Random Vibration Analysis for an All-Plastic Crash Energy Management Solution using Abaqus®

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Abstract: Automobiles account for about one-quarter of carbon dioxide emissions (IEA, 2009), a major contributor to the greenhouse effect. Concerns over the prospect of global climate change are leading to growing pressure on automobile manufacturers from consumers and governments the world over, to improve fuel economy as a means to conserve oil and control pollution. This has increased the importance of lightweight materials and designs and automotive manufacturers are looking at incorporating alternative lightweight materials in relevant systems. Automotive body and chassis system is one of the probable areas wherein the designs using lightweight materials can play a major role, since this generally accounts for nearly one-third of the weight of the car.

Development of the alternative lightweight designs requires extensive validation from performance perspectives before being introduced in a production car. Computer aided engineering (CAE) plays a very important role in the performance validation by enabling checking of numerous potential design concepts at a virtual level before being evaluated with physical prototype testing. One of the most important evaluations involves checking the ability of the proposed design to withstand vibrations while the car is driven on a rough road. This paper focuses on random vibration simulation of a lightweight automotive chassis member using Abaqus^{TM}. Potential high stress regions are highlighted through the simulation and necessary design improvements are suggested.

Keywords: Random Vibration, Abaqus, Crash, Energy Management, Thermoplastic

1. Introduction

Automobiles are a major contributor to the greenhouse gas emissions. Environmental concerns are leading to growing pressure on automobile manufacturers from consumers and governments the world over, to improve fuel economy as a means to reduce greenhouse gas emissions and control pollution. This has increased the importance of lightweight materials and designs and automotive manufacturers are looking at incorporating alternative lightweight materials in relevant systems. Automotive body and chassis system is one of the probable areas wherein the designs using lightweight materials can play a major role, since this generally accounts for nearly one-third of the weight of the car.

Development of the alternative lightweight designs requires extensive validation from performance perspectives before being introduced in a production car. Computer aided engineering (CAE) plays a very important role in the performance validation by enabling checking of numerous potential design concepts at a virtual level before being evaluated with physical prototype testing.

The next generation lightweight thermoplastic crash energy management solutions are rigorously tested on numerical platforms for their primary functions which is to adequately absorb energy as required from the regulatory and Original Equipment Manufacturer (OEM) requirements. Explicit dynamic simulations using commercial codes such as LS-DYNA, RADIOSS AND PAMCRASH are performed to check the crash performance. However, in addition to adequate performance under crash conditions, the thermoplastic solutions also need to withstand loads during normal operating conditions such as vibration loads coming from various road profiles. The vibration from the road undulations is random in nature where vibration excitations consist of many frequencies at the same time and the amplitude at these frequencies varies randomly with time. Hence deterministic dynamic analysis procedures can't be used since all necessary parameters of the analysis can't be uniquely determined or known. In this situation, statistical or probabilistic methods are used to characterize the vibrations and response under these vibration conditions.

Random vibrations, in general, have been observed for a long time because of the effects on structures of natural phenomena such as earthquakes and ocean waves. This has been studied in a mathematical framework since about the turn of 20th century starting with Albert Einstein (Paez, 2006). However, the modern field of random vibrations of mechanical systems and probabilistic structural dynamics, in general, has gained importance as the awareness that real mechanical environments are stochastic has broadened and occurrence of failures of most real physical systems due to random vibrations has been observed (Paez, 2006). Hence, different industries are adopting random vibration analysis into their standard design cycles (Fuente, 1999).

The acceleration waveform shown in Figure 1 (Van Baren, 2012) for dashboard vibration of a vehicle traveling on Chicago Drive near Hudsonville, MI is a typical case of random vibration. As can be seen, the vibration signature contains many frequencies at the same time and the amplitudes of these different frequencies vary with time.

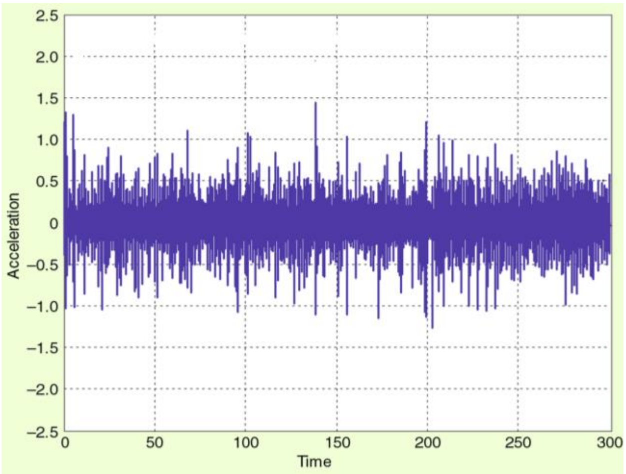

Figure 1 Typical random vibration example

Product manufacturers put instrumentation at desired locations on the product and measure the vibration data coming due to the random loading. This data is characterized using probabilistic methods to be used for part designs. In some industries, this data is specified as a standard random vibration profile to be used for industrywide design activities such as ISTA standard profile for packaging industry (Wallin, 2007) and SAE J2380 PSD profile for battery industries (Binshan, 2012). One such standard vibration spectrum for design of materiel installed within a jet aircraft store as per MIL standard 810 is shown in Figure 2.

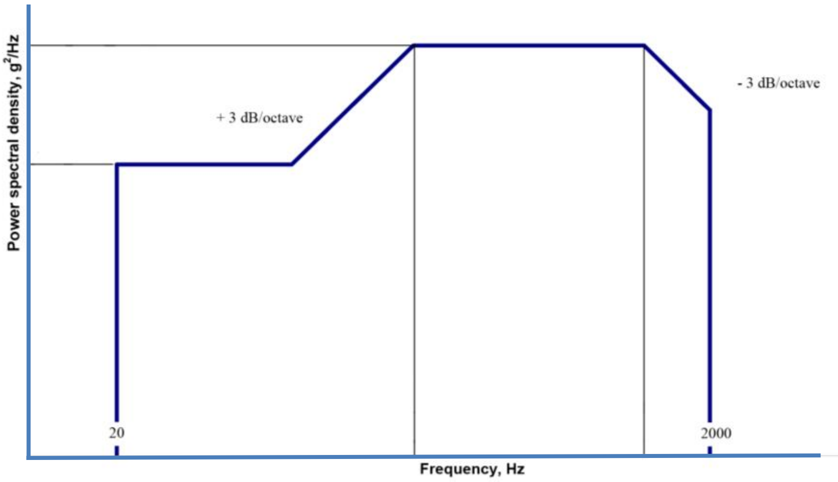

Figure 2 Typical example of a standard random vibration profile in aviation industry

However, in the case of automotive industry, each OEM has its own standard random vibration profile (Scwab, 1995). This paper explains the design requirements of an automotive crash energy management module and specifically talks about numerical evaluation for random vibrations coming due to road undulations in detail.

2. Automotive rail extension system

Rail extension is a part in the vehicle chassis which comes behind the bumper beam and acts an interface between the bumper beam and vehicle rail. This is primarily designed to absorb the majority of crash energy in the event of a medium speed (15kmph) offset crash and limit vehicle damageability. However, there are multiple other design requirements as illustrated in Figure 3. As can be seen from the figure, the rail extension system not only has to meet multiple performance requirements, it also ensures this performance at various temperatures starting from a low ambience temperature to a high temperature such as 70°C since it's nearer to the engine. Additionally, it should be e-coat capable to allow online painting of body-in-white (BIW) if needed. Out of the various performance requirements, one of the most important requirements apart from crash functionality is the ability to carry bumper beam assembly weight under various operating conditions such as under different road undulations.

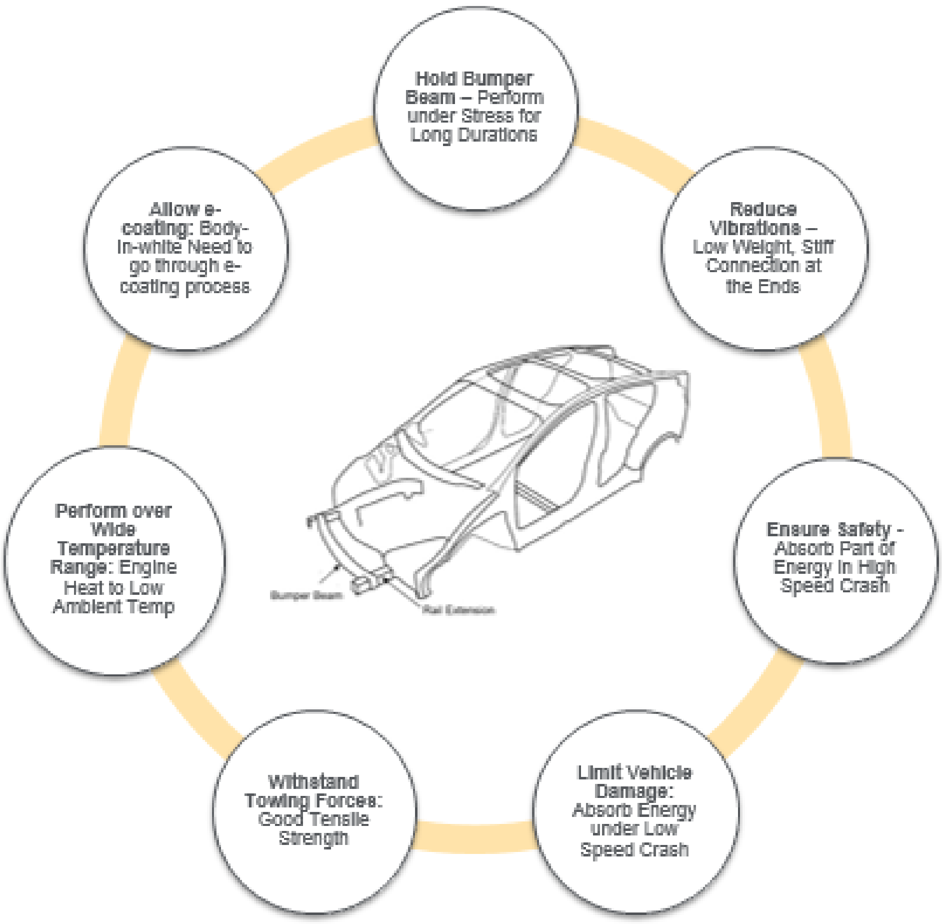

Figure 3 Design requirements for an automotive rail extension

As discussed in the previous section, vibration excitations due to road undulations are random in nature and various OEMs have their own vibration profile. Sample OEM random vibration profiles from literature are shown in Table 1.

Spetrum -1						Spectrum-2						Spectrum-3					
X-Spectrum		Y-Spectrum		Z-Spectrum		X-Specturm		Y-Spectrum		Z-Spectrum		X-Specturm		Y-Spectrum		Z-Spectrum	
Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz
5	.0009	5	.004	5	.004	5	.004	5	.04	5	.00042	5	.017	5	.0025	5	.04
24	.003	11	.23	21	.0031	10	.02	14	.20	14	.09	10	.08	12	.013	12	1.0
30	.11	500	.04	29	.07	40	.013	58	.20	60	.002	40	.08	34	.013	25	.10
36	.003			35	.0028	180	.00006	95	.013	500	.0002	55	.006	71	.0018	123	.01
138	.00025			51	.0022	500	.00006	500	.013			153	.001	85	.0001	500	.01
142	.001			57	.018							500	.001	500	.001		
149	.0002			66	.0019												
210	.00004			82	.0015												
500	.00004			86	.0103												
				92	.0013												
				110	.00101												
				115	.007												
				120	.00093												
				235	.00022												
				238	.00051												
				241	.00020												
				278	.00012												
				500	.00012												

Table 1 Sample OEM random vibration profiles from literature

Currently, rail extension modules are made of metals, mostly high-strength steels. A typical steel design is shown in Figure 4. The design consists of a front plate, the main rail extension and a rear plate. The main rail extension in turn consists of two separate steel plates which are welded at the interface. The front plate is used to attach the rail extension to the bumper beam whereas the rear plate is used to attach rail extension to the rail. SABIC has advanced lightweight thermoplastic solutions for this application the details which is explained in more detail in next section.

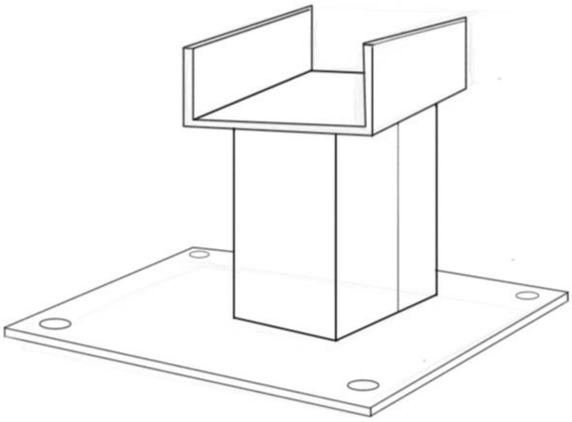

Figure 4 A Typical metal rail extension design

3. Lightweight thermoplastic rail extension solution from SABIC

A typical lightweight thermoplastic rail extension solution from SABIC Innovative plastics is shown in Figure 5. The usual material of construction of these parts is NORYL GTX^{TM} 910 which is blend of polyphenylene ether (PPE), polystyrene (PS) and polyamide (PA). These solutions provide significant advantages over metal designs such as

1. ~ 30 -50% lesser weight
2. Reduced part count (1 in thermoplastic solution compared to 4 for steel design)
3. Higher energy absorption efficiency.
4. Superior material damping compared to metals resulting in an advantage under vibration loading.
5. Similar e-coat capability

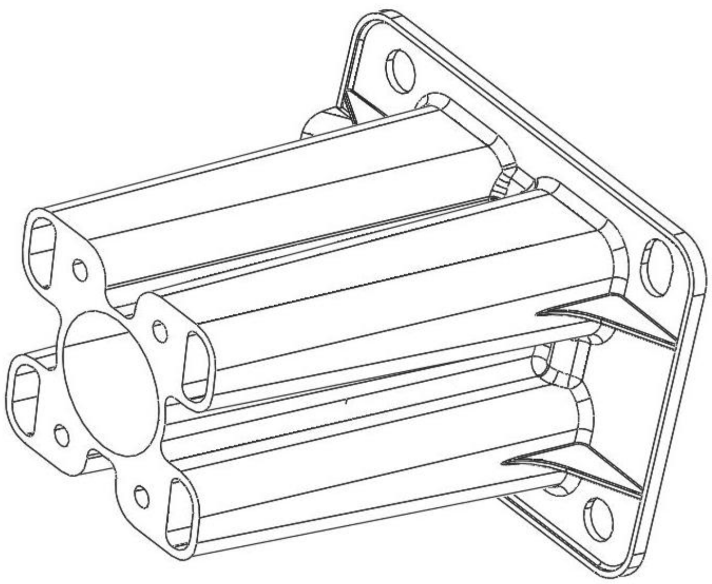

Figure 5 Lightweight thermoplastic rail extension solution from SABIC

4. Random vibration load used for numerical verification of thermoplastic rail extension solution

The last spectrum from the OEM profiles shown in Table 1 is used to verify the performance of the proposed thermoplastic rail extension in the current work. These loading spectrums are shown graphically in Figure 6.

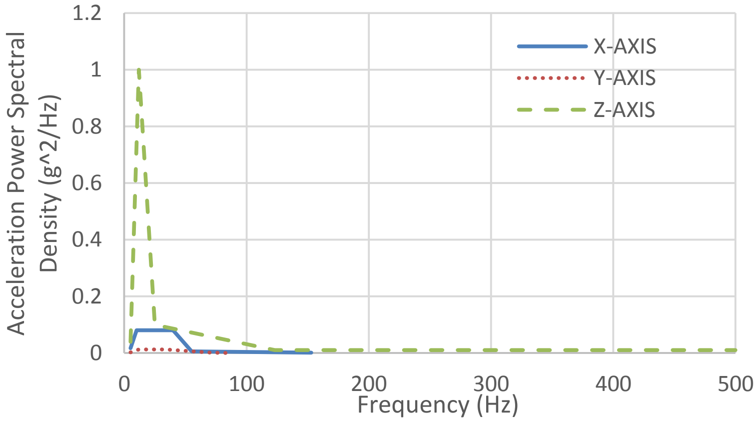

Figure 6 Random vibration loading spectrum used in numerical verification

As can be seen, the random vibration input is specified in terms of an acceleration power spectral density (PSD) which is a form of random base motion. PSD is a statistical measure defined as the limiting mean-square value of a random variable. It is used in random vibration analyses because the instantaneous magnitudes of the response can be specified only by probability distribution functions that show the probability of the magnitude taking a particular value (Harvey, 2000). PSD is measured as the mean-square magnitude per unit bandwidth of the output of an ideal filter with unity gain responding to the vibration as follows:

$$W(f) = F(f)^2 RMS/\Delta\, f$$

Where by convention the bandwidth f is usually chosen to be 1Hz.

5. Finite element model

The system considered for the numerical verification is shown in Figure 7. As can be seen, the system consists of the rail extension and bumper beam only. Bumper energy absorber isn't considered in this exercise due to its much lower weight compared to the bumper beam.

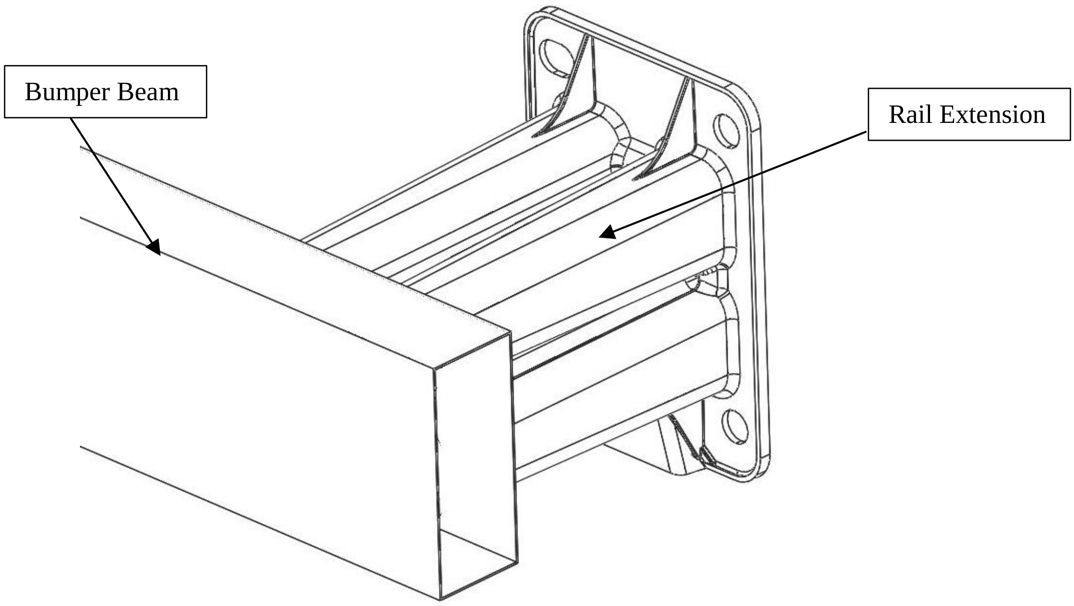

Figure 7 System considered for numerical verification

Corresponding finite element model is shown in Figure 8. The number of nodes and elements for this model is 112,294 and 113428 respectively. Average thickness of rail extension in this model is 4.75mm. Rail extension is joined to the bumper beam through insert molded bolts. This is modeled as MPC elements in Abaqus® commercial code. Model is constrained in all directions at the four bolt locations each on LH and RH sides. These boundary conditions are defined through *BOUNDARY card which is specified outside of the analysis steps.

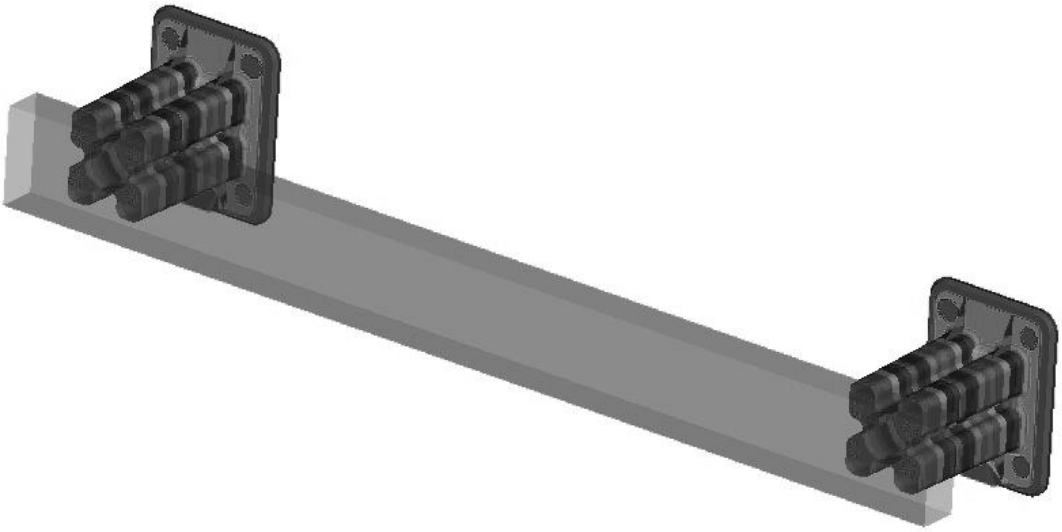

Figure 8 Finite element model

Only elastic material data is required for the random response simulation which are given below.

Bumper Beam and Rail Mount Plate: Material - Steel

Density: 7890 kg/m^3

Young's Modulus: 205 GPa

Poisson's Ratio: 0.3

Rail Extension: Material - NORYL GTX™ 910

Density: 1107 kg/m^3

Young's Modulus: 2.5 GPa

Poisson's Ratio: 0.4

6. Damping input

Damping parameter is a critical input in vibration simulations. It needs to be specified to prevent unbounding of the responses if the forcing frequency is equal to an Eigen frequency of the structure (Thompson, 1996). Damping can be defined for all or some of the modes used in the response calculation. It can also be given for a specified mode number or for a specified frequency range if we have the damping values available for frequencies (Abaqus, 2009). However, most common way is to define a constant damping value for all the modes.

The damping value can be obtained from the Dynamical Mechanical Analysis (DMA) test results (Menard, 2008). DMA is a technique where a small deformation is applied to a sample in a cyclic manner and material's response to stress, temperature, frequency and other values are studied. A force motor is used to generate the sinusoidal wave and cyclic displacement response is measured through a Linear Variable Differential Transformer (LVDT). The resulting displacement (or strain) displays a phase lag with respect to the applied sinusoidal force (or stress) as illustrated in Figure 9. This is a measure of damping of the material.

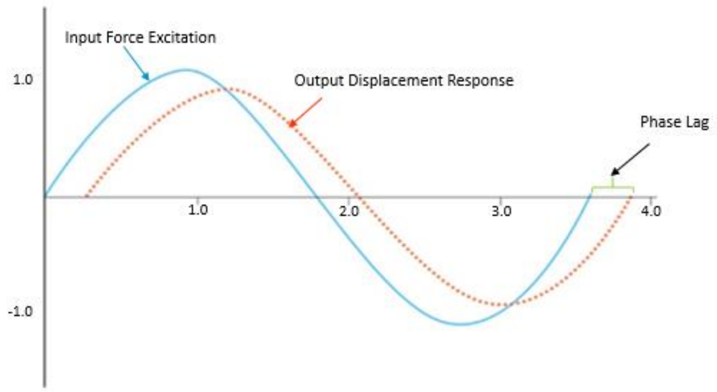

Figure 9 Phase lag of output displacement response to applied sinusoidal force showing damping

In-phase component is expressed in terms of the storage modulus, and an out of phase component in terms of the loss modulus. The storage modulus, either E' or G', is the measure of the sample's elastic behavior. The ratio of the loss to the storage is the "tan δ" and is a measure of the energy dissipation of a material or damping.

In order to use this damping information in the analysis, we need to convert "tan δ" in to damping coefficient or damping ratio (ξ). The relationship between ξ and "tan δ" is as shown below (Rao, 2001).

$$\xi = 0.5 * \tan \delta$$

DMA test results for NORYL GTX™ material is shown in Figure 10. As can be seen, tan delta for the material at room temperature is 0.064. From above equation, damping ratio to be used in the analysis is 0.032 or a damping coefficient of 3.2%. This is much higher compared to metals (Pure Aluminum has a damping coefficient of 0.95%) (Chung, 2001).

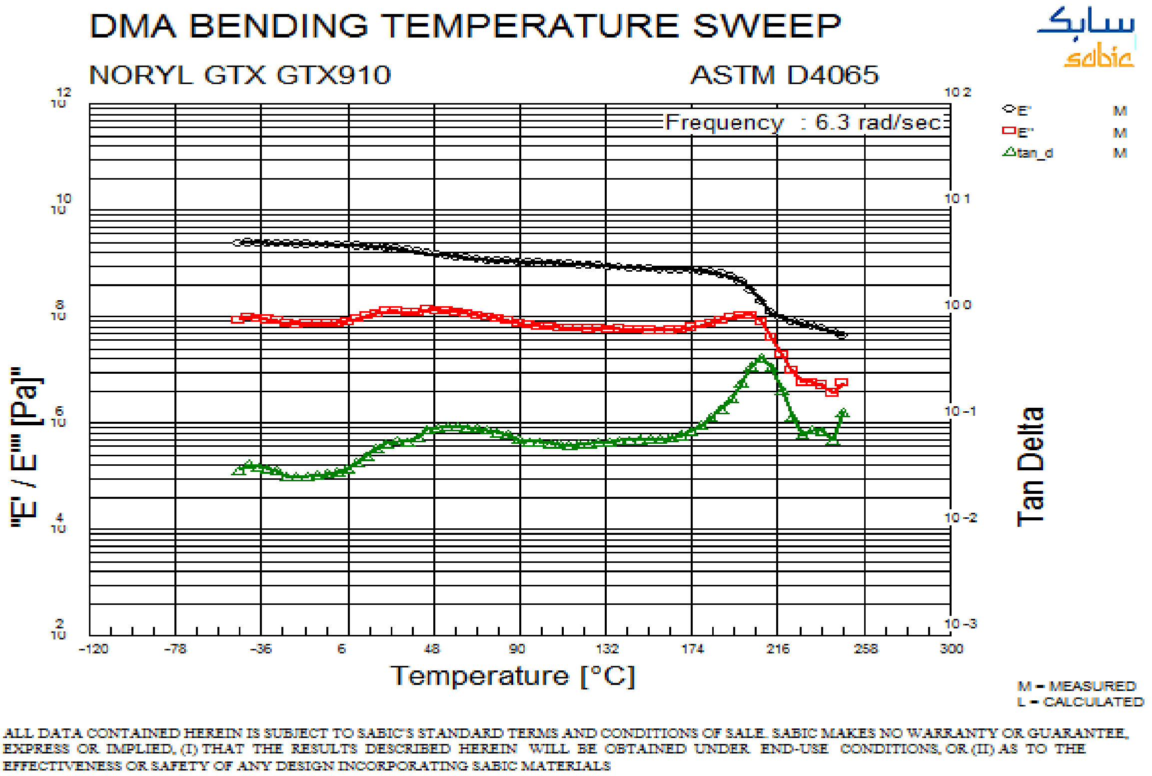

Figure 10 DMA test results for NORYL GTX™ 910

7. Random response specific inputs

Random vibration (or response) analysis is a two-step simulation which is performed in frequency domain and based on using a subset of the modes of the system which must first be extracted by using the Eigen frequency extraction procedure (Abaqus, 2009). Hence the first step should be an Eigen value simulation to find out the normal modes of the structure. The number of modes extracted must be sufficient to model the dynamic response of the system adequately, which is a matter of engineering judgment. As a rule of thumb, the number of modes should cover the entire frequency range covered by the input excitation. This may require running the modal analysis a few times to decide on actual no of modes required for random response analysis. The frequency analysis is used to generate the transfer function. The input PSD multiplies the transfer function to form a response PSD (Guo, 2008).

Random response analysis is performed as the second step where specific random response inputs such as damping values, frequency range of interest etc. are specified and outputs such as RMS stress value are requested. The details about damping value has been covered in previous section. Other specific random response inputs are covered here.

a. Frequency range and number of modes of interest:

Frequency range of interest should be specified looking at the random vibration excitation input spectrum. The lower and higher ends of interest are taken as 5Hz and 500Hz to cover the entire loading spectrum. Five number of calculation points is specified to reduce analysis time. Hence, the response is calculated at three points between the lowest frequency of interest and the first Eigen frequency in the range, between each Eigen frequency in the range, and between the last Eigen frequency in the range and the highest frequency in the range as illustrated in Figure 11.

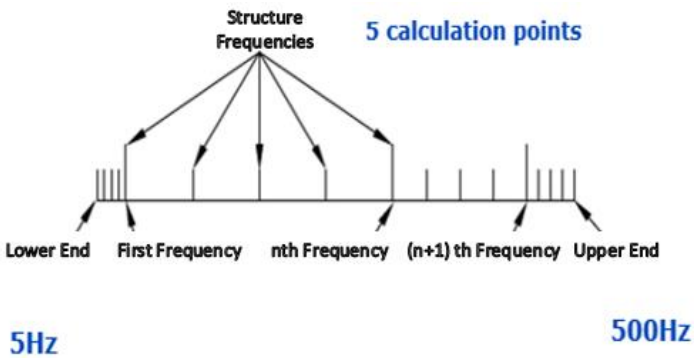

Figure 11 Frequency range and number of calculation points

b. Selecting modes to be used:

The modes to be used in modal superposition should be selected in random response load step. The modes can be selected individually by specifying a frequency range or through automatic mode number generation by Abaqus®.

c. Base motion and cross-correlation:

The excitation is defined by the base motion and is assigned to a numbered load case through the *BASE_MOTION keyword. Here the motion input type is specified as an acceleration input along with direction of motion (degree of freedom of motion).

This load case is then referenced in the cross-correlation definition. Cross-correlation is between the applied base motions. Three types of correlation can be defined: correlated, uncorrelated and moving noise. Moving noise type is used only in cases of random loads, not random base motions. Between correlated and uncorrelated types, correlated approach is more comprehensive where all terms in the cross-spectral density matrix are considered, which implies that the loads on all degrees of freedom within the load case are fully correlated (statistically dependent on each other). Hence, correlated type cross-correlation is defined.

Power spectral density function is also referenced in the correlation definition. PSD function is specified outside of the analysis step.

d. Output:

Required outputs are requested through *OUTPUT card. In random response analysis, the value of a variable is its power spectral density. This is post-processed to make sure the responses peak at the structure natural frequencies. Root mean square values of the Von-Mises stress is output through RMISES variable as the final output from the analysis to check for fatigue life.

8. Results

STEP-1: NATURAL FREQUENCY ANALYSIS

The items of interest from this step are normal mode values and mode shapes. Primary Eigen mode values are tabulated in Table 2.

Mode #	Frequency (Hz)	Mode Description	Primary Motion
1	63.332	Pitching Mode	Z-Direction
2	93.041	Lateral Mode	Y-Direction
3	95.902	Breathing In/Out	X-Direction

Table 2 Primary Eigen modes in three directions

STEP-2: RANDOM RESPONSE ANALYSIS

The items of interest from this step are root mean square (RMS) values of field variables, particularly stresses and deflections. For the current problem, primary interest is to ensure that the part has a fatigue life as specified. In the simplest sense, design is checked for an infinite life condition. For that, maximum RMS Von-Mises stress is checked against the endurance limit. In the current problem, we have used Segalman method of calculating RMS Von-Mises stress which is an efficient method of calculation without sacrificing accuracy and is output directly through ABAQUS™ solver (Segalman, 1998). To ensure 3σ level of confidence, RMS (or 1σ) Von-Mises (or equivalent) stress is multiplied by 3 and checked against endurance limit of the material.

Figure 12 shows variation of maximum Von-Mises stress PSD against frequency for the three loading spectrums. As can be seen, the stress response spectrum clearly shows resonance conditions at the structure normal modes. The Von-Mises stress is seen to be maximum for Z-direction loading. Figure 13 shows the variation of RMS (or 1σ) Von-Mises stress against frequency for Z-loading which is maximum from among all directional loading. It can be verified that the maximum RMS stress in Figure 13 is the square root of the area under the maximum Von-Mises stress PSD curve for the corresponding directional loading. Figure 14 shows a RMS Von-Mises stress contour at the maximum frequency of interest for Z-loading.

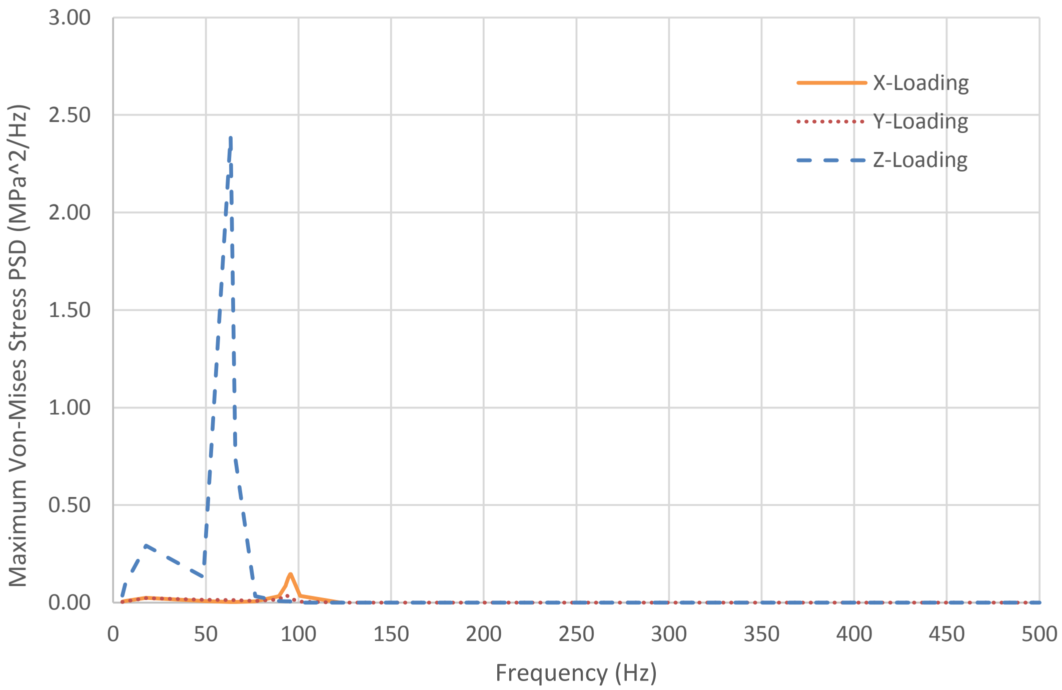

Figure 12 Variation of maximum Von-Mises stress PSD vs. frequency for the three loading spectrums

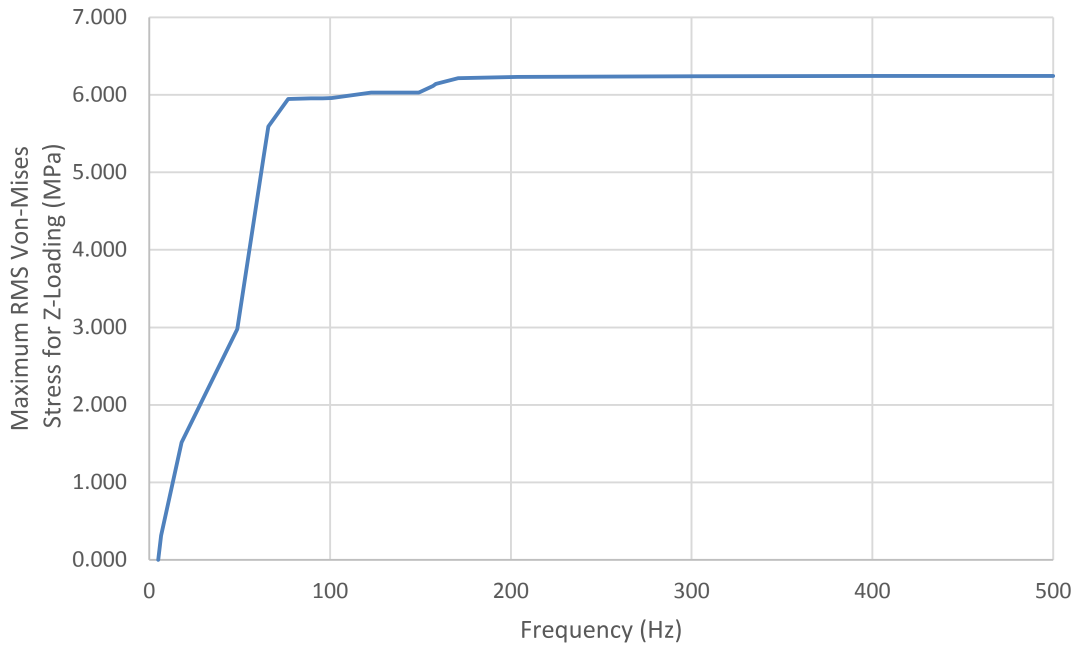

Figure 13 Variation of maximum RMS Von-Mises stress vs. frequency for z-loading

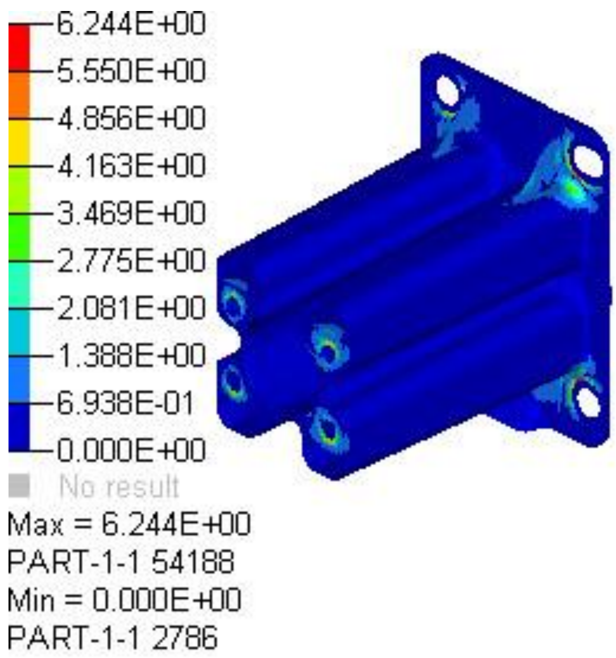

Figure 14 RMS Von-Mises stress contour at the maximum frequency of interest for z-loading

9. Life prediction

To estimate the life of the part, the maximum RMS Von-Mises stress is used. Maximum RMS (or 1σ) Von-Mises stress under the given loading is 6.244MPa. To check for infinite life (million cycles) at 1σ, 2σ and 3σ levels, we multiply this stress value by 1, 2 or 3 respectively. This results in maximum 1σ, 2σ and 3σ equivalent stresses of 6.244, 12.488 and 18.732MPa respectively. These are then compared against the endurance limit of the material. Figure 15 shows the stress-life (S-N) curve for NORYL GTX™ 910 material which shows a stress of 25MPa for a million cycles which is taken as the endurance limit. Comparing maximum equivalent stresses at different confidence levels to endurance limit of 25MPa, design is safe for infinite life (to withstand million cycles) at a 3σ confidence level (99.73% probability that the part will endure a million cycles without failing) (Joglekar, 2003).

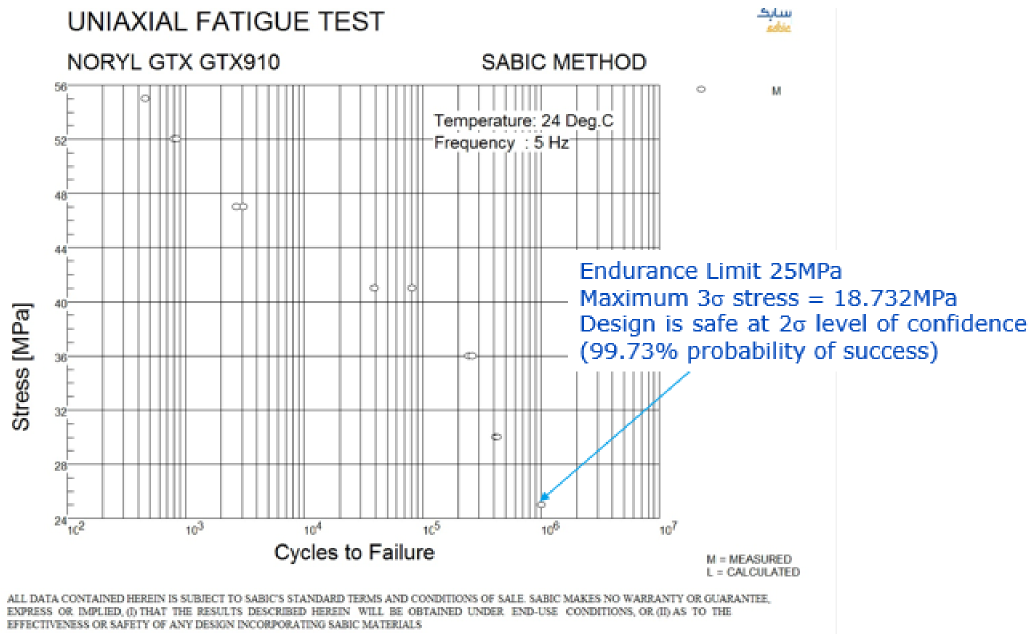

Figure 15 Stress-Life (S-N) curve of NORYL GTX™ 910 material

10. Conclusions

Thermoplastic rail extensions provide weight savings, reduce part count and provide higher material damping compared to metallic rail extensions while maintaining similar online painting capability. While being lightweight, these alternative solutions retain ability to sustain random vibration loads due to road undulations similar to current metal designs. This paper numerically verified the random vibration capability of a typical thermoplastic rail extension design using ABAQUS® commercial code which proved to be an efficient tool to evaluate the part performance under random vibration load conditions.

11. References

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12. Acknowledgements

I would like to thank my managers Mr. TK Poovanna and Dr. Scott Fisher and other SABIC management staff for their permission to publish this paper. My sincere thanks to Mr. Haridranath Sharma, Dr. Prasad Dasappa, Dr. Sekar Pandian and Mr. Dhanendra Nagwanshi of SABIC Innovative Plastics for going through the manuscript of this paper meticulously and providing valuable suggestions for improvement. Thanks are also due to Mr. Arunachala P for support in CAD related work and Mr. Bhusan Deshmukh for help in generating illustrative figures for the paper.

DEFINITIONS/ABBREVIATIONS

CAE	Computer Aided Engineering
OEM	Original Equipment Manufacturer
PSD	Power Spectral Density
DMA	Dynamical Mechanical Analysis
MPC	Multi-point constraint
LH	Left Hand
RH	Right Hand
LVDT	Linear Variable Differential Transformer
PPE	Polyphenylene ether
PS	Polystyrene
PA	Polyamide