Seismic Vibration Analysis on Industrial Control Panels through Finite Element Analysis

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Abstract: ***Industrial drives are electrical devices which are used to control the speed of motor by controlling frequency of the power input to the motor used in industries and in heavy duty applications like mining, weaving, cranes, hoists, marine, etc. In these application locations, vibration tends to be very heavy and also the dust particles and other environmental factors may affect the efficiency of the drive and even may lead to its failure. In addition to the above - mentioned vibrations these drives are also used in industries which are situated in seismic zones and the drive panels may fail during the time of earthquake. In this paper, considering the worst - case scenarios, the seismic analysis of drive enclosures (Drive panels) is performed as per seismic standards in ICC (International Code Council) considering zone IX earthquake zone. In this study, the critical parts are identified, stress and response acting on them are observed and the required design changes are considered to bring down the stress level to safe stress and make the panel robust in seismic zone.***

Keywords: Seismic vibration, Modal analysis, Seismic standards, Industrial drive panels, Industrial control panels, Random vibration, Earthquake vibration

1. Introduction

Seismic analysis is the calculation of response of a structure or a non - structure to earthquake vibrations. During earthquake electrical drives operating at high power tends to get damaged and cause failure of the cabinet structure and spatial interactions between the components inside the drive can lead to a lot of serious damage to the drive as well as the devices that are operated by the drive. At worst case catastrophic effect can cause loss of life.

For seismic vibration we have considered some parameters like floor response spectrum, ground acceleration, material properties of individual products, etc. The displacement and stresses at critical points are studied from the simulation result. The life of a control panel during earthquake depends on the intensity of earthquake, rigidity of the structure, stability of the panel, etc.

2. Product Introduction

2.1 Description of the Product

The structural model of the drive panel consists of filter with its electrical components like capacitors, inductors, etc. These are considered for Random Vibration analysis. The panel rests on the base plinth which is mounted to the floor using fasteners at each corner and the back plate is mounted to the panel using L – brackets. The passive filter components are mounted to the back plate. For effective air flow through the electrical components, they are covered by metal chassis. Above the chassis the contactor is placed for easy access. These ducts are also fastened to the back plate. A pair of lifting bars are placed above the enclosure for easy lifting of the complete product using hoists during transport and installation.

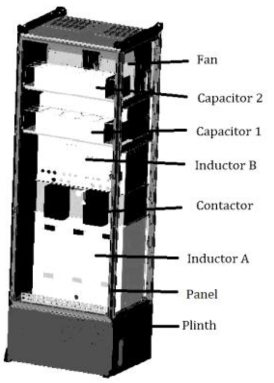

Figure 1: Industrial Drive Panel

2.2 Structural Model

The electrical components that are put in the panel for operation were not explicitly modeled. The mass of the electrical components is considered as distributed mass or pointed mass at their center of gravity.

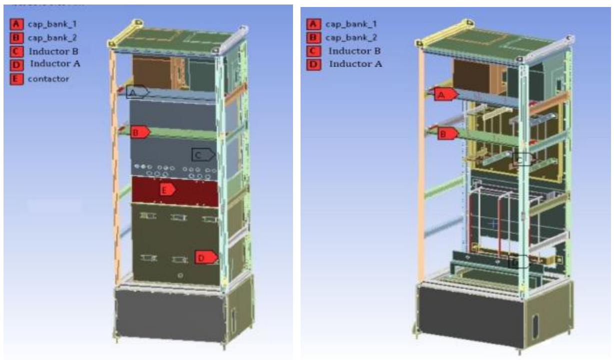

Figure 2: Structural Model

The external chassis/support structures of the electrical components are modelled, and the distributed mass is given on the surfaces of the chassis. The final mass of the model after applying the material to the individual components should be nearly equal to the actual mass of the product.

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3. Interpretation of PSD curve from response Spectrum Curve

According to AC 156 standard, the acceleration data is taken based on the seismic zone and the floor acceleration. The response spectrum curve is given by the standard and it is formulated. From this curve, the random vibration curve is interpreted using

ASD = $(g^2/2)$ *bandwidth (1)

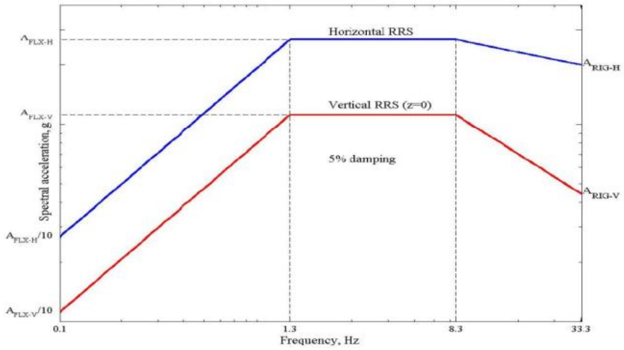

Figure 3: AC156 Standard Curve For Seismic Evaluation

The Power Spectral Density (PSD) curve is derived from the response spectrum curve. This random vibration curve is used as input for the seismic analysis.

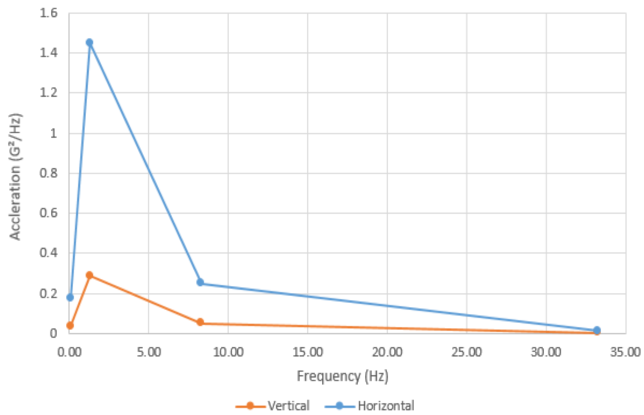

Figure 4: Interpreted PSD Curve for Simulation

4. Random Vibration Analysis

Random vibration analysis is performed since earthquake vibration frequencies are unpredictable and they occur in a random manner. Most commonly Response Spectrum Analysis (RSA) is performed but it's not acceptable because in RSA, the frequencies are chosen in a linear manner from 0.1 to the last frequency in range.

Initially free – free vibration analysis is performed to check the contacts before simulation. Frictional contacts are used for fasteners with friction coefficient 0.16. After free – free vibration, modal analysis is performed to find the natural frequencies of the product. The number of modes must be at least 1.5 times of the frequency range used for simulation and also the Mass Participation Factor must be greater than 85%

The critical components are chosen based on the support structures and the mass acting on them. The components are considered critical because when they fail, the damage to the product and the environment can cause disastrous effect.

The critical components considered for analysis are;

1) Backplate hinge
2) Capacitor support
3) Inductor A
4) Plinth
5) Inductor B plate
6) Panel

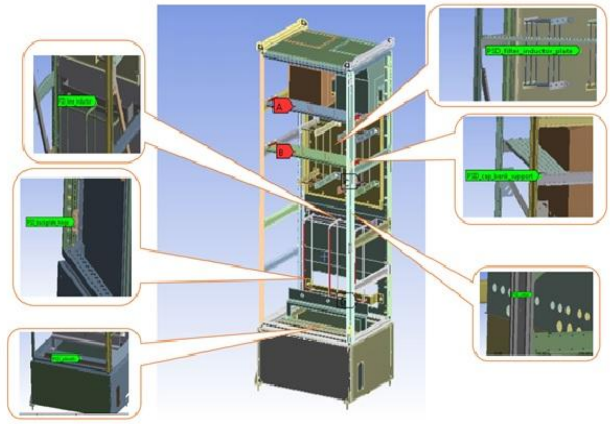

Figure 5: Critical Components Chosen

4.1 Initial design

The initial design of the product is chosen as model for vibration analysis. Table I shows the acceleration responses observed at the critical components.

Table 1: Resonant frequencies observed for existing design with modal frequency of 8.39Hz, 16.67Hz, etc.,

Component	Directional peak acceleration					
	X		Y		Z	
	Freq. [a] (Hz)	Accln. [b] (G)	Freq. (Hz)	Accln. (G)	Freq. (Hz)	Accln. (G)
Backplate hinge	-	-	8.3937	0.1627	8.3724	2.5994
	16.558	1.44	16.601	0.28	16.605	0.3926
Capacitor support	-	-	8.3879	0.1313	8.3937	679.56
	16.596	173.15	-	-	16.61	15.157
Inductor A	-	-	8.4068	0.1036	8.3879	364.09
	16.596	25.52	-	-	-	-
Plinth	-	-	-	-	8.3319	0.422
	16.528	0.25	-	-	-	-
Inductor B plate	-	-	8.3951	0.2196	8.4	701.29
	16.589	23.04	-	-	-	-
Panel	-	-	8.388	0.088	8.3879	498.82
	16.596	108.2	-	-	16.657	12.3

Freq. [a] – Frequency, Accln. [b] - Accleration

From Table 1 it is observed that resonance occurs at almost all the critical components. In random vibration analysis, only the equivalent stress can be found out because the frequency is chosen in random manner and also there is no chance of finding at what frequency what stress acts on the product. Directional stress is also not possible since it is a triaxial simulation. The maximum stress acting on every element during the run time of the simulation is considered and plotted as a single stress result at the end of simulation.

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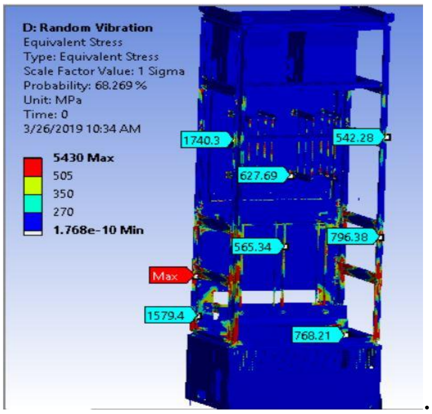

Figure 6: Stress Result of Initial Design

From Figure 6 it is observed that, the stress values are very high such that the product cannot pass the seismic testing. Hence the support structures are added based on Design 2 available and the same procedure is followed.

4.2 Design - 2

There are already existing solutions for the panel structure against seismic vibrations by adding supports to the Panel frame. They cannot be used directly because of certain electrical constrains and also they are very costly. But for study purpose they are considered. Apart from that, the bolting location in base plinth tends to fail in most of the cases in vibration testing. Hence additional modifications are added to plinth to make it stable during vibration.

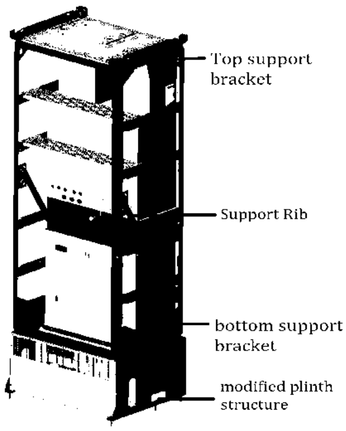

Figure 7: Design 2 with Modified Plinth

To the existing plinth, additional spacers are used at bolting location so that the stress transferred from bolt to the plinth is avoided since the spacer component absorbs the stress thus saving the plinth from failure. The number of bolts are increased from 2 to 3 bolts so that more contact area is achieved at the base and hence the stress is distributed to 3 bolts. Additional cutout is given on the plinth surface for easy access of the 3rd bolt at the center of the plinth.

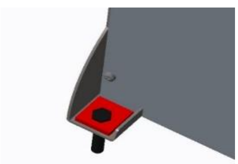

Figure 8: Spacer Used at Bolting Location

Table 2: Resonant frequencies observed for design 2 with modal frequency of 14.657, 17.1058, etc

Component	Directional peak acceleration					
	X		Y		Z	
	Freq. (Hz)	Accln (G)	Freq. (Hz)	Accln. (G)	Freq. (Hz)	Accln. (G)
Backplate hinge		-	14.647	0.232	14.614	0.7085
	17.055	1.1066	17.113	0.1797	17.137	0.112
Capacitor support	-	-	14.651	0.679	14.647	106.18
	17.113	107.65	-	-	17.075	7.2189
Inductor A	-	-	14.663	0.068	14.651	91.157
	17.094	19.546	-	-	-	-
Plinth	-	-	-	-	14.587	0.2717
	17.024	0.214	-	-	-	-
Inductor B plate	-	-	14.651	0.3572	14.659	173.28
	17.087	17.088	-	-	-	-
Panel	-	-	14.651	1.34	14.641	64.553
	17.094	91.871	-	-	17.075	4.42

From Table 2 it can be seen that the acceleration values have been reduced and the stress values have been reduced to a greater extend compared to the initial design but still there is more chance for failure since the acceleration levels are far higher than safe levels and the stress results are higher than the ultimate strength of the material which can be a cause for failure. Also, it is difficult for electrical connections with this existing support. Hence these support structures are redesigned and the simulation is proceeded.

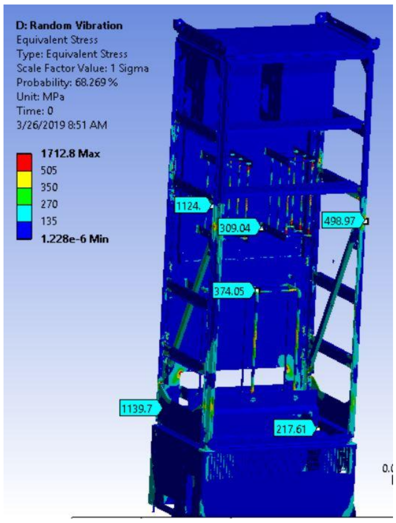

Figure 9: Stress Result of Design – 2

4.3 Redesigned model

The Design 2 is now redesigned to meet the electrical constrains and then the response is studied. The results are observed and based on the response the amount of damping required can be observed. Additional support brackets are added to improve strength of the critical components. Also the plinth height is changed from 400mm height to 200mm.

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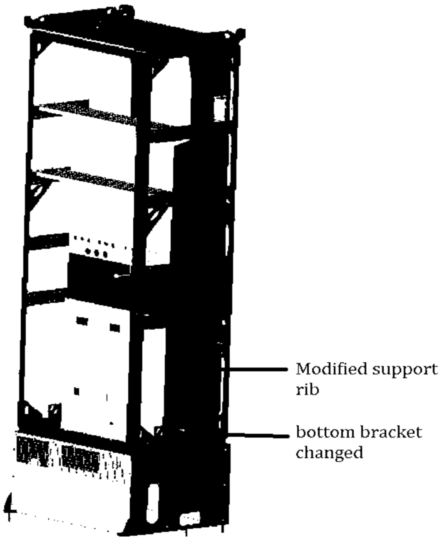

Figure 10: Redesigned Model

Table 3: Resonant frequencies observed for Design 2 with modal frequency of 14.657, 17.1058, etc.

Component	Directional peak acceleration					
	X		Y		Z	
	Freq. (Hz)	Accln (G)	Freq. (Hz)	Accln. (G)	Freq. (Hz)	Accln. (G)
Backplate hinge	-	-	11.373	0.244	11.347	0.641
	13.869	1.078	13.906	0.404	13.885	0.402
Capacitor support	-	-	11.376	0.394	11.373	262.28
	13.9	439.27	13.894	0.038	13.894	70.873
Inductor A	-	-	11.373	0.263	13.373	67.915
	13.894	12.598	13.904	0.561	13.9	17.855
Plinth			-	-	11.293	0.332
	13.843	0.22	-	-	-	-
Inductor B plate			11.376	0.433	11.376	363.17
	13.894	23.587	-	-	-	-
Panel	-	-	11.373	0.1225	11.326	0.656
	13.869	1.0161	13.914	0.3969	13.9	0.463

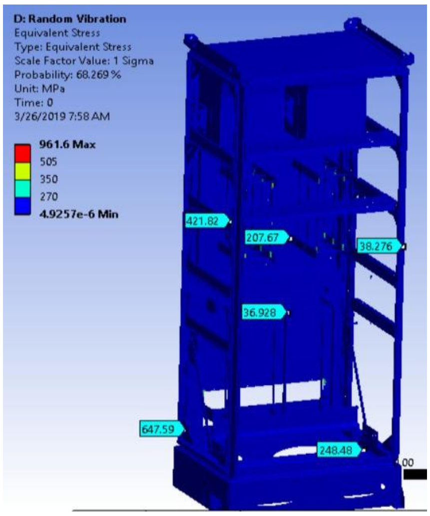

Figure 11: Stress Result of Redesigned Model

From Table 3 it can be seen that the acceleration values have been increased in certain components and the stress values have been reduced to a greater extend compared to the initial design but still there is more chance for failure since the acceleration levels are far higher than safe levels even though the stress results are at acceptable range. Hence to reduce the low frequency acceleration, we can directly use dampers to the redesigned model and control these peak accelerations at the given range.

4.4 Damped model

The dampers are used from Vibrotech SMR dampers. Based on the mass of the product and the number of dampers required, the damper is chosen. For our application, to use this damper, 200mm high plinth is required since the dampers height is only 180mm and can be varied upto 30 mm height only. Hence for seismic application only 200mm plinth is recommended.

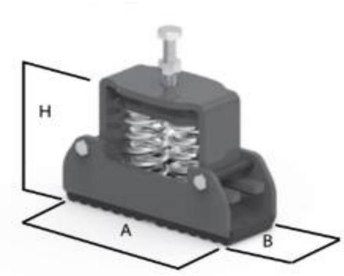

Figure 12: Vibrotech SMR Damper

Totally 4 dampers are added to the base plinth, 2 at front and 2 at rear side with a support plate in plinth because as per the damper specifications they can have a maximum load of 250kg, and our product is 958 kg. The response at the critical locations is studied after simulation with damper.

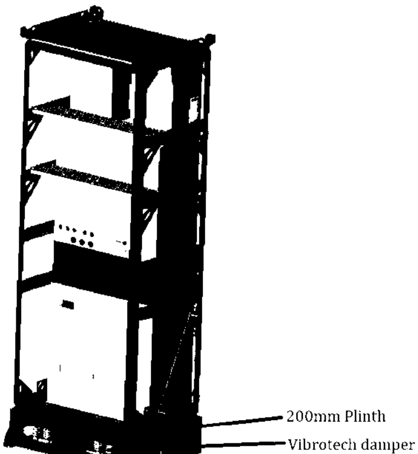

Figure 13: Redesigned Model with Dampers Added

Table 4 shows that almost all the resonant frequencies are damped in the given frequency range. The acceleration values are far less compared to the previous models and the stress results are also found to be very minimal compared to all other previous models.

Table 4: Resonant Frequencies observed for Damped model with modal frequency of15.66, 18.886, etc.

Component	Directional peak acceleration					
	X		Y		Z	
	Freq. (Hz)	Accln (G)	Freq. (Hz)	Accln. (G)	Freq. (Hz)	Accln. (G)
Backplate hinge	18.371	0.125	18.838	0.019	-	-
Capacitor support	-	-	-	-	15.303	1.4679
	18.838	7.1926	-	-	18.886	0.719
Inductor A	-	-	-	-	14.922	0.3586

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	18.636	0.65114	18.819	0.031	18.953	0.1205
Plinth	-	-	-	-	-	-
Inductor B plate	18.756	1.119	-	-	-	-
Panel	18.851	5.534	-	-	-	-

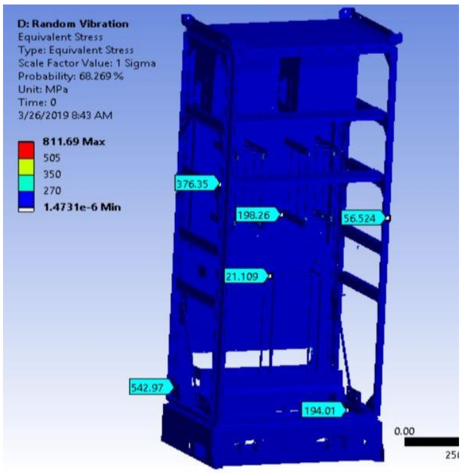

Figure 14: Stress Result of Damped Model

Figure 14 shows that the resonant frequencies are damped to a greater extend and the stress levels are very low which means the product is safe with the selected dampers.

5. Comparison of Results

The responses obtained from the simulated data after Random vibration analysis are compared for all 3 axes on all the critical components chosen. From the comparison graphs we can see that the damped model does not have any peak response within the given frequency range, which means there is no resonance in the product. The comparison graphs are shown.

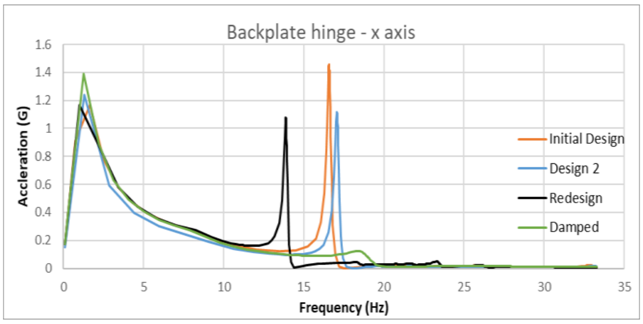

Figure 15: Response Comparison – Backplate Hinge – X – Axis

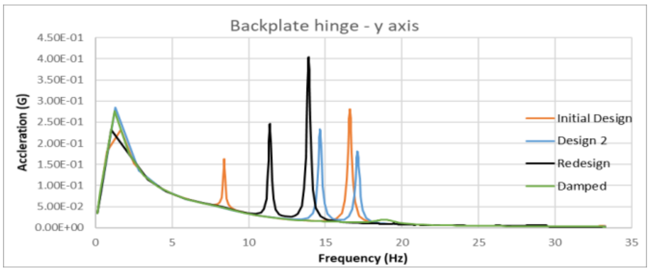

Figure 16: Response Comparison – Backplate Hinge – Y - Axis

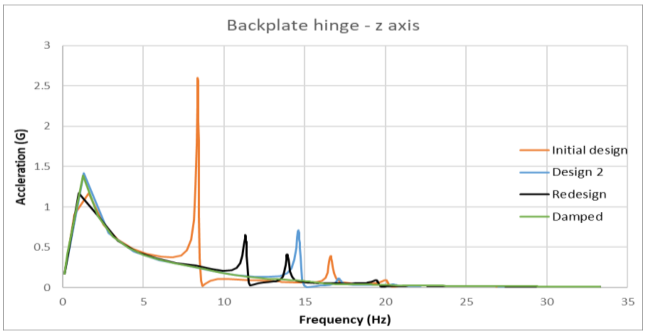

Figure 17: Response Comparison – Backplate Hinge – Z – Axis

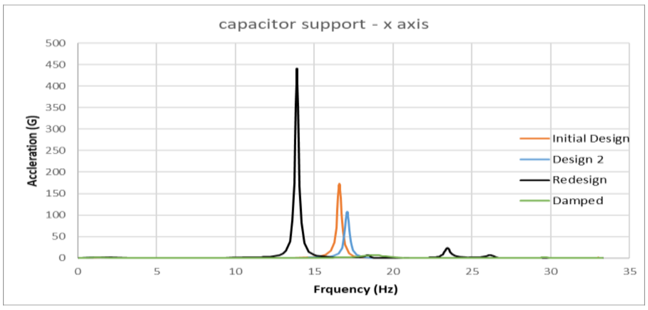

Figure 18: Response Comparison – Capacitor Support – X – Axis

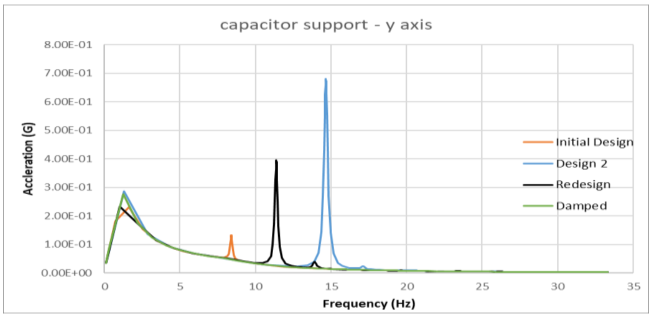

Figure 19: Response Comparison – Capacitor Support – Y – Axis

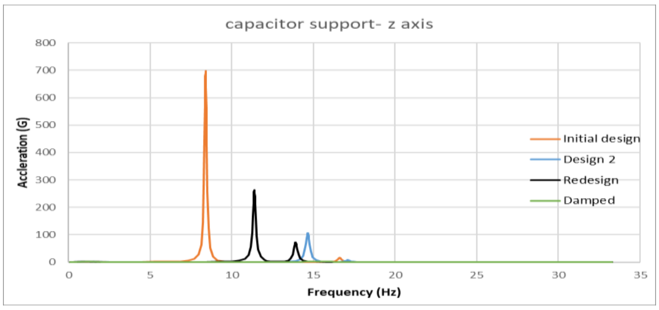

Figure 20: Response Comparison – Capacitor Support – Z – Axis

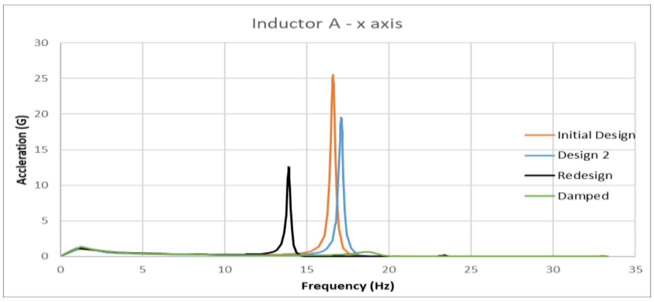

Figure 21: Response Comparison – Inductor A – X - Axis

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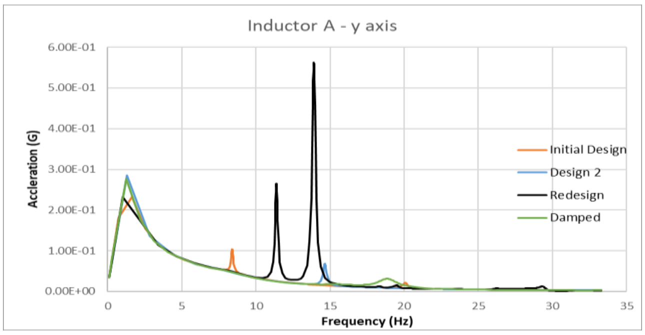

Figure 22: Response Comparison – Inductor A – Y - Axis

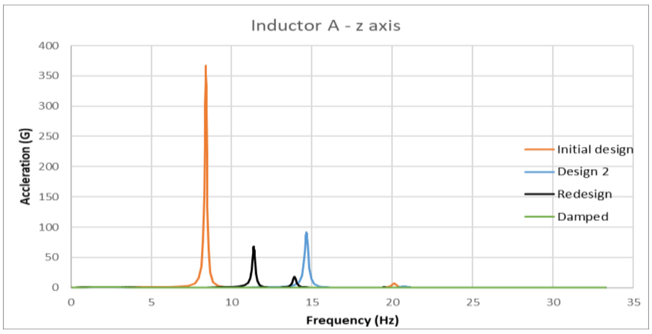

Figure 23: Response Comparison – Inductor A – Z - Axis

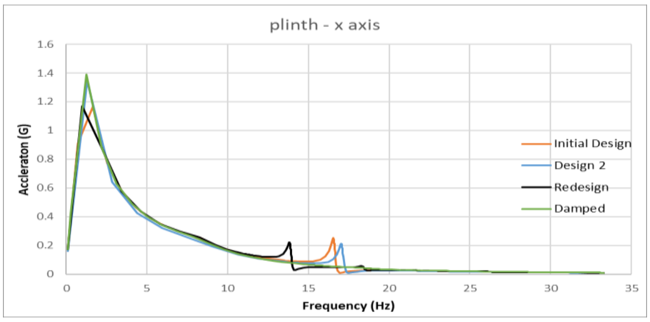

Figure 24: Response Comparison – Plinth –X Axis

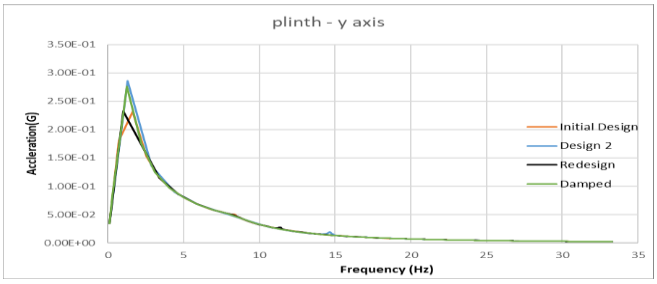

Figure 25: Response Comparison – Plinth –Y Axis

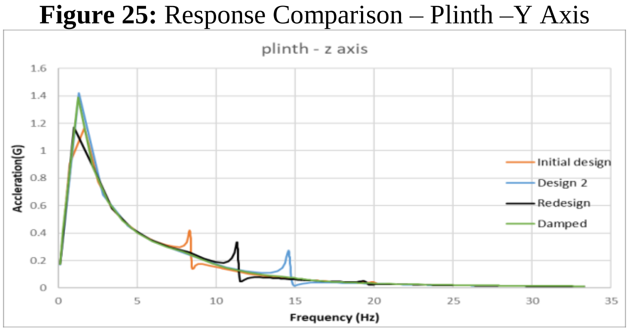

Figure 26: Response Comparison – Plinth –Z Axis

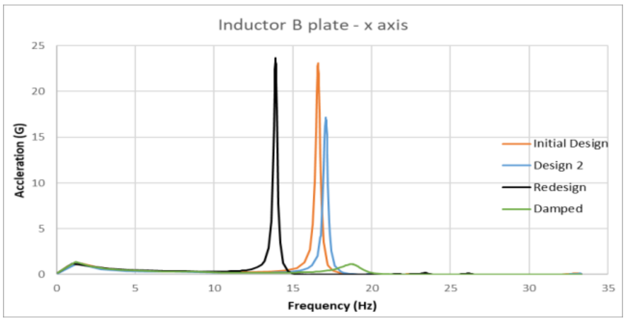

Figure 27: Response Comparison – Inductor B Plate – X - Axis

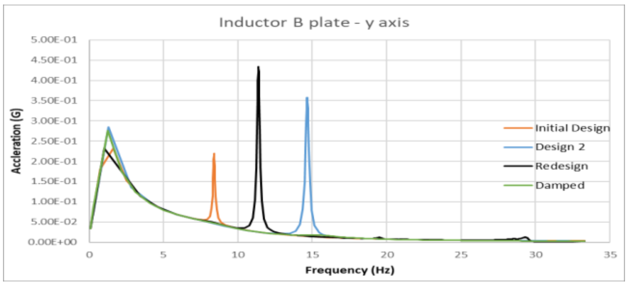

Figure 28: Response Comparison – Inductor B Plate – Y - Axis

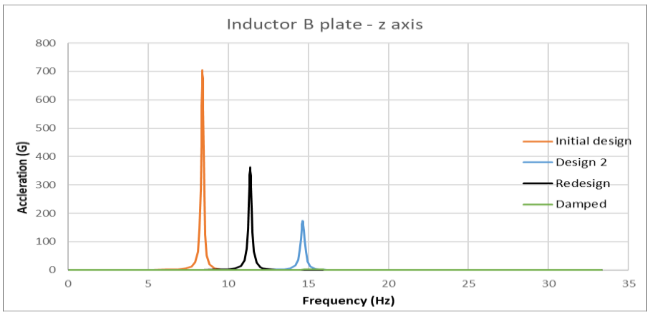

Figure 29: Response Comparison – Inductor B Plate – Z - Axis

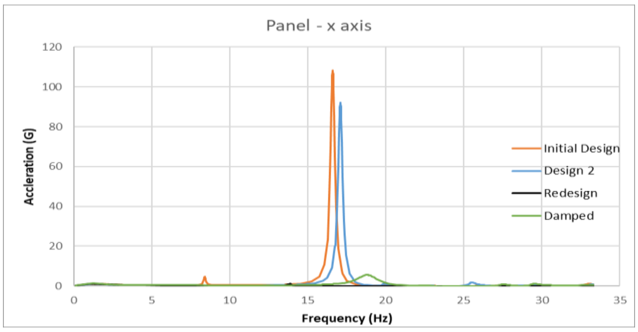

Figure 30: Response Comparison – Panel –X - Axis

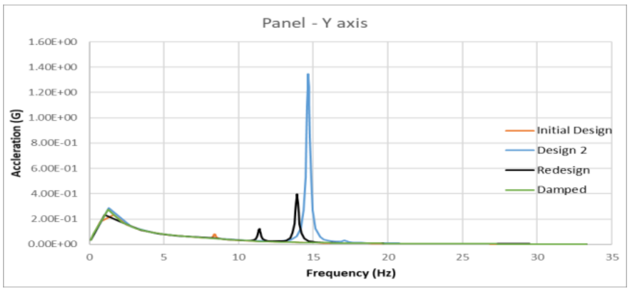

Figure 31: Response Comparison – Panel –Y - Axis

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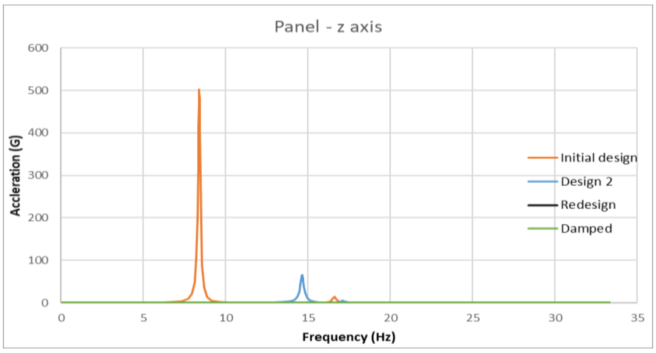

Figure 32: Response Comparison – Panel –Z - Axis

Table 5: Stress comparison in critical components

Component	Equivalent stress (MPa)			
	Initial Design	Design 2	Redesigned model	Damped model
Backplate hinge	1579.4	1139.7	647.59	542.97
Capacitor support	1740.3	1124	421.82	376.35
Inductor A	565.34	374.05	36.928	21.109
Plinth	768.21	217.61	248.48	194.01
Inductor B plate	627.69	309.04	207.67	198.26
Panel	542.28	498.97	38.276	56.24

From Table 5 we can see that the stress values are below the safety limit and hence the product can withstand seismic vibrations. Also the acceleration values are damped to a greater extend which means there is very minimal chance of resonance in the given frequency range.

6. Conclusion

The response of the structural model of drive panel is studied and improved to reduce the effect of vibration in seismic applications. Though shifting the resonance frequency will make the product reliable according to the standards, vibration is still present. Also the Base - plinth is more predominant to fail at bolting locations and hence the proposed design for the base plinth is mandatory for its reliability. The vibration path is likely to follow the enclosure and via the horizontal side support and affect the structure. Thus the effective way of reducing the vibration would be by restricting the level of vibration transferred all through the structure using the dampers proposed. The base - plinth experiences the primary excitation and the level of vibration transferred can be brought down by using vibration isolation sheets or rubber bushes which would drop the vibration level. The future scope is to incorporate these possibilities and observe the fall in vibration level which would give an idea of the functionality of the vibration isolators and also aid in bringing up a robust design to the market.

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