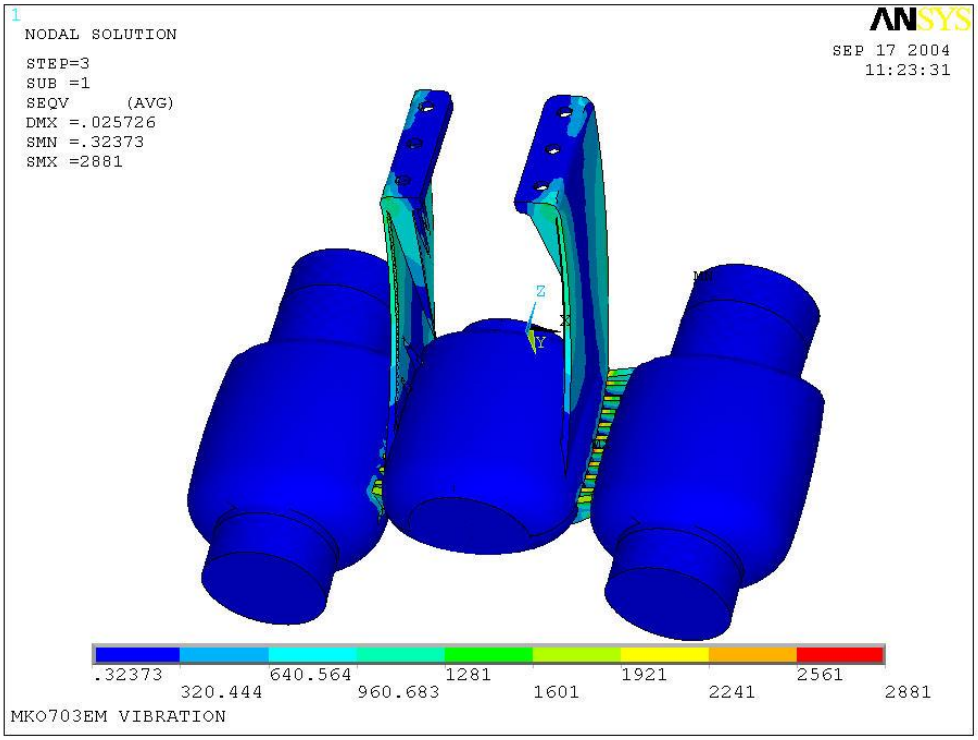

Fig.10 1σ stress distribution (case2, bottom view)

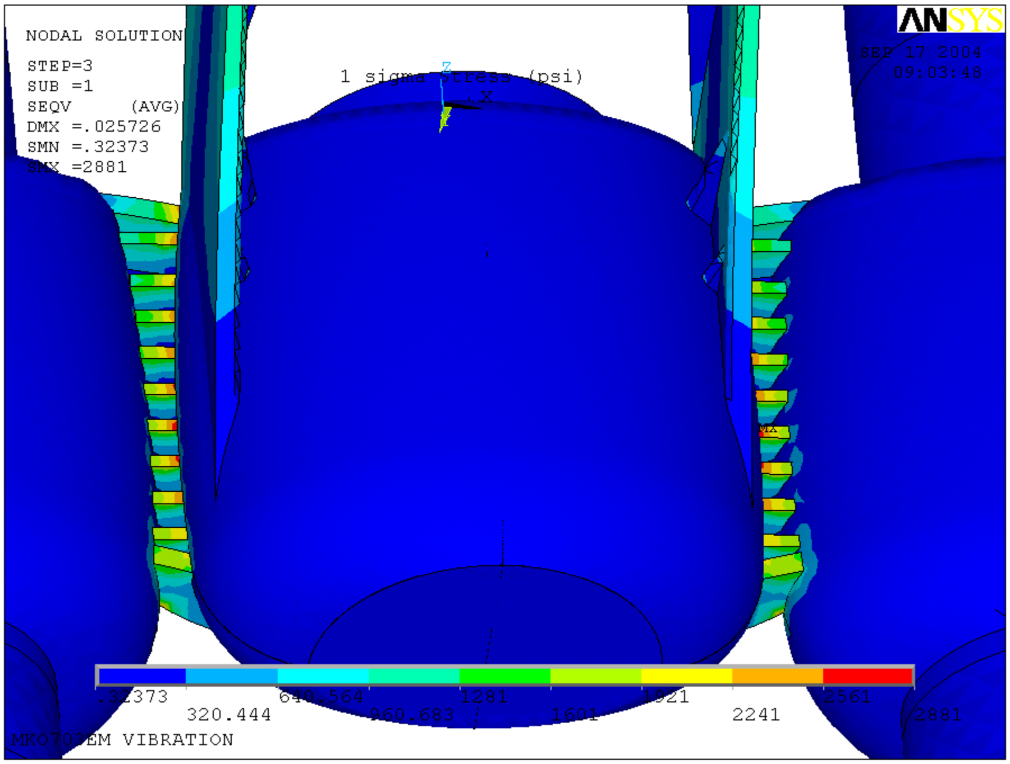

Fig.11 1σ stress distribution (case2, zoomed view, high stress area)

Discussing of the FEA results

By using Gaussian distribution [4], 1σ values occur 68.3% of the time, 2σ values occur 27.1% of the time and 3σ values occur 4.33% of the time [4]. To consider worst case scenario, assuming the 1σ stress shown in Fig.8, 1σ is cycling from maximum 2881 psi (tensile) to minimum –2881psi (compressive) at a frequency of 105 Hz (the fifth frequency mode) and mean stress is equal to 0, therefore, the alternating stress is equal to the 2881 for 1σ stress. From the S-N fatigue curve (Fig. 12), one can find the 1σ, 2σ, and 3σ stresses vs. number of cycles in the Table 1 and 2. One can calculate the time for each stress level using formula (1) to (3).

$$1\sigma \text{ time} = \text{Number of cycles} / 105 \text{ Hz} / 68.3\% \quad (1)$$
$$2\sigma \text{ time} = \text{Number of cycles} / 105 \text{ Hz} / 27.1\% \quad (2)$$
$$3\sigma \text{ time} = \text{Number of cycles} / 105 \text{ Hz} / 4.33\% \quad (3)$$

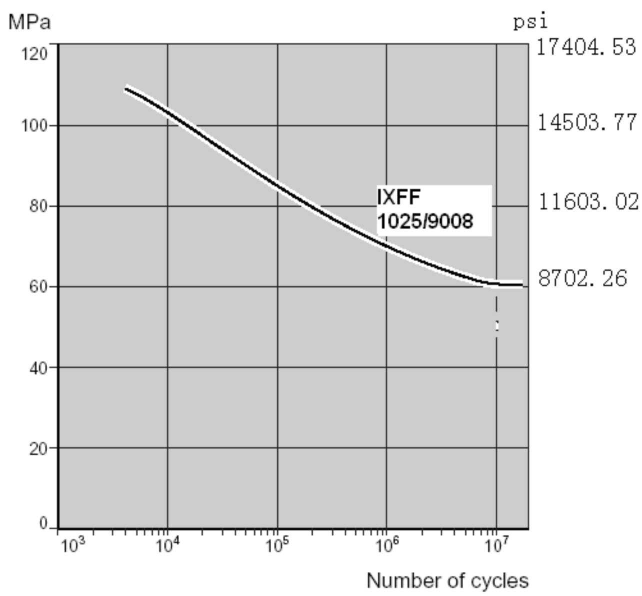

Fig. 12 S-N curve of material IXFF 1025

Table 1: Fatigue Calculation of the stress (Damping Ratio 0.01)

	Stress (psi)	Number of cycles	time of Vibration of 105 Hz
1σ	6,820	infinite	infinite
2σ	13,640	20,000	12 minutes
3σ	20,460	Less than 3000	Less than 11minutes

Table 2: Fatigue Calculation of the stress (Damping Ratio 0.02)

	Stress (psi)	Number of cycles	time of Vibration of105 Hz
1σ	2,881	infinite	infinite
2σ	5,762	infinite	infinite
3σ	8,640	$2X10^7$	1222 hours

The 101 dual EM assembly will not fail on the shaker table for 8 hours, with the input of the MIL-STD 810F curve, will be highly dependent on the actual damping ratio of the assembly. Since the actual damping ratio is close to 2%. The structure is safe for random vibration testing.

3. Conclusion

1. The damping ratio of the assembly has a role affect on the stress level as well as the life cycle of the overall system.
2. The successful simulation of random vibration of 101 dual EM assembly proves that the finite element method provides a useful and cost-effective tool for the investigation of the random vibration process involves complicated structures.

Reference

1. Anon, ANSYS User's Manual Version 8, ANSYS, INC., Canonsburg, PA.
2. Anon, MathCAD 7 User's Guide, MathSoft, INC., Cambridge, MA.
3. Online document:http://www.mfg.mtu.edu/cyberman/machtool/machtool/vibration/damping.html , Department of Mechanical Engineering at Michigan Technological University
4. Dave S. Steinberg, Design Electronic Equipment for Random Vibration and Shock, handout, June, 1982.