IAC-21-C2.3.9.x64779

Structural and Vibrational Analysis of Structure of a 3U CubeSat

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Abstract

This paper presents the analysis of the static and dynamic performance of the structure of a 3U CubeSat designed by the students of Team Anant, BITS Pilani by Finite Element Method software. The loads are investigated and checked according to the strength and stiffness requirements specified by the candidate launchers. The material and dimensions of the structure of the satellite are following the requirements imposed by the P-POD. The complex model was defeatured to reduce the running time for the simulations. Boundary conditions used for simulation of the P-POD and modelling of bolts in the model using the software are described. The static response of the structure was simulated for the quasi-static launch loads along different orientations due to uncertainty in the final orientation of the satellite in the Launch Vehicle. Modal analysis was performed to check for compliance with the stiffness requirements in both longitudinal and lateral directions. Simulations were performed for the Sinusoidal and Random Vibration Test Levels specified by the launch provider. Fatigue life analysis of the structure was performed to ensure that it will successfully bear the launch loads.Topology optimization was done to reduce the stresses acting on key components. The structure of the 3U CubeSat meets all the loading requirements and it is ensured that the stresses and deflections in the structure are well within acceptable levels.

Acronyms/Abbreviations

P-POD : Poly-Picosatellite Orbital Deployer
PSLV: Polar Satellite Launch Vehicle
ISRO: Indian Space Research Organisation
ADS: Antenna Deployment System

1. Introduction

This paper aims to provide an insight into the vibrational simulations carried out on our 3U Cubesat. Our team, known as Team Anant, aims to launch a 3U Cubesat into LEO for multispectral imagery over specific forest areas for data collection.

Vibration analysis is necessary to find out all the loads acting on our satellite and the stresses due to each of these loads. In order to prevent damage to our satellite, certain simulations are carried out in order to calculate these stresses on our satellite, and the satellite structure is optimized to get a final model capable of withstanding these harsh conditions

Section 2 lays out the fundamental understanding of vibration theory. Section 3 discusses the parameters applied in our simulations and Section 4 is about applying these simulations in our paper. Section 5 acts as our conclusion and future work. Appendix A (Images) has the results of all simulations performed and Appendix B contains a flowchart linking all our tasks for vibrations analysis of a satellite

2. Simulation conditions

Space payloads experience a harsh vibration environment during launch. The payload will be exposed to a vibration environment which simulates if our payload will be able to survive the rigors of our launch event

The vibrations simulated using ANSYS include Quasi-static , Modal, Random and Harmonic tests. The satellite may experience shocks, but shock testing is not covered during simulation and will be checked during physical vibration simulation of our engineering model.

2.1 Random Vibration testing

Random vibrations are generated at the spacecraft base by the operation of propulsion systems and acoustic vibrations of the adjacent structure. This involves testing our satellite to excitation at all frequencies in a particular bandwidth (5 Hz to 100 Hz). The amplitude of these vibrations is random with a Gaussian distribution. We cannot test our model at each frequency in a bandwidth as there is an infinite number of them. Hence we divide our bandwidth into narrow frequency bands.

2.2 Sinusoidal Vibration testing

The satellite is exposed to a sinusoidal excitation that sweeps across a frequency band or dwells upon a single frequency. The sine sweep rate could be linear (Hz per sec), or logarithmic (octaves per min).

Sine dwells are controlled to specific amplitudes which are specified in terms of either amplitude, velocity or displacement. This test allows us to determine effects due to resonance and to simulate environments that have dominant narrowband frequency components.

2.3 Shock testing

Shock testing simulates the effects of short duration, high amplitude phenomena such as pyrotechnic device detonation,high explosion shocks. This kind of testing does not provide us with stress or strain values. It only provides us with an idea of the damage potential of a particular shock wave.

3. Simulation procedure

3.1 Geometry

We made use of the educational license of Autodesk Fusion 360, a commercially available CAD software to create the actual design of the complete satellite. However, this model consists of intricate details such as chamfers, fillets and small holes, and using this model for the simulations would lead to any increase in computational time without contributing significantly to the simulation results.

Hence, a simplified model consisting of the primary, secondary and tertiary structures of the satellite was created by removing complex geometrical features. For the remaining components, point masses were used which replicated the effects of these components without utilising excess computational power. For every point mass, seven properties are to be clearly defined: The mass, The X, Y and Z coordinates of its centre of mass, The moments of inertia about an axis passing through the origin along the X, Y and Z axes.

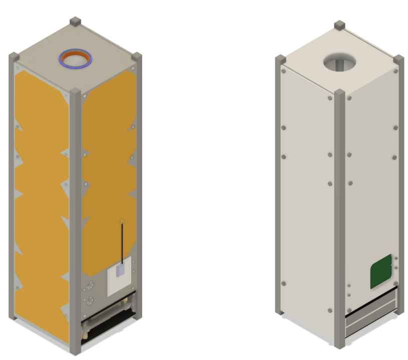


Fig. 1. Actual CAD (left) and Simplified CAD (right)

3.2 Material assignment

Upon importing the CAD model into ANSYS, the first step would be to ensure that materials are correctly assigned to each component. Different materials found in the database of ANSYS are assigned as follows:

- An Epoxy/Glass fiber laminate known as FR-4 is used for the PCBs.
- Brass is used for the PCB spacers.
.
- Acetal resin or POM is used for the doors of the Antenna deployment module.
- Aluminium 6061 alloy is used for all the remaining components.

3.3 Meshing

A mesh element size of 3.5 mm was used for the majority of components present in the structure. Certain components such as nuts, bolts and PCB spacers used a size of 1 mm owing to their small size. The mesh currently consists of 433451 nodes and 163351 elements.

Moreover, ANSYS provides some valuable tools to check mesh quality using the Mesh metrics option. The two main meshing parameters to check are Orthogonal quality and Skewness. Orthogonal quality measures the perpendicularity of elements. It ranges from 0 to 1, where a value of 0 is worst, and a value of 1 is best. Skewness determines how close to ideal a face or cell is. It also ranges from 0 to 1, where a value of 0 indicates an equilateral cell (best) and a value of 1 indicates a completely degenerate cell (worst). The given mesh has an average Orthogonal quality of 0.71791 and an average Skewness of 0.30812, which indicates a fairly decent mesh.

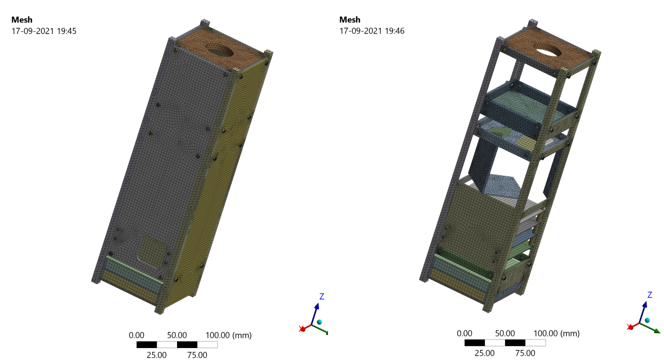


Fig. 2. Meshing of the components

.3.4 Launch vehicle conditions

ISRO has given the necessary loading conditions of the PSLV launch vehicle which will be used for carrying out the structural and vibrational simulations:

- Each rail of the satellite shall have a sufficient structural strength considering that the rail is subject to compression force at 46.6 N due to a preload from the backplate and mainspring of the deployer.

- The satellite should withstand a quasi-static acceleration of the following magnitudes during launch:

Table 1. Quasi-static loads

Direction	Acceleration
Longitudinal	9 g compression
	3.5 g tension
Lateral	+ 2 g

The worst case quasi-static acceleration was determined and it came out to be 13 g.

- The minimum fundamental frequency of the satellite should be no less than 100 Hz.

- The satellite should withstand the given random vibration environment of the launch vehicle:

Table 2. Random vibration loads

Frequency (Hz)	Acceleration (PSD)
20	0.002
110	0.002
250	0.034
1000	0.034
2000	0.009

- The satellite should withstand sine-equivalent vibrations as per the following table:

Table 3. Sinusoidal vibration loads

Direction	Frequency range (Hz)	Acceleration
Longitudinal	5-8	34.5 mm DA
	8-100	4.5 g
Lateral	5-8	24 mm DA
	8-100	3 g

A factor of safety of 2 is multiplied with all the forces and accelerations mentioned above before running the simulations.

3.5 Boundary conditions

The boundary conditions replicate the situation where the rails are in contact with the inner surface of the P-POD.

For the structural simulations, the following boundary conditions will be used:

- Displacement X = 0 will be applied to the ± X faces of rails.

- Displacement Y = 0 will be applied to the ± Y faces of rails.

- Displacement Z = 0 will be applied to the - Z face of rails.

The vibrational simulations follow all the above boundary conditions along with a Displacement Z = 0 applied to the + Z face of the rails as well.

In the structural simulations, both the compressive force and the quasi-static acceleration are applied simultaneously but in two different cases: One where they act in the same direction and the other where they are perpendicular to each other.. For the random vibration simulations, there are three cases where the PSD acceleration is applied along the X, Y and Z axes respectively. For the sinusoidal vibration simulations, there will be two cases: One in the frequency range 5-8 Hz and the other in the frequency range 8-100 Hz.

For all the simulations mentioned above, the pretension due to the fastening of the nuts and bolts will also have to be taken into account. A bolt pretension of 4 N will be applied to all the nuts and bolts present in the model. As there are a large number of contact regions in the model, they have all been set to 'Bonded' to reduce computational time. From the results, topology optimisation was carried out on the connectors, lens mounts and the ADS module.

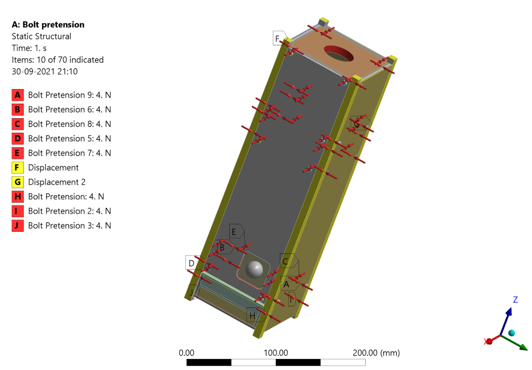


Fig. 3. Displacement supports and Bolt pretension

4. Simulation results

Table 4. Structural case 1 (compressive force and acceleration in the same direction)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	5.1405 MPa	0.4210 MPa	7.1090 MPa	1.3927 MPa
Lower lens mount	0.9163 MPa	0.1291 MPa	1.4749 MPa	0.3301 MPa
ADS module	1.4322 MPa	0.2188 MPa	1.6257 MPa	0.2560 MPa

Table 5. Structural case 2 (compressive force and acceleration perpendicular to each other)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	4.2574 MPa	0.2974 MPa	3.9885 MPa	0.7722 MPa
Lower lens mount	0.8387 MPa	0.1182 MPa	0.8665 MPa	0.1715 MPa
ADS module	2.2529 MPa	0.2381 MPa	3.5322 MPa	0.2985 MPa

Table 6. Modal frequencies

Mode number	Unoptimised case	Optimised case
1	126.45 Hz	111.15 Hz
2	184.69 Hz	123.56 Hz
3	199.74 Hz	134.09 Hz
4	235.84 Hz	134.17 Hz
5	268.2 Hz	163.74 Hz
6	275.47 Hz	184.14 Hz

Table 7. Random vibration case 1 (loading along X-axis)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	0.0125 MPa	0.0007 MPa	1.3754 MPa	0.2574 MPa
Lower lens mount	0.0795 MPa	0.0029 MPa	0.0192 MPa	0.0019 MPa
ADS module	0.2949 MPa	0.0057 MPa	0.1682 MPa	0.0028 MPa

Table 8. Random vibration case 2 (loading along Y-axis)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	0.0208 MPa	0.0012 MPa	1.3887 MPa	0.2548 MPa
Lower lens mount	0.0075 MPa	0.0008 MPa	0.0097 MPa	0.0016 MPa
ADS module	0.7690 MPa	0.0098 MPa	0.5759 MPa	0.0050 MPa

Table 9. Random vibration case 3 (loading along Z-axis)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	0.0629 MPa	0.0036 MPa	0.0245 MPa	0.0024 MPa
Lower lens mount	0.0337 MPa	0.0034 MPa	0.0279 MPa	0.0026 MPa
ADS module	0.4973 MPa	0.0158 MPa	0.2651 MPa	0.0050 MPa

Table 10. Sinusoidal vibration case 1 (5-8 Hz frequency range)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	0.0194 MPa	0.0011 MPa	0.8819 MPa	0.1356 MPa
Lower lens mount	0.0118 MPa	0.0011 MPa	0.0092 MPa	0.0013 MPa
ADS module	0.0525 MPa	0.0032 MPa	0.0932 MPa	0.0013 MPa

Table 11. Sinusoidal vibration case 2 (8-100 Hz frequency range)

Component	Unoptimised case		Optimised case	
	Maximum stress	Average stress	Maximum stress	Average stress
Upper lens mount	0.0382 MPa	0.0022 MPa	2.9069 MPa	0.4468 MPa
Lower lens mount	0.0250 MPa	0.0022 MPa	0.0296 MPa	0.0037 MPa
ADS module	0.1009 MPa	0.0063 MPa	0.4809 MPa	0.0066 MPa

5. Results and Conclusion

As a result of topology optimisation, the total mass of the satellite decreased from 2944 grams to 2525 grams, a reduction of 14.23%. Using the results obtained in the simulations, topology optimization is done and the stress values and modal frequencies are compared between the two models. The stresses from the structural simulations were comparatively higher compared to those of the vibrational simulations in both the unoptimised and optimised models. Moreover, the stresses acting on the satellite are well below the yield strength of the material. The lowest fundamental frequency of the satellite is also above 100 Hz in both the cases, thus satisfying this condition. The frequencies of the optimised model were lower compared to the unoptimised model.

However our project is still in the design phase and many components have been taken with many approximations. Once all the components are finalised, the model can be simulated with greater accuracy. The model will be subject to physical vibrational testing to verify the simulation results at a later stage.Our work would be quite helpful in planning out simulations for future Cubesat programs, such as in deciding simulation parameters, different types of simulations to be run, and launch conditions to be accounted for during simulations.

With more components of various subsystems being finalised, the model can be accurately designed and simulated. With the change of components, stresses may or may not change, depending on which we may need to optimize the structure again. For the final model, we will run physical vibration tests in shaker machines which will allow us to compare our results with the results obtained via simulation.

6. References

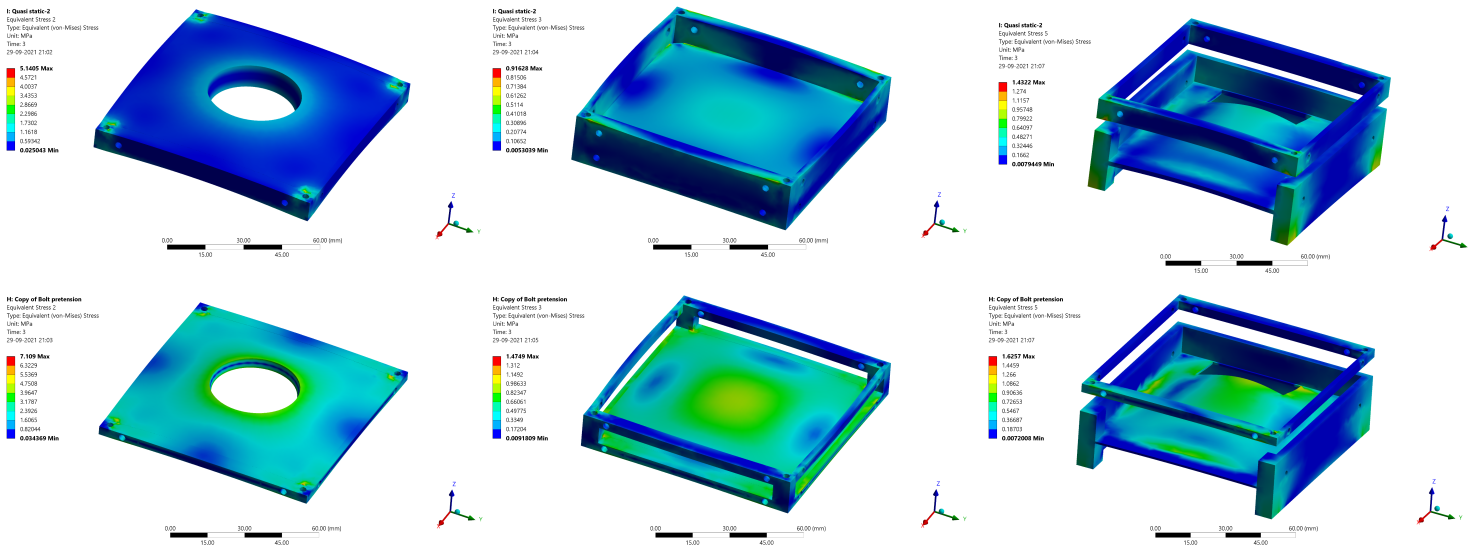
[1]. Laura Feria del Rosario et al. ,Design of the Mechanical Structure for the Teidesat Cubesat, July 2018 Final Bachelor thesis, Universidad de La Laguna.

[2]. Julie Fagerudd, Stress Simulation of the SEAM CubeSat structure during launch, 2015, Degree project in Solid Mechanics, Second Level, Stockholm, Sweden.

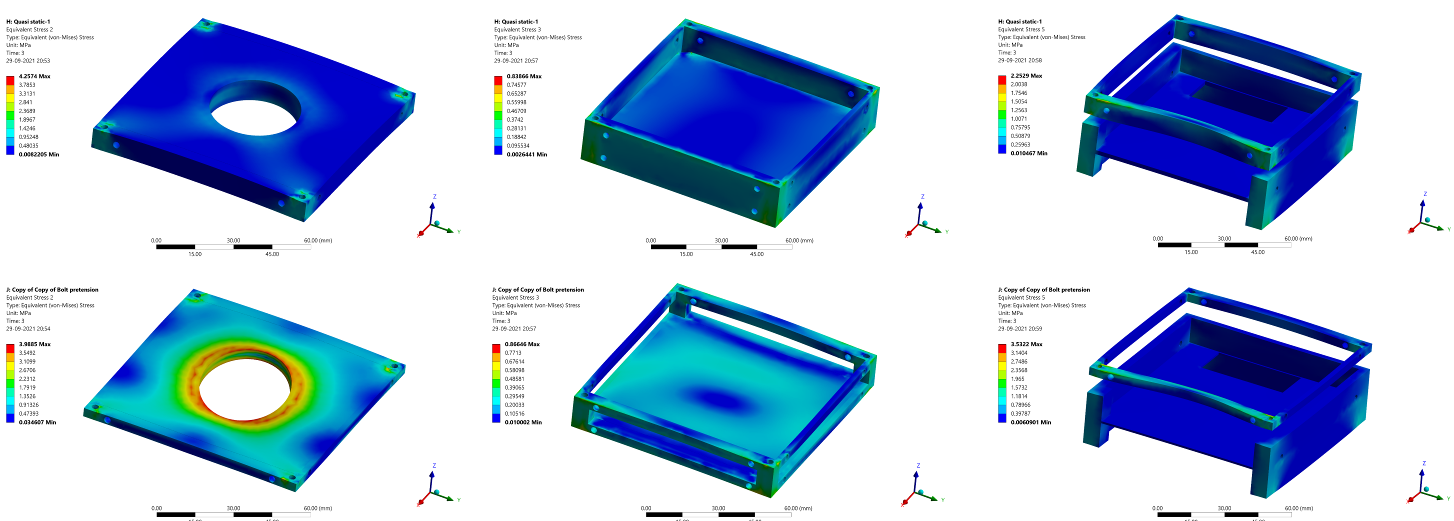
[3]. Delbert R Wilson, Vibration testing for Small Satellites, Boeing Aerospace Corporation.

Appendix A (Results)

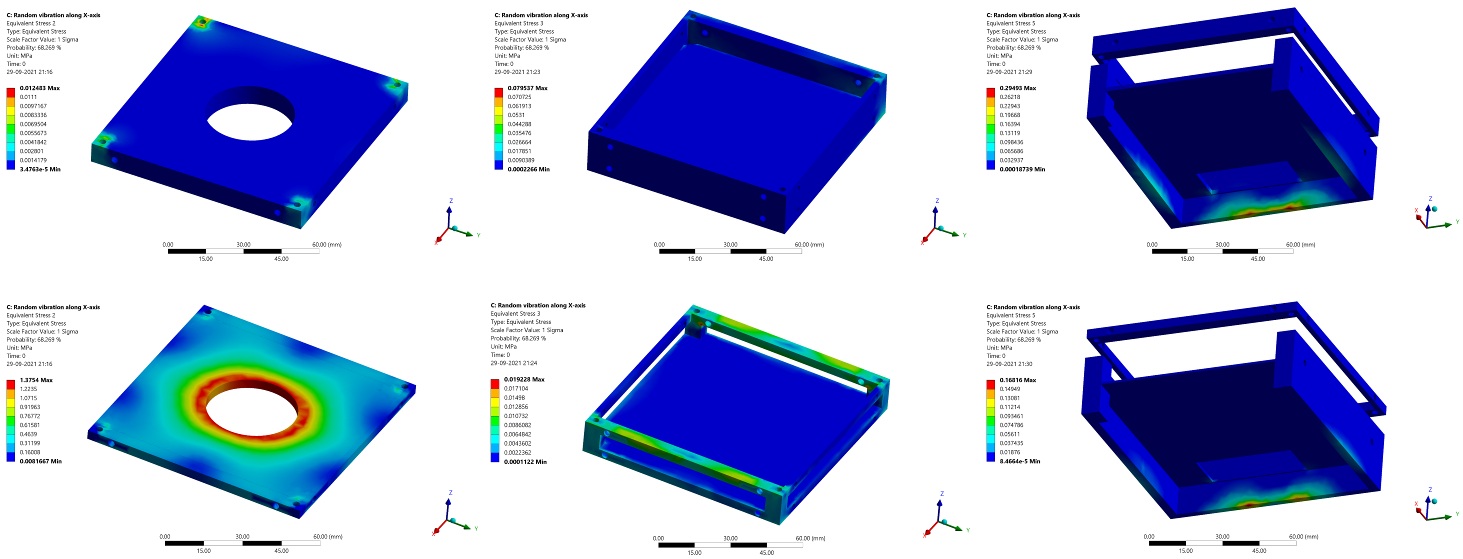
Structural case 1:



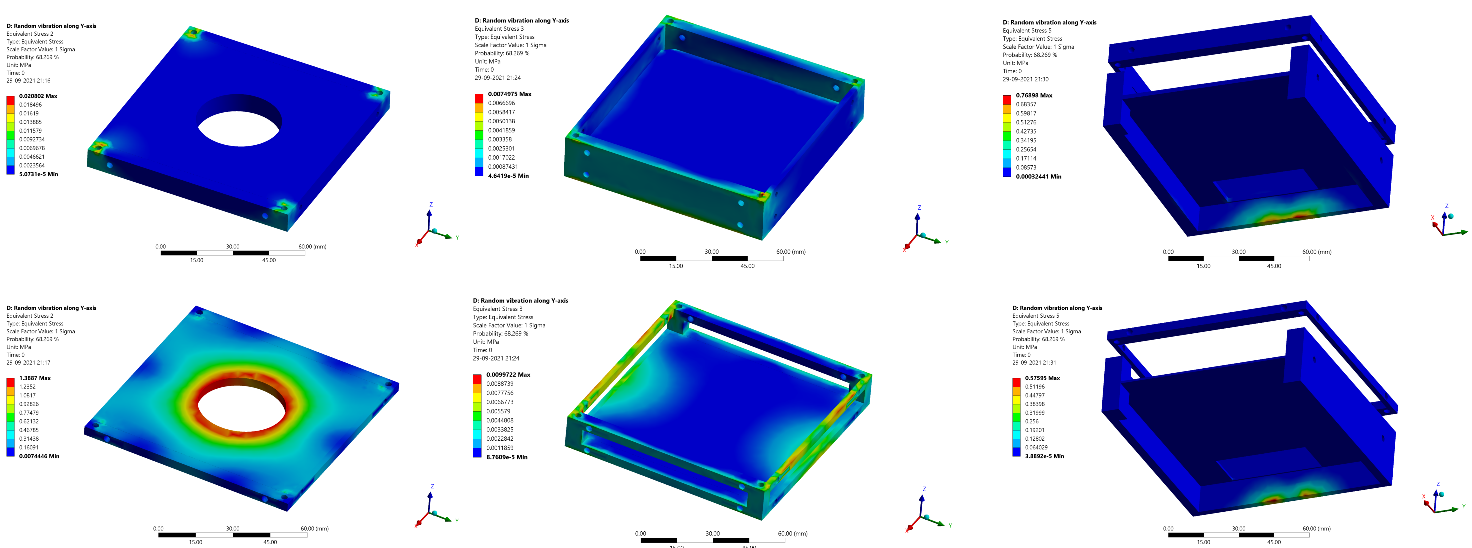
Structural case 2:



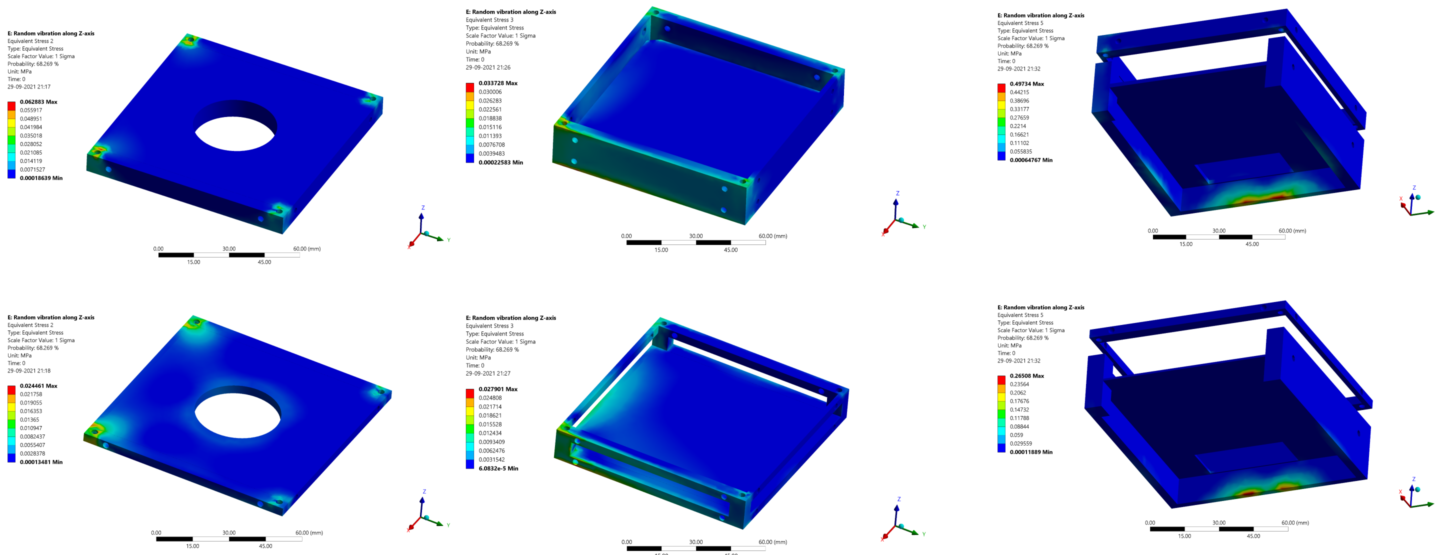
Random vibration case 1:



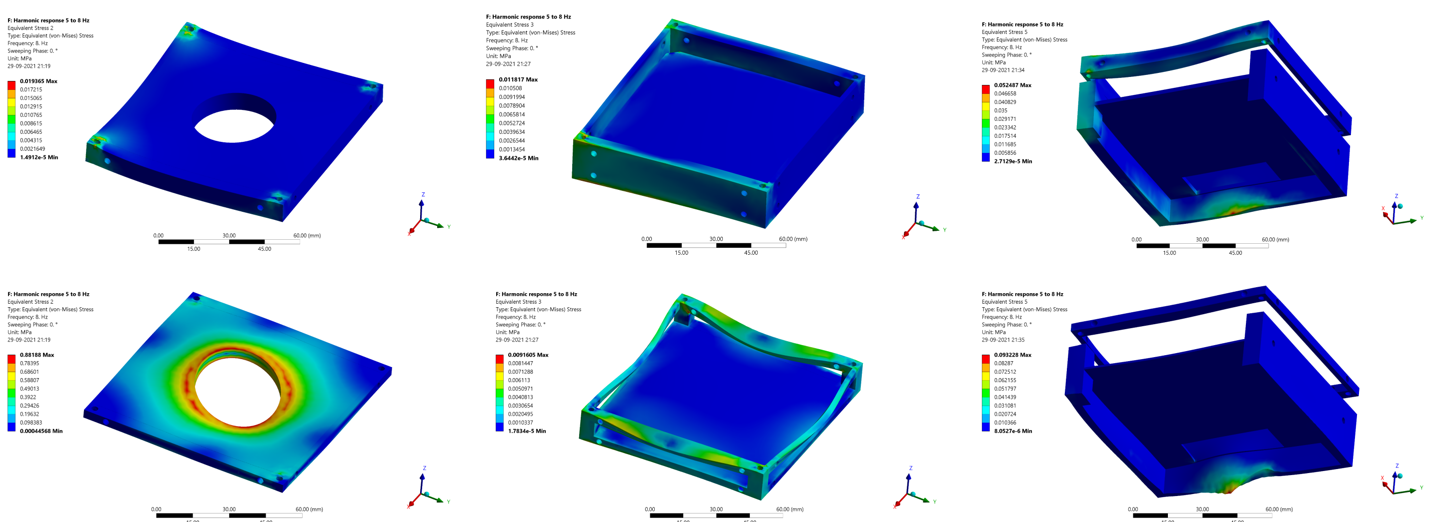
Random vibration case 2:



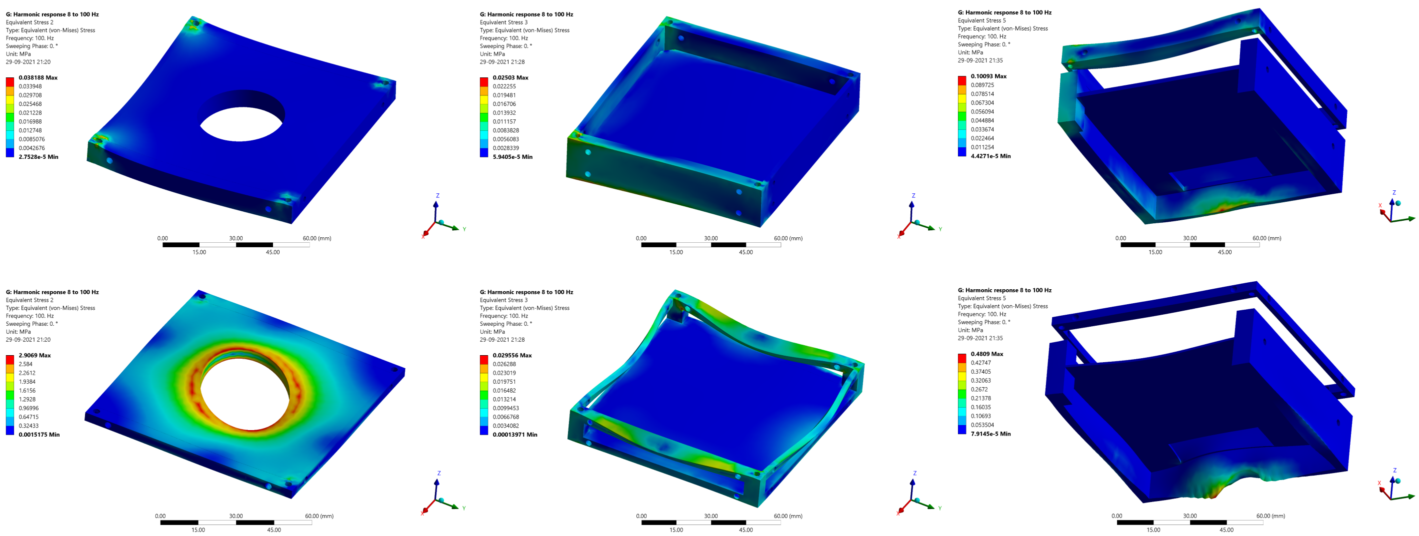
Random vibration case 3:



Sinusoidal vibration case 1:



Sinusoidal vibration case 2:



Appendix B (Flowchart)

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