

Considerations of Vibration Fatigue for Automotive Components

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ABSTRACT

As an automobile is driven, its components and structures consistently experience the random excitations from road inputs and periodic vibration from engine firing. This could cause durability issues if the component structure isn't fully validated. Vibration fatigue is a field of study regarding the assessment and improvement of a component's or system's robustness to vibration inputs.

This paper introduces aspects of vibration fatigue to help designers, release engineers, and test engineers better understand the requirements, testing methodologies available, and strategies for improving vibration fatigue performance for the design and validation testing of their products. Vibration characteristics such as typical vibration levels and frequency content for varied areas in the automotive environment are introduced. Methodologies available for conducting actual vibration testing are introduced with listed advantages and disadvantages. Finally, aspects that could improve a component's or system's vibration fatigue durability are recommended.

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INTRODUCTION

Proving grounds testing is often the last phase of durability validation testing for automotive components and systems. Performance issues that may arise during this phase of testing can require design revisions and additional validation testing with subsequent delays in vehicle launch. This paper focuses on the importance of understanding vibration fatigue requirements for a given component and employing proven test strategies to reduce risk and promote successful vehicle PG validation testing.

When a component is designed, the focus is typically on functionality, weight, cost, durability, packaging, etc. Vibration fatigue analysis can be limited due to a lack of dynamic vibration information or awareness. However, there are studies that could and should be performed to eliminate future vibration fatigue issues. In this paper several real-world examples are given to illustrate the root cause of past issues and what studies should be conducted at the initial design phase to avoid them. These studies cover typical areas in the automotive environment including components mounted to the powertrain, body, and frame.

Replicating the issues seen from the field or at the proving ground on a shaker bench is a difficult task because it involves many aspects such as boundary conditions, acceleration amplitude, customer usage characteristics, and methods used for the validation process. It is generally accepted that boundary conditions on the actual shaker table must be very similar to that on the vehicle. Acquisition of the actual vibration amplitude at the given customer usage condition or similar proving ground condition could be applied as vibration input. However, no papers are published that fully discuss what type of test

should be used for replicating vibration failures. In this paper, the methodologies available with their advantages and disadvantages are discussed to provide a guideline for future references. Application limitations and recent developments are noted.

In addition, aspects that could improve vibration fatigue performance are clearly presented in this paper. Fundamental theory is provided and examples of each recommendation are introduced for given failure examples in the early section of the paper. In summary, this paper tries to provide insights on vibration fatigue to designers, release engineers, and test engineers for designing and validating products to meet durability requirements.

VIBRATION CHARACTERISTICS FOR AUTOMOTIVE COMPONENTS

When designing an automotive component, there are numerous design best practices considering manufacturing capability, assembly difficulty, quality assurance, etc. For a special requirement, such as consideration of vibration fatigue, designers usually do not have enough experience. Thus, to provide design guidance, characteristics for vibration in automotive application are introduced in two separate categories; engine mounted components and body/frame mounted components. Real world examples are followed to better explain the guidance.

Case Study 1: Engine Mounted Components

Components mounted on the engine are dominantly experiencing the periodic excitation from engine firing. Thus, almost all the focus when studying failure of engine mounted components is on engine order vibration. Typically acceleration will be mounted on the

engine block or cylinder head for obtaining excitation inputs. Accelerations are measured during a sweep from engine idle to fuel cut off speed with full load condition at wide open throttle (WOT). Waterfall analysis is performed to study and extract the dominant engine orders. If a resonance frequency exists in the structure, a highlight will be observed in the waterfall results. The physical test (either frequency sweep test or swept sinusoidal test) will then be developed and conducted after scaling the engine orders for a reasonable test duration and a given fatigue cycle target.

The optimal design goal, with respect to engine vibration, is to avoid any structural resonance within normal engine speed operating range. The fundamental frequency range during engine operating range can be determined as shown in Equation 1. However, dominant engine orders are more focused on since the vibration levels are commonly highest among all engine harmonic orders. The frequency range of the dominant order can be obtained and depends on the type of engine as shown in Equation 2 (assuming four stroke engines). For example, a four stroke I4 engine sweeps from 600 rpm to 6000 rpm and has dominant order of 2nd order frequency range of 20 Hz to 200 Hz.

$$f_{Base\ Order} = \left(\frac{RPM_{idle}}{60}, \frac{RPM_{fuel\ cut\ off}}{60} \right) \quad (1)$$

$$f_{Dominant\ Order} = \left(\frac{RPM_{idle}}{60} \times Cylinder\#_{in\ operation} \times \frac{1}{2}, \frac{RPM_{fuel\ cut\ off}}{60} \times Cylinder\#_{in\ operation} \times \frac{1}{2} \right) \quad (2)$$

Besides the resonance frequency itself, the amplification factor (Q) and damping are also important factors. As shown in Figure 1 for the case of an inline 4 engine, the resonance frequency of the component (520 Hz) is beyond the dominant engine operating frequency range (200 Hz). However, if the same component is installed on a V8 engine which has 4th order dominance and the frequency goes up to 400 Hz, the engine excitation will be amplified 2.6 times (at 400 Hz) which could cause issue. If somehow the damping is increased for this component, the amplification factor and vibration level will then be decreased. Interestingly, although the resonance frequency of the part is away from the dominant engine orders in the case of the I4 engine application, the component could still be damaged by other harmonic engine orders at the condition of a high Q as shown in this example. Even at a tiny vibration excitation of 0.1 g, the component will experience an acceleration of 8.3 g response at the resonance frequency (520 Hz).

Besides components mounted on the engine block or cylinder head, there are many cases of components mounted on accessories. Special attention should be paid not only to engine order excitation on the block, but also to the frequency response of the accessory where the component mounts to. One component of interest mounts on the air induction duct of an I4 engine. The air induction duct is installed on the engine at three attachments; one mounts to the cylinder head, one mounts to air box to body, and one mounts to the throttle body using a band clamp. Vibration responses were measured on vehicle level at

the proving ground on the component, throttle body, air induction duct, and the fastener location where the duct is mounted to the cylinder head. WOT sweeps and waterfall analyses were conducted to study the frequency content at each location. Figure 2 shows the waterfall result in the vertical direction of the component measured by a tear drop accelerometer. Several severe responses are observed from the figure such as during the engine speed ranges 2200 rpm to 3200 rpm, 4300 rpm to 4900 rpm, and above 5600 rpm. Those peak resonances occur during the engine operating range and thus should be improved. Based on the frequency content analysis, both the air induction duct and throttle body contribute to the vibration excitation input to the component, with resonance of the air induction duct contributing to the two low speed excitations and the throttle body contributing to the response above 5600 rpm.

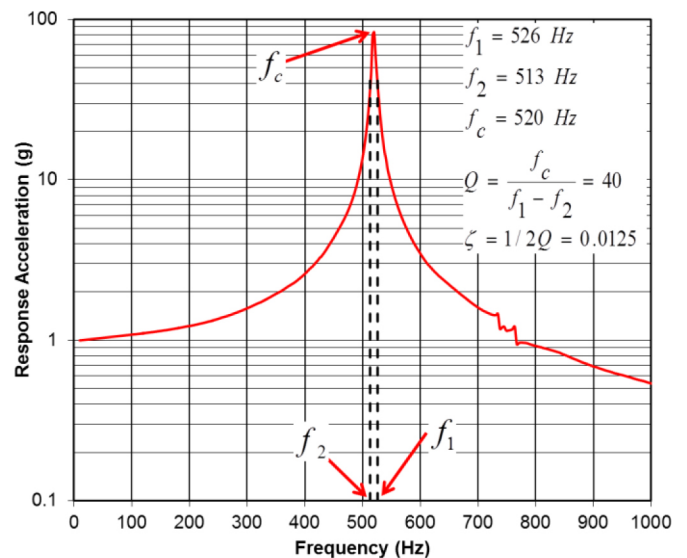


Figure 1. An example of a structure's frequency response (at a given 1 g excitation input)

As automotive OEMs compete with each other, engines with improved fuel economy (more power but less weight) continue to be produced using turbochargers, superchargers, Multi-air technology, etc. Those engines introduce extraneous vibration excitations besides the main engine firing excitation. For example, the primary shaft of an engine accessory operates at 2.35 times crankshaft speed and has 5 teeth introducing a 11.7th order vibration which is high frequency (above 1000 Hz) and high amplitude. This is the region where electronics (soldering or wear type failure) are sensitive to vibration. For components mounted close to or on that accessory, special attention should be paid to that frequency order.

Ideally engine mounted components should be designed having the first resonance frequency above the engine operating range. However, in practice, there are components whose resonance frequencies cannot be improved upon without great difficulty such as engine covers and air induction ducts. For example, the engine cover lowest resonance frequency usually is in the range of 20 Hz to 30 Hz which causes it to shake each time the engine speed passes through this range. Thus, typically engine covers are not hard mounted on the engine and rubber grommets are used to provide more damping effect

to reduce the overall vibration level. Additional cautions should be paid to validate the durability of this type of component, and several rounds of validation by both simulation and physical tests are needed.

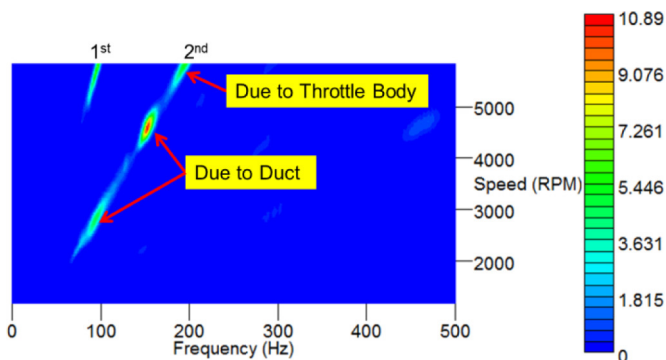


Figure 2. Waterfall analysis result for the acceleration measured on the component in vertical direction

It is commonly seen as a requirement from the design release or simulation groups that acceleration on certain parts needs to be lower than a particular value. For example, the engine wire harness group typically requires the acceleration on the harness connector be lower than 15 g. However, it should be very clear that those requirements can only be used for static simulations such as simulating the stress/strain value on the components based on that given acceleration. For any dynamic simulation, this requirement is not valid because a component has varied acceleration levels at different frequencies.

Case Study 2: Body/Frame Mounted Components

Frame or body mounted components mainly experience random excitations from the road surface compared to the periodic excitation for engine mounted parts. Damage to the body/frame mounted components will be affected by the acceleration excitation amplitude, excitation frequency, and the frequency response of the component itself. Figure 3 shows the maximum acceleration amplitude versus frequency (amplitude spectrum) for common road surfaces. Note that those vibration amplitudes are different in concept from those commonly seen in time series data. Acceleration in a raw time series format is a combination of amplitudes for all frequency content. For example, peak time series acceleration corresponding to the longitudinal amplitude spectrum will be much higher than 1.4 g as shown in the peak amplitude spectrum in Figure 3.

As shown in Figure 3, the vibration level in the vertical direction is the highest overall while the vibration level in the longitudinal direction (fore/aft direction) is the lowest. The maximum acceleration at each frequency is shown. Thus, if a frequency response curve of a structure is available (as shown in Figure 1), the expected peak acceleration could be predicted and used for CAE strength or durability simulation. The curve shown in Figure 3 can also be used for running the actual test. For example, for a structure with known resonance frequency of 25 Hz, a frequency sweep test using these curves could be conducted till failure. However, these curves have not been recommended to be a pass/failure criterion as they are just an enveloped amplitude spectrum and do not have any duration requirement.

The second take away from Figure 3 is the dominant frequency content of road excitation input. As shown, peak amplitudes occurs 9 Hz to 13 Hz, 9 Hz to 40 Hz and 9 Hz to 24 Hz in longitudinal, lateral and vertical directions respectively. Amplitude gradually decreases outside those frequency ranges. Adding a safety factor is generally recommended for body/frame mounted components where the resonance frequency should be at least above 50 Hz. In the frequency range above 50 Hz but lower than 100 Hz, there are still observable vibration excitations. Thus, for better confidence, above 100 Hz could be a design criterion. However, it should be noted that in practice a tradeoff usually has to be made considering cost, durability, packaging etc.

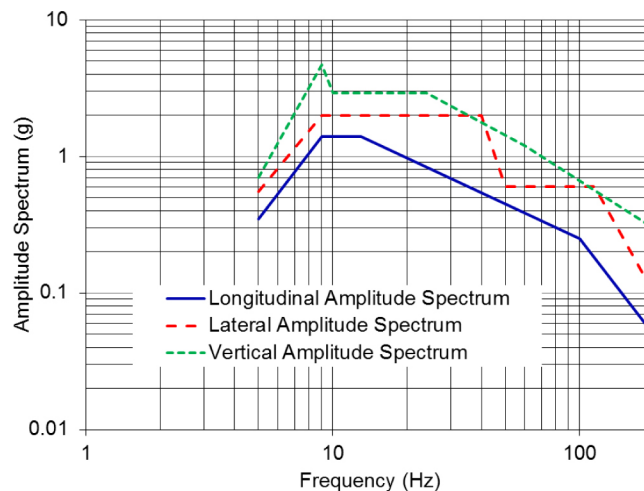


Figure 3. Peakhold amplitude spectrum for common road surfaces

Another key factor for the body/frame mounted component is how many times an acceleration level occurs (same question as how long the duration should be for different acceleration levels). Figure 4 shows a worst case input acceleration histogram distribution for body/frame mounted components that could be experienced during their whole design life. This is developed by conducting level crossing studies for accelerations measured at various locations on the body/frame and from compact cars to heavy duty trucks. Repeats have been scaled to account for the whole design life of the vehicle such as 150,000 miles for cars. As shown, acceleration input to the body/frame could be as high as 15 g, 8.5 g, and 6 g in vertical, lateral, and longitudinal directions respectively. Figure 4 provides a reference to determine the acceleration level when trying to replicate failures without having the actual acceleration information collected. Note, Figure 4 is a worst case and it is always recommended to have the actual vibration signals collected for a more representative simulation or experimental validation.

One important lesson learned for body/frame mounted components is that one should always avoid a cantilever design. There is a urea system bracket which is mounted to a frame using three bolts at frame side while totally free on the other end. A resonance sweep test was conducted on the bench in the vehicle lateral direction with a response accelerometer installed on the bar at the top of the bracket.