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Avionics Structural Analysis Case Study

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Introduction

Electronics designed for the aerospace industry must standup to rigorous shock, vibration and temperature environments. Before the products are deemed flight worthy, they must adhere to strict flight substantiation or air-worthiness requirements, be verified through analysis, and then qualified through testing. This study walks through a typical structural evaluation of an avionics chassis designed for an aerospace application, briefly illustrating each one of these analyses types:

- Frequency Analysis
    - Modal analysis
    - Harmonic analysis
    - Octave Rule evaluation
- Random Vibration Analysis
    - Acceleration Spectral Density (ASD) inputs
    - Stress and Deflection
    - Miner's Cumulative Damage
    - Circuit Card Assembly (CCA) Component Analysis
- Shock Analysis
    - Response Spectrum
    - CCA Component Analysis

Analysis Setup

The first steps in any product structural analysis are understanding the environment and then defining the analysis model.

Understanding the environment: Let us consider the environment of a space flight vehicle. For this case, NASA has developed a set of test standards that can be used as an analysis baseline. These standards can be found at https://standards.nasa.gov/, and more specifically, this study will reference the GSFC-STD-7000A standard for General Environmental Verification developed at the Goddard Space Flight Center. This standard provides guidelines for analysis, test, qualification, and acceptance of products and electronics used for this space flight vehicle application.

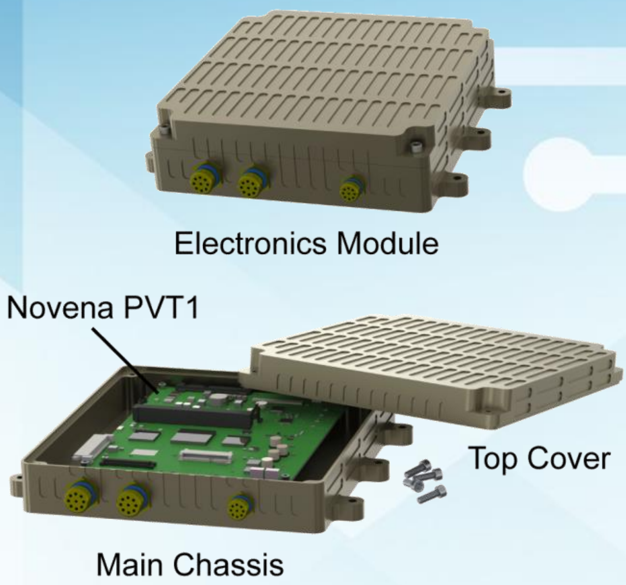


Figure 1. Avionics Module

Defining the analysis model: The analyst should understand the objective to be accomplished, the inputs and outputs, and then modify the CAD data to ensure that the mesh is optimized relative to the fidelity needed in the solution. The CAD model can often be simplified by defeaturing, idealization, and clean-up. Components and features that are not considerably affecting the stiffness or the mass of the system should be removed for efficiency. In this case, our avionics chassis has been reduced to the following analysis model comprised of a machined AL 6061-T6 chassis, defeatured connectors, and a CCA populated with components of concern as shown in Figure 2.

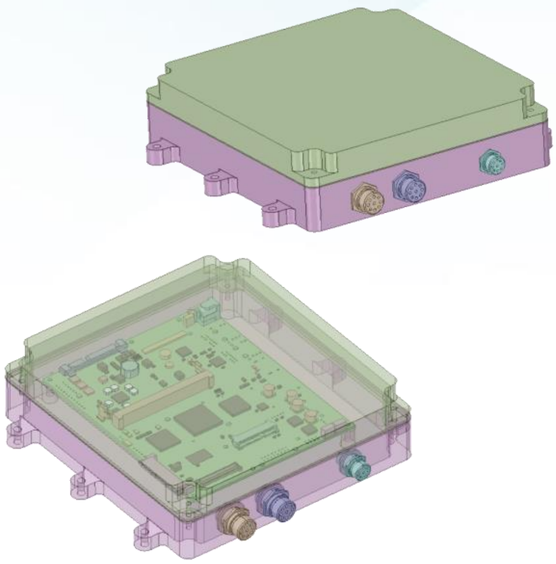


Figure 2. Analysis Model

Frequency Analysis

Modal Analysis

The next step is to determine how the system might be excited during a dynamic loading event by performing a modal analysis. The model is constrained in the analysis environment with supports through the bolted holes to simulate the actual use case environment while on the vehicle. A modal analysis is performed to extract the primary resonant frequencies and their corresponding mode shapes. We've limited the frequency range to 2000 Hz for this analysis to reduce computation time. The primary modes and frequencies of the CCA are shown in Figures 3-5.

The primary modes and frequencies of the avionics chassis are shown in Figures 6-8.

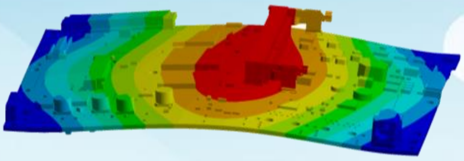


Figure 3. CCA 1st Mode - 160Hz

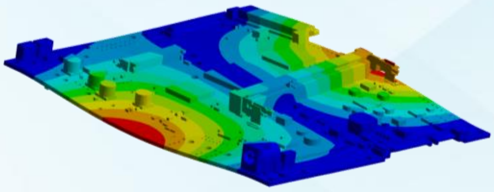


Figure 4. CCA 2nd Mode - 330Hz

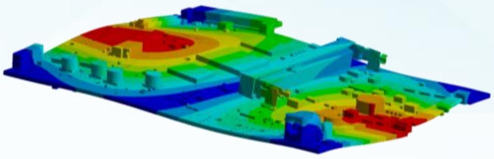


Figure 5. CCA 3rd Mode - 425Hz

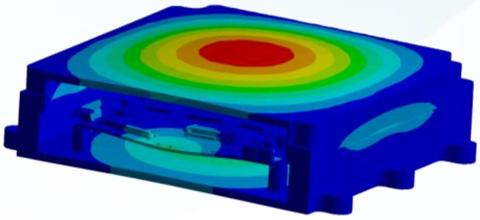


Figure 6. Chassis 1st Mode - 970Hz

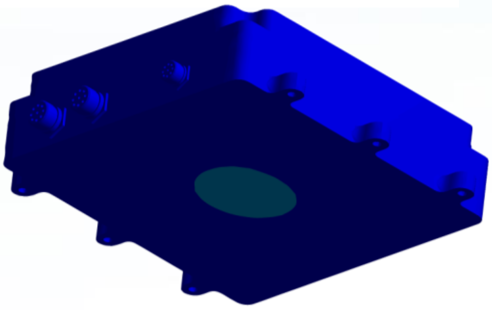


Figure 7. Chassis 2nd Mode - 1140Hz

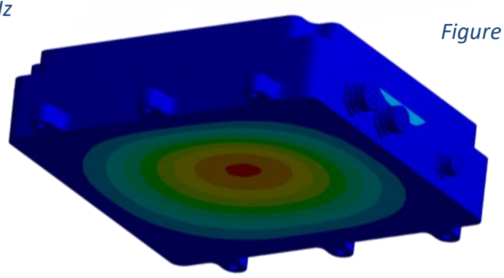


Figure 8. Chassis 3rd Mode - 1150Hz

Frequency Analysis

Modal Analysis (continued)

For the sake of brevity, we will assume that the shock and vibration input direction is in a single, vertical axis that is orthogonal to the top of the avionics chassis as this will be the direction that produces the most severe stresses and deflections for this application.

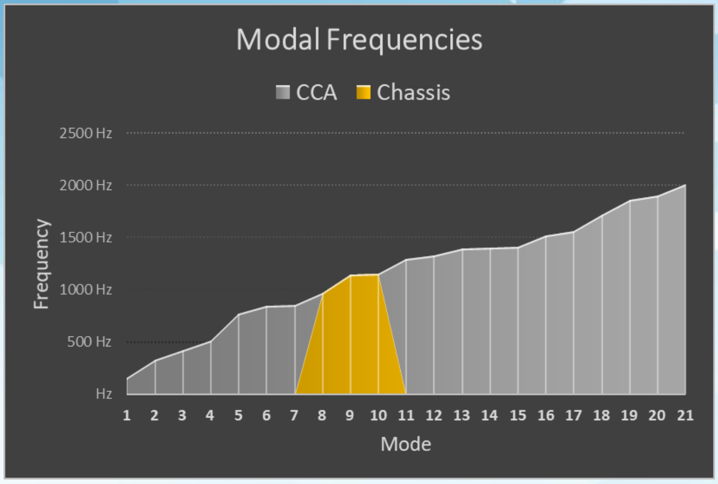


Figure 9. Modal Frequencies of CCA & Chassis to 2kHz

A graphical representation of the modes is shown in Figure 9. The modes that will most likely be excited by the input for both the CCA and the chassis will also have the highest mass participation along that same axis. A simple definition for mass participation is percent of mass moving in each mode. Generally, a majority of the structural mass will be represented in the first 10 to 20 modes, as is the case for the CCAs. The chassis cumulative mass participation does not rise above ~40% within the first 20 modes due to the high stiffness and robust mounting. In most cases the structural dynamic behavior is adequately simulated by the lowest frequency modes. For a uniform dynamic load (equal across the frequency spectrum), stresses and deflections are inversely proportional to the squared value of the model frequency. Thus, higher frequency modes contribute little to the overall structural dynamic response, unless the dynamic environment also has much (exponentially) higher acceleration levels at higher frequencies.

For this analysis, the mass participation relative to the modes is shown Figure 10. In looking at the data, one can see that the cumulative mass participation of the CCA in this axis exceeds >95% in the first 2000 Hz, while the bulk of the participation from the chassis doesn't start peaking until well beyond the 2000 Hz initial sweep. What this implies is that the chassis has much greater stiffness than the CCA, which is to be expected for this analysis model.

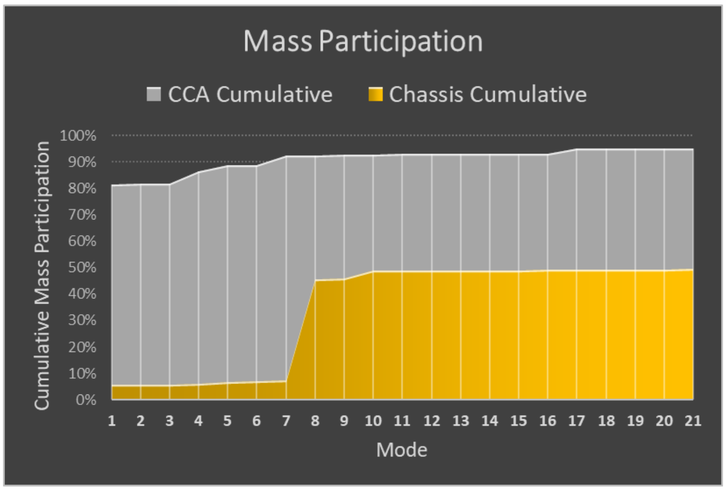


Figure 10. Cumulative Mass Participation

Frequency Analysis

Harmonic Analysis

Another beneficial analysis that visually shows how the system will react to a given input is a harmonic analysis. Assuming a damping (ζ) of 5%, for purposes of simplicity and a general rule of thumb for mechanical systems, if one excites the system with a base motion input of 1G, then the output of the harmonic analysis would indicate the amplification, or queuing, of the system. The output extracted from the main processor in the center of the CCA is shown in Figure 11.

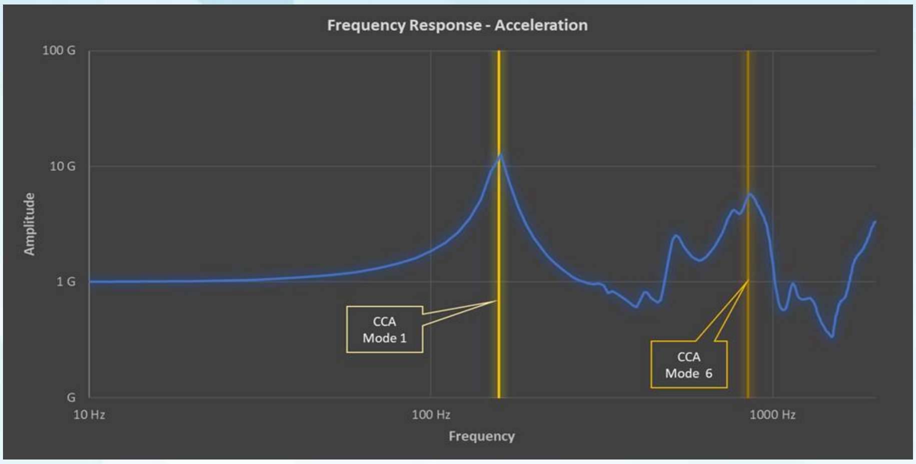


Figure 11. Acceleration Frequency Response of ARM Processor Chip

Looking at the harmonic analysis output, we can see that the peaks in the acceleration plot coincide with the modal frequencies, especially with respect to the CCA. In particular, one can see that the lowest mode of the CCA is being excited at the resonant frequency of 160 Hz resulting in amplification from the base motion input to an amplitude of approximately 12 G with a 1 G sinusoidal input. In fact, each of the subsequent peaks corresponds to a CCA mode shape and does not appear to significantly couple with the chassis modes. The amplification above the 1 kHz range is not generally an immediate cause of concern since damage producing deformations of the design are inversely proportional to the square of the frequency, as illustrated in Figure 12.

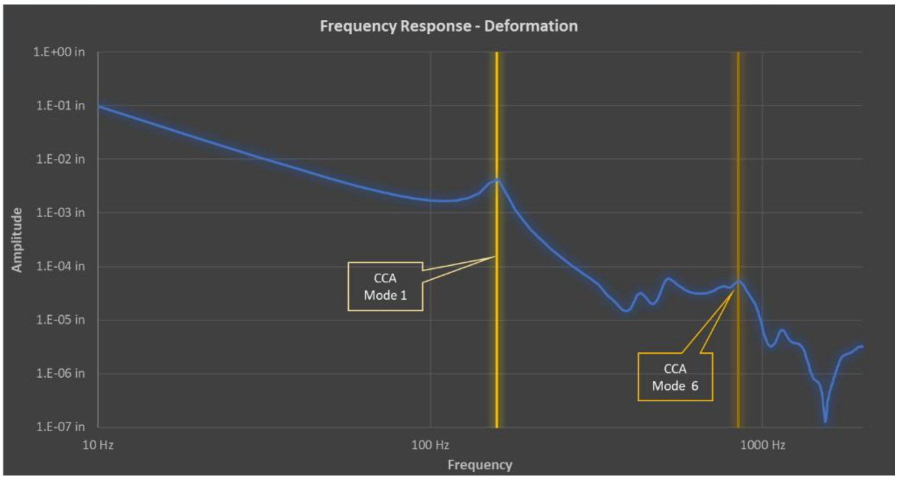


Figure 12. Deformation Frequency Response of ARM Processor Chip

Frequency Analysis

Octave Rule

The next major design criterion to evaluate the analysis model against is the Octave Rule. A dynamic coupling exists in a system with more than one degree of freedom (DOF). The CCA and its support structure can be considered as a two DOF system, with the support structure being an input to the response of the CCA. If the natural frequency of the support structure and the CCA are close to one another, then the CCA response can be greatly magnified. To prevent this from occurring in the design, their natural frequencies should be well separated, by an octave – or a factor of two, from each other to prevent this dynamic coupling from amplifying the response.[1] In this case, the reverse octave rule applies since the chassis stiffness is much greater than the CCA stiffness. To conform to the reverse octave rule, the natural frequency of the CCA must be at least a factor of 2 less than that of the support structure. The fundamental natural frequencies of the CCA and of the chassis are 160 Hz and 970 Hz, respectively – representing a ratio of 1:6.1 and is more than adequate to satisfy this design criteria.

Random Vibration Analysis

Moving beyond the frequency analysis of the system and the Octave Rule consideration, we consider the random vibration environment. As previously mentioned, the GSFC-STD-7000A provides a minimum Qualification specification in the form of an Acceleration Spectral Density (ASD). The specification is shown below for payloads weighing less than 50 lbs.[2]

Table 2 4 3
Generalized Random Vibration Test Levels
Components (ELV)
22 7 kg (50 lb) or less

Frequency (Hz)	ASD Level (g^2/Hz) Qualification	Acceptance
20	0 026	0 013
20 50	+6 dB/oct	+6 dB/oct
50 800	0 16	0 08
800 2000	6 dB/oct	6 dB/oct
2000	0 026	0 013
Overall	14 1 G_{rms}	10 0 G_{rms}

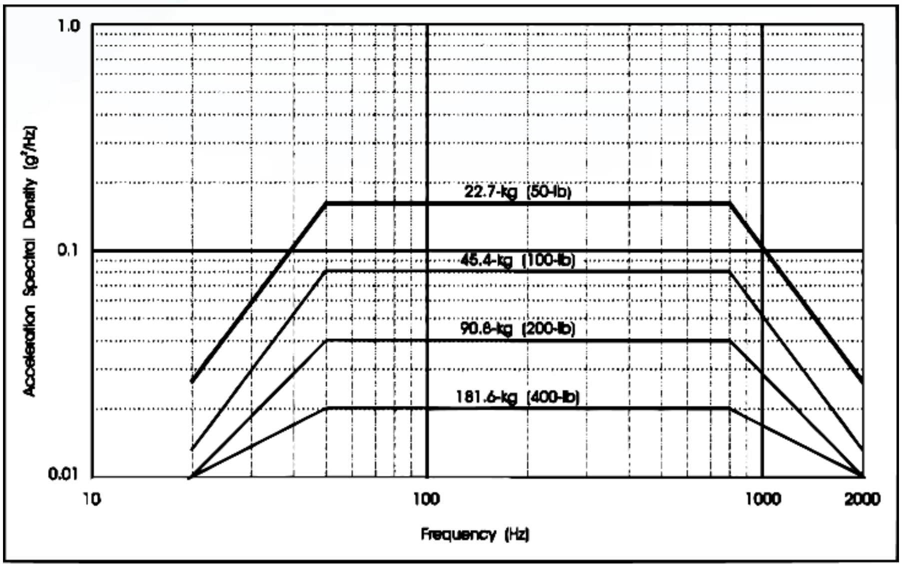
The acceleration spectral density level may be reduced for components weighing more than 22 7 kg (50 lb) according to

	Weight in kg	Weight in lb	
dB reduction	= 10 log(W/22 7)	10 log(W/50)	
ASD(50 800 Hz)	= 0 16•(22 7/W)	0 16•(50/W)	for protoflight
ASD(50 800 Hz)	= 0 08•(22 7/W)	0 08•(50/W)	for acceptance

Where W = component weight

The slopes shall be maintained at + and 6dB/oct for components weighing up to 59 kg (130 lb) Above that weight the slopes shall be adjusted to maintain an ASD level of 0 01 g^2/Hz at 20 and 2000 Hz

For components weighing over 182 kg (400 lb) the test specification will be maintained at the level for 182 kg (400 pounds)



Random Vibration Analysis

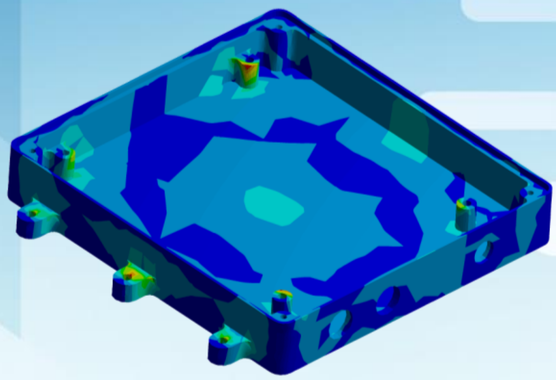


Figure 13. Stress Plot of Avionics Chassis

Applying this ASD as an input into the analysis model, the 3σ stress within the avionics chassis can be extracted and is presented in Figure 13.

In this study, the chassis happens to be very stiff and likely over designed for this application. The magnitude of the 3σ stress, predicted to be approximately 6 ksi, is much lower than the endurance limit of the material and should not be at risk of failure from the prescribed 14.1 G_{rms} input prescribed.

Fatigue and Miner's Cumulative Damage Calculation

While our study predicted that the chassis would easily survive the vibration environment, let us assume that the 3σ stress on the aluminum chassis is probed and found to be approximately 13.5 ksi. To determine if this value constitutes a possible fatigue failure, a Miner's Damage calculation can be performed based on the fatigue properties of the aluminum material and the expected duration of the testing. We will assume that this chassis is manufactured from AL 6061-T6, which has been extensively documented with respect to its endurance limits and fatigue life – normally found on an S-N curve, shown in Figure 14.[3]

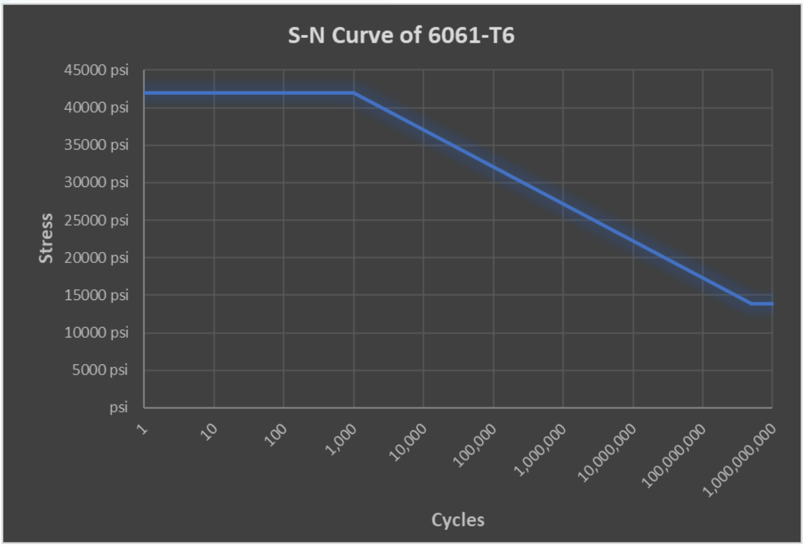


Figure 14. S-N Curve for AL 6061-T6

Random Vibration Analysis

Fatigue and Miner's Cumulative Damage Calculation (Continued)

The corresponding Miner's Damage calculation will use the chassis' first fundamental mode for the cycle time and 3.8 hr total duration. The 1σ, 2σ, and 3σ stresses (13.5ksi assumption) for the damage calculation are used to determine the R value.[3] The R value should remain below the value of 1 to indicate that a failure does not occur during the 3.8 hrs time limit. Thus, as shown below:

$$R_n = \frac{n_1}{N_1} + \frac{n_1}{N_1} + \frac{n_1}{N_1} + \cdots = 1.0$$

$$1\sigma\ n_1 = \frac{970\ cycles}{s} \times \frac{3600\ s}{hr} \times 3.8\ hrs \times 68.3\% = 8.94\ \times 10^6\ cycles$$

$$2\sigma\ n_2 = \frac{970\ cycles}{s} \times \frac{3600\ s}{hr} \times 3.8\ hrs \times 27.1\% = 3.55\ \times 10^6\ cycles$$

$$3\sigma\ n_3 = \frac{970\ cycles}{s} \times \frac{3600\ s}{hr} \times 3.8\ hrs \times 4.33\% = 5.63\ \times 10^5\ cycles$$

$$1\sigma\ N_1 = 1000 \times \frac{42{,}000\ psi}{4{,}200\ psi}^{6.4} = 1.62\ \times 10^9\ cycles_{to\ failure}$$

$$2\sigma\ N_2 = 1000 \times \frac{42{,}000\ psi}{6{,}750\ psi}^{6.4} = 1.21\ \times 10^8\ cycles_{to\ failure}$$

$$3\sigma\ N_3 = 1000 \times \frac{42{,}000\ psi}{13{,}500\ psi}^{6.4} = 1.43\ \times 10^6\ cycles_{to\ failure}$$

$$R_n = \frac{8.94\ \times 10^6}{1.62\ \times 10^9} + \frac{3.55\ \times 10^6}{1.21\ \times 10^8} + \frac{5.63\ \times 10^5}{1.43\ \times 10^6} + \cdots = .429$$

$$Margin = \frac{1}{R_n} = 2.3$$

$$Time_{Fail} = 2.3\ \times 3.8\ hrs = 8.7\ hrs$$

Using a 3σ stress limit of 13.5ksi, the Miner's rule suggests that chassis would have survived the duration of qualification testing with a margin of 2.3 and a predicted failure time of 8.7 hrs.

Random Vibration Analysis

CCA Component Failure Evaluation

To evaluate CCA survival, an empirical formula is used to determine if any of the major components are expected to fail. The formula for the allowable deflection of the CCA is[1]:

$$Z_{3\sigma\ \text{limit}} = \frac{0.00022\ B}{C h r \sqrt{L}} \quad \text{inches}$$

B	=	length of the circuit board edge parallel to the component, inches
L	=	length of the electronic component, inches
h	=	circuit board thickness, inches
r	=	relative position factor for the component mounted on the board, $0.5 \leq r \leq 1.0$
C	=	Constant for different types of electronic components $0.75 \leq C \leq 2.25$

The 3σ corresponding deflection was an output parameter from the random vibration analysis. The maximum deflection appears to occur in the same manner as the first fundamental mode shape; bending through the center of the CCA as shown in Figure 15.

The predicted 3σ deflection is 0.031 inches, occurring approximately at the center of the CCA. In this example, we will focus on the ARM i.MX6 processor on the CCA and determine if this chip is at risk of failure.[4]

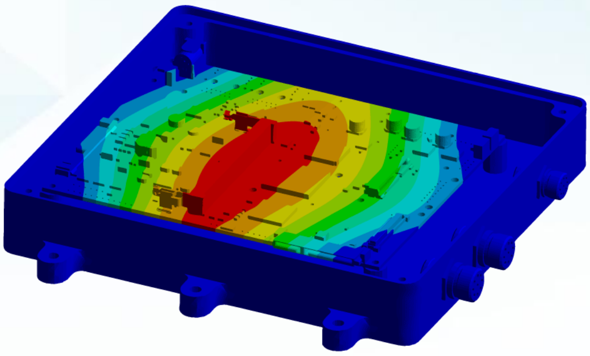


Figure 15. Deformation Plot of Avionics Chassis

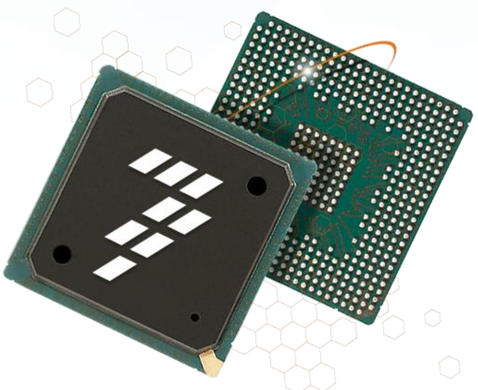


Figure 16. i.MX6 Processor - Image from NXP

These particular processors have a surface mounted ball grid array (BGA) type package construction, which Steinberg characterizes with a C value of 1.75.[1] Inputting the rest of the values, the allowable 3σ deflection in this model is:

$$Z_{3\sigma\ limit} = \frac{.00022\ (5.9)}{1.75\ (.065)(1)\sqrt{.909}} = 0.013\ inches$$

Random Vibration Analysis

CCA component failure evaluation (Continued)

Based on the empirically derived formula, the maximum allowable deflection of the CCA is 0.013 inches. The predicted 3σ deflection is approximately 2.4 times the allowable deflection. This indicates that the ARM Processor is at risk of failure.

There are several methods to employ in order to address this failure risk. Depending on the state of the design, the most straight forward method would be to add additional mounting points towards the center CCA to limit the deflection induced by the random vibration input. However, this method will also increase the natural frequency of the CCA and possibly start violating the Octave Rule as previously evaluated. Iteration will be needed, but this adequately demonstrates the importance of analysis during the design process. For this white paper, we will assume that this analysis proved positive and will move to the next study.

Shock Analysis

Response Spectrum

The NASA Standard also provides a Shock Response Spectrum (SRS) to evaluate the analysis model for space flight vehicles.[2] This SRS is shown below:

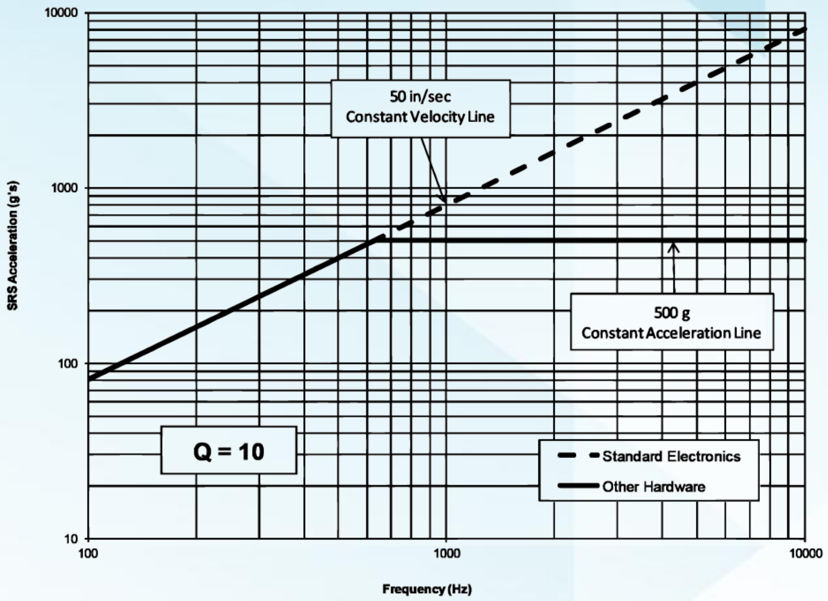


Figure 2.4-1 Shock Response Spectrum (SRS) for assessing Component Test Requirements

Rather than approximating static loads from this 1 DOF SRS input using the Dynamic Design Analysis Method (DDAM), a response spectrum analysis is performed using the SRS input and the resulting stress of the chassis is shown in Figure 17.

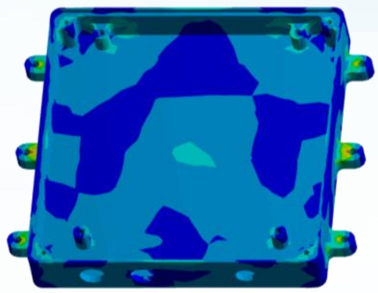


Figure 17. Stress Plot of Avionics Chassis

The maximum stress in the housing was predicted to be approximately 11.5 ksi. While this is much higher than those previously reported during the random vibration study, we can assume that the shock input for this environment is limited to less than 200 occurrences. Therefore, the maximum stress value can be evaluated against the Ultimate / Yield strengths of the material. The Factor of Safety is calculated below along with the minimum Safety Factor required for flight hardware design applied to limit loads, required for Margin of Safety calculations[3]:

$$FoS_{yield} = \frac{35,000\ psi}{11,500\ psi} = 3.04 \qquad FoS_{ultimate} = \frac{42,000\ psi}{11,500\ psi} = 3.65$$

$$MS_{yield} = \frac{35,000\ psi}{11,500\ psi * 1.25} - 1 = 1.43 \qquad MS_{ultimate} = \frac{42,000\ psi}{11,500\ psi * 1.4} - 1 = 1.61$$

Shock Analysis

Response Spectrum (Continued)

The stress induced by the shock event has a very large Factor of Safety against the ultimate and yield strengths of the material. The Margin of Safety is also exceedingly high, suggesting that there is plenty of room for weight reduction. We could have addressed these items if we were to iterate to solve the random vibration CCA failure indicated above.

CCA component evaluation

Similar to the random vibration evaluation, the best method to evaluate the CCA for failure over shock is to return to Steinberg's empirical data. Steinberg states that the maximum desired displacement of a CCA during a shock event is approximately 6X the displacement calculated for the random vibration threshold.[1] The displacement plot from the response spectrum analysis is shown in Figure 18.

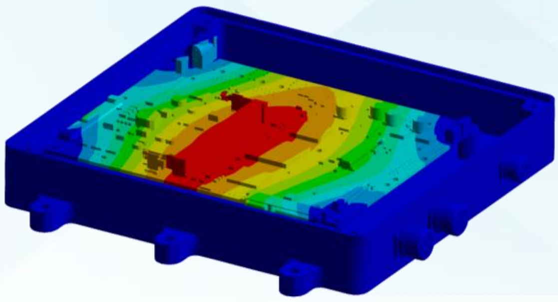


Figure 18. Maximum Deformation Plot of Avionics Chassis

The maximum displacement from the response spectrum analysis is predicted to be 0.066". The maximum allowable displacement and corresponding Factor of Safety is as follows:

$$Z_{shock} = \frac{.00132\ (5.9)}{1.75\ (.065)(1)\sqrt{.909}} = 0.076\ inches$$

$$FoS_{Zshock} = \frac{.076\ inches}{.066\ inches} = 1.15$$

While the deformation of the CCA exceeded the random vibration threshold, the deflection limit over shock is much more forgiving. Based on the results above, both the CCA and the chassis are expected to successfully pass the shock testing.

Conclusion

As the industry focuses on getting lighter and smaller all the while obtaining better performance, following good design practices is critical. As we've seen from this study, even though the chassis is over designed for this application, the results from the analysis of the CCA shows concern for reliability and indicates that failures are likely to occur.

Iteration is a natural function of design, but in almost all cases, it is cheaper and more expedient to identify the problems through analysis than it is to correct the failures in the field. Performing analyses such as those described in this document to cap off the design phases of your project provide opportunities to predict risk areas allowing iterative improvement without the expense of test and discovery phases. In turn, your product development cycle is condensed and the end result is a successful design that can be qualified and deemed flight worthy.

References

The methods outlined here are universally accepted by the avionics industry and can be found in the following references:

[1]Steinberg, Dave. Vibration Analysis for Electronic Equipment. John Wiley & Sons, 2000
[2]https://standards.nasa.gov/
[3]Juvinall and Marshek. Fundamentals of Machine Component Design, John Wiley & Sons, 2000

The CCA model used in this study is the Novena PVT1 from the following:

[4] https://www.kosagi.com/w/index.php?title=Novena_Main_Page

FEA Tool: ANSYS
CAD Tool: SolidWorks