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# Comparison of Vibration Durability Test Specifications with respect to Fatigue Damage & Validation Study

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## Abstract

In this study, Random and Harmonic vibration durability test specifications for Automotive Lighting systems are compared according to their damage content. Harmonic and Random vibration durability test specifications which have different durations and Acceleration Power Spectrum Densities (PSD G) are plotted and compared according to damage accumulated on single degree of freedom (SDOF) system. Outcome of this study is the fast evaluation of fatigue strength of common or similar parts used on lighting systems which are dedicated to different customers, different car segments; and evaluation of the severity of new customer vibration durability test specifications and evaluation of special design precautions for new design projects. Shorter duration PSD G's and Harmonic sweep tests which have equivalent Fatigue Damage Spectrums (FDS) can be created and instant design improvements can be tested and evaluated. In addition, in this study, random test time reduction formula was validated with physical tests.

**Keywords:** FDS, Random Vibration Durability Test, Sine Sweep Test, Harmonic Vibration Durability Test, Accelerated Testing

## 1. INTRODUCTION

Automotive Lighting Systems are tested in 3 axis direction with specified duration of acceleration power spectrum density (PSD G) and Harmonic Sweep g inputs defined on Automotive Manufacturer's test specifications on the vibration shaker at the laboratory. At the end of these durability tests, there must not be any cracks, breakage on the lighting assemblies and their detail parts. PSD G acceleration input has been calculated from the acceleration time histories measured on the specified point of the car body during field tests and created by transformation of them to frequency domain with statistics methods. PSD G represents the energy transferred in each frequency to the device under test. Every customer vibration durability specification has different PSD G levels with different durations. Assembly strategy of automotive lighting system to car body, mechanical structure of the assembly brackets, internal detail parts assembly strategies and their materials are defined and designed similarly from project to project in general. Resonance frequencies of the Head Lamp assemblies, Rear Lamps, and individual resonances of detail parts are calculated by Finite Element Analysis (FEA) or by resonance search on the vibration shaker in the laboratory during design validation stage. If PSD G input curve and the resonance frequencies of the lighting device are inspected together, it is understood that there is need of shifting resonances of the lighting device over 100Hz in order to escape from high amplitude accelerations at low frequencies during resonance motion. Because each of the customers vibration test specifications have different PSD G levels and different test durations; it is required to compare severity of them. For instance, a common lighting module used on different lighting devices dedicated for different customers may pass vibration durability test of one customer specification but another device which has the same module used on a different lighting device dedicated for another customer may fail during vibration durability test of another customer. Furthermore, for the same customer, a lighting device used on a different car segment may fail during vibration durability test specified for another car segment. This is true for new vibration durability test specifications that are said to be implemented for the future projects of the customer. Also, some customers have still harmonic sweep vibration durability tests. One lighting device which passed a harmonic sweep durability test may fail during a random PSD vibration durability test or vice versa. If these concerns are taken into account; it is clear that there is need of a medium in which different vibration durability test specifications can be compared. This medium can be the FDS, "Fatigue Damage Spectrum".

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Kihm, F., Halfpenny, A., and Munson, K.,(2016) in their study; they have demonstrated creating PSD vibration durability test profile from a stationary time domain signal which has same fatigue damage spectrum. Also, they have proposed a method for further reducing the test duration by using a more realistic non-stationary, non-gaussian random excitation signal which has high Kurtosis. Similar study has been performed by DeJong, R., Jung, S., and Van Baren, J., (2016). The they have demonstrated increasing the FDS level by mixing impulses with random excitations.

Kihm, F., Halfpenny, A., and Beaum, B. (2015) studied on an automotive engine or gear box mounted component which is subjected to loading from both engine and random road profile, in order to evaluate its mechanical durability. They have explained how to create swept-sine-on-random vibration durability test profile by accelerated testing approach based on Fatigue Damage and Extreme Response Spectra. Similar study on an automotive engine mounted component has been performed by Wang, L., Burger, R., Lee, Y. and Li, K (2013). They have devoloped a random vibration durability test profile from a customer usage data by an approach based on FDS.

Wei-Lun Chang, Ken-Yuan Lin, Chin-Duo Hsueh and Jung-Ming Chang, (2011), have explained the methods for devoloping a random vibration durability test specification from a time domain signal which is derived from proving ground or customer usage. Furthermore, they have performed several vibration durability tests on an automotive headlamp and determined b value (Basquin Exponent) of the materials by the use of vibration durability test acceleration formula. One can find several other studies in the literature about creating vibration durability test profiles based on equivalent FDS approach.

In this study, severity of several vibration durability test specifications for automotive lighting products have been compared by calculation of their FDS's. Furthermore, validity of accelerated testing formula for plastic parts has been inspected by performing several vibration durability tests on an automotive Day Time Runing Light (DRL).

## 2. FATIGUE DAMAGE SPECTRUM (FDS) CALCULATIONS

### 2.1 FDS Calculation Method from a Random PSD for a SDOF system

If the average value of the signal does not change with respect to time and conforms to Gaussian distribution, after PSD is defined, FDS can be constructed by the application of below methodology (Lalanne, C., 2002):

Calculating the one degree of freedom system response by the use of PSD and calculating the root mean square (RMS) values of relative acceleration, velocity and displacements.

Calculation of average frequency and average maximums at the unit time.

Calculation of system response irregularity factor.

Calculation of peak probability density.

In order to calculate FDS of a Random PSD G, Lalanne gives below formulation.:

$$D=\frac{K^b}{C}\times\frac{n_p^+\times T}{z_{rms}}\times\int_0^{+\infty} z_p^b\times\left\{\frac{\sqrt{(1-r^2)}}{\sqrt{(2\times\pi)}}\times e^{-\frac{z_b^2}{2\times(1-r^2)\times z_{rms}^2}}+\frac{r\times z_p}{2\times z_{rms}}\times e^{\frac{-z_p^2}{2\times z_{rms}^2}}\times\left[1+erf\left(\frac{r\times z_p}{z_{rms}\times\sqrt{2\times\left(1-r^2\right)}}\right)\right]\right\}dz_p \quad (1)$$

With a given damping ratio and b value (slope of S-N curve), $D(f_0)$ can be calculated by changing the $f_0$ value. If K and C are unknowns, they could be left as 1 for comparative calculations.

This formula needs to be numerically integrated. A simplified formula which gives an approximate value proposed by Plasskit and Halfpenny as below (Plaskitt, Halfpenny,2015):

$$FDS\left(f_0\right)\cong f_0\times T\times\frac{K^b}{C}\times\left[\frac{Q\times G\left(f_0\right)}{2\times\left(2\times\pi\times f_0\right)^3}\right]^{\frac{b}{2}}\times\Gamma\left(1+\frac{b}{2}\right) \quad (2)$$

### 2.2 FDS Calculation Method for SDOF System from a Constant g Harmonic Sweep

Lalanne gives the formula as below(Lalanne, C., 2002);

$$D=\frac{K^b}{C}\times f_0\times\frac{T_1\times\ddot{x}_m^b}{\left(4\times\pi^2\times f_0^2\right)^b}\times\int_{h_1}^{h_2}\frac{dh}{\left[\left(1-h^2\right)^2+\frac{h^2}{Q^2}\right]^{\frac{b}{2}}} \quad (3)$$

This formula needs to be numerically integrated. A simplified formula which gives an approximate value proposed by Plasskit and Halfpenny as below(Plaskitt, Halfpenny,2015):

$$FDS\left(f_0\right) \cong \frac{K^b}{C} \times \frac{60 \times f_0}{\rho \times \ln(2)} \times \left[\frac{A\left(f_0\right)}{\left(2 \times f_0\right)^2}\right]^b \times \frac{\pi}{2} \times Q^{b-1} \times b^{\frac{-1}{\sqrt{\pi}}} \tag{4}$$

## 2.3 Random PSD G Durability Test Profile and Harmonic Sweep Durability Test Profile Examples and Comparison of their FDS's.

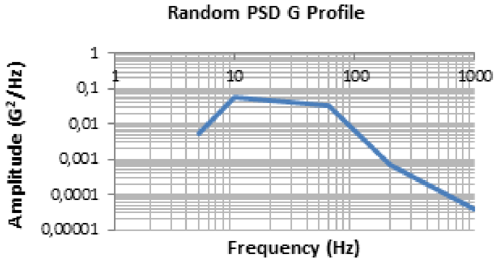


Figure 1. 8h duration Z axis, 5-1000Hz, PSD G rms: 1.81g

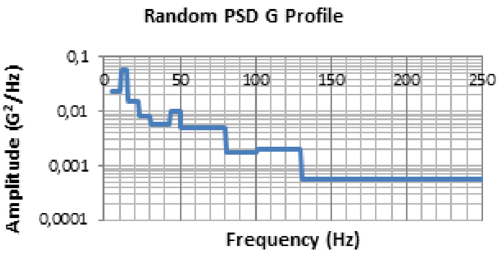


Figure 2. 24h duration Z axis, 5-250Hz, PSD G rms: 1.03g

Harmonic Sweep test, 8 hours duration between 20Hz – 220Hz, 4g constant acceleration, sweep rate 1 octave / min.

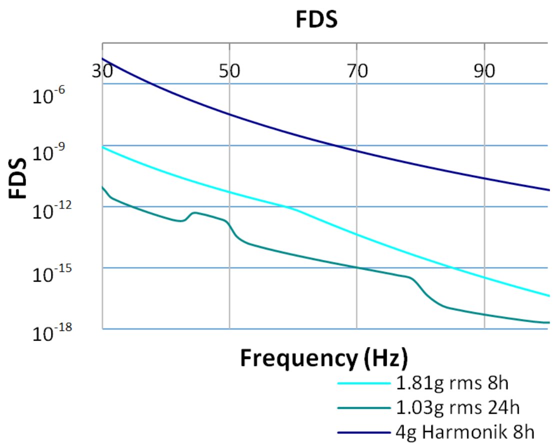


Figure 3. Comparison of FDS's of Random PSD G and 4g Harmonic Sweep

In Fig.3, it is understood that 8 hours duration test with a 1.81 grms value is more severe when compared to 24 hours duration test with a lower grms value of 1.03g. At all resonance frequencies, it is clear from the figure that 8 hours duration harmonic sweep with constant 4g amplitude test is the most severe one.

## 2.4 Creation of a Shorter Duration PSD G which has the same FDS

In order to see the effects of modifications on failed parts quickly, during product validation stage, instead of long duration Random PSD G tests, shorter duration PSD's which have the same FDS with the original PSD can be created. Average damage is proportional to test duration, to b constant and to grms value. In order to create a new PSD G below formula is used. In this formula, whatever the stress level, damping ratio is assumed to be constant. Lalanne proposed below formula (Lalenne, C., 2002);

$$\frac{G_{Reduced}}{G_{Real}} = \left(\frac{T_{Real}}{T_{Reduced}}\right)^{\frac{2}{b}} \quad (5)$$

As an example, an 8 hour duration PSD G can be converted to a 2 hour PSD G which has the same FDS. Figure 4 shows the result.

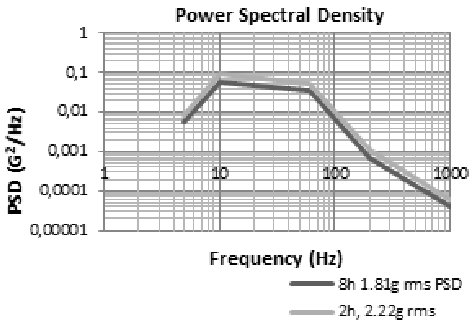


Figure 4. 8h PSD G and 2h PSD G

In order to validate, when FDS of 2 hour duration PSD G is calculated and compared with the FDS of 8 hour duration PSD G, it is understood that FDS values at all frequencies are coincide. However, ERS, extreme response of the system of 2 hour duration PSD G is higher. In order to avoid problems that can be arised from the Shock Response Spectrum (SRS), Extreme Response Spectrum (ERS) of accelerated test and SRS of field data must be compared. ERS of accelerated test must not be over SRS of the field data.

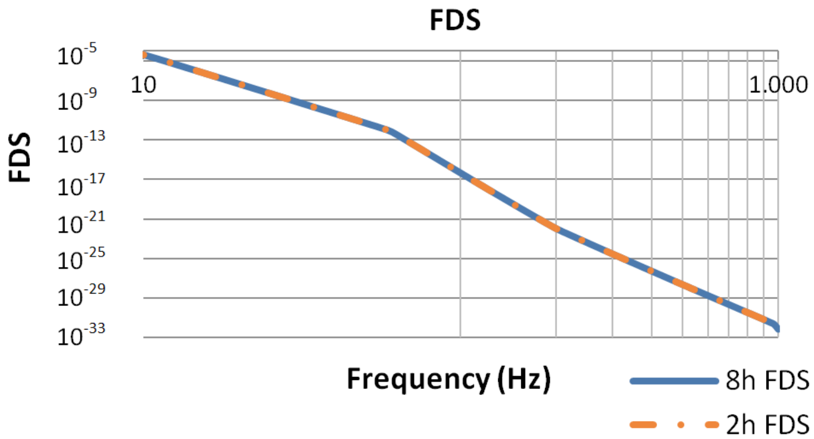


Figure 5. FDS curves for 8h duration PSD G and accelerated 2 h duration PSD G test

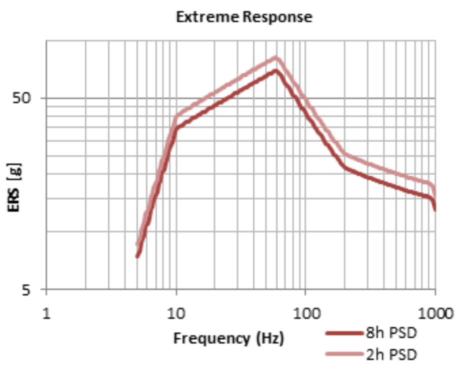


Figure 6. ERS curves for 8h duration PSD G and accelerated 2 h duration PSD G test

## 2.5 Creation of Shorter Duration Constant g Harmonic Sweep Test Which has Same FDS

Lalanne proposed below formula for constant g harmonic sweep tests (Lalanne, C., 2002);

$$\ddot{x}_{m\,reduced} = \ddot{x}_{m\ real}\left(\frac{T_{Real}}{T_{Reduced}}\right)^{\frac{1}{b}} \tag{6}$$

According to formula given, if 8 hour test with 40 mm/s$^2$ g is accelerated to 2 hours duration; 47.56 mm/s$^2$ acceleration is found.

Below figure shows that, FDS's of both original and the accelerated tests coincide at all resonance frequencies.

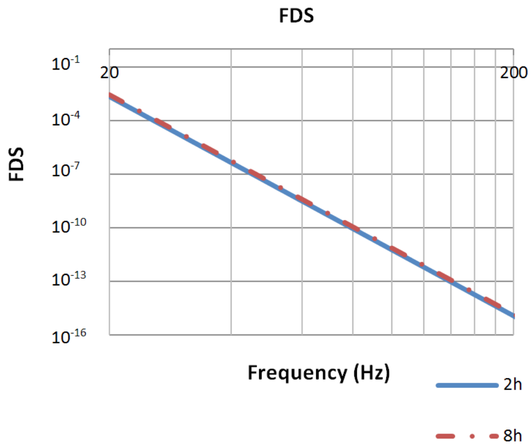


Figure 7. FDS for 8h constant g harmonic sweep test and accelerated 2h test

This study can be extended to calculate FDS's of shock tests.

In order to make comparisons ,below constants were assumed.

| | |
|---|---|
| k=1N/m3 | Stiffness |
| b=6.64 | Basquin Exponent |
| Q=50 | Damping Factor |
| C=1N/m2 | Basquin coefficient |

## 3. VALIDATION STUDY

In order to validate the applicability of formula proposed by Lalanne mentioned in previous paragraphs for accelarating random vibration durability tests of Automotive Lighting devices; 3 different grms value random PSD's applied on a Day Time Running Light (DRL) by an electrodynamic shaker. At the same time, these PSD's are applied as load inputs to the Finite Element Model (FEM) of DRL. Modal parameters like damping ratio's, resonance frequencies, modal constants, etc, of DRL was updated before by resonance search tests according to Modal Testing & Analysis Theory (Telli, Çetin, S. Ediz, B., 2017). As a result, updated FEM was used.

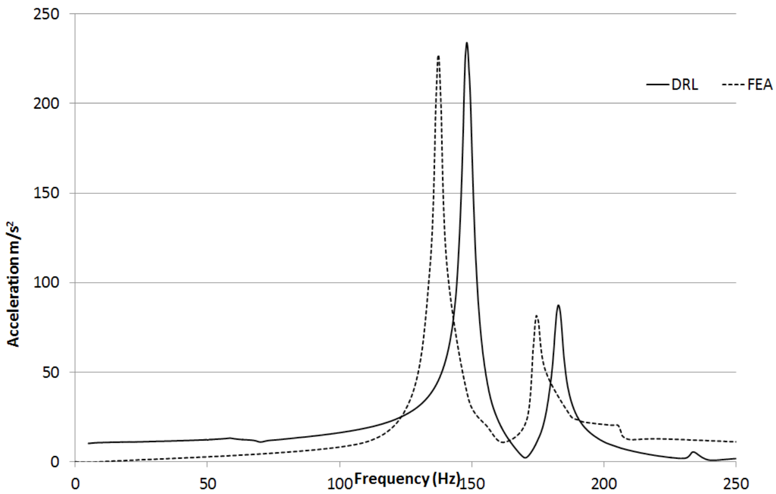


Figure 8. Comparison of resonance search result and FEA acceleration response result

By inspection of resonance frequencies, in order to resonate and break the part, a severe PSD G is created which has 3.37 grms value.

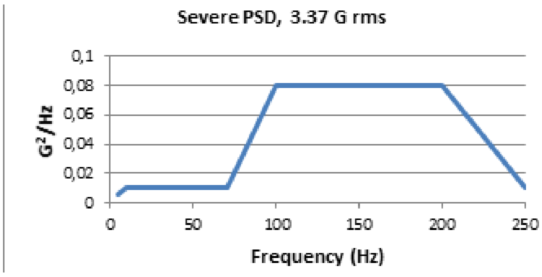


Figure 9. Severe PSD G

This PSD G was used as FEA load input for random vibration FEA and critical stress concentration region has been found as below.

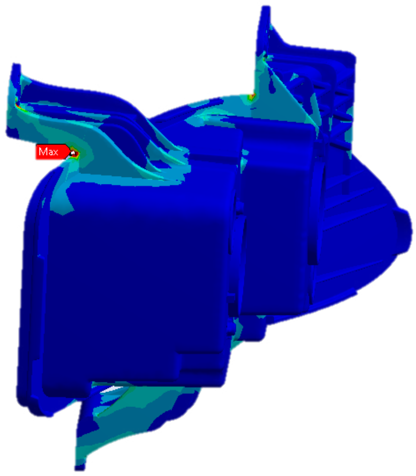


Figure 10. Stress Map and Critical Region Found after FEA Random Vibration Analysis

During vibration durability tests, acceleration signals have been collected by accelerometers from a defined point on DRL.

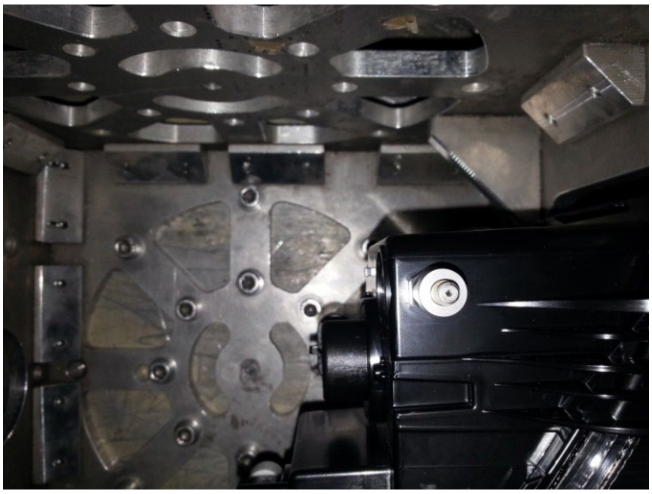

Figure 11. Accelerometer Position

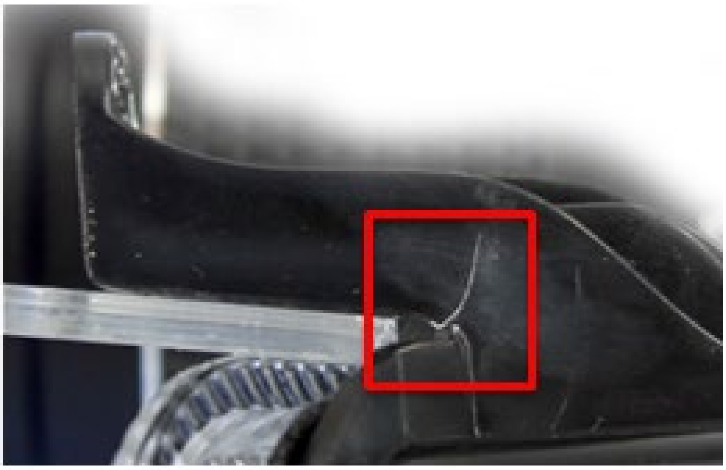

Figure 12. Crack Propagation on the forecasted critical area

After 2 hours; test stopped and as shown on fig. 12, on the critical region, a crack found as previously found after FEA. Crack propogation has been observed. When response acceleration time history shown in Fig.13 is inspected, it is found that after 2460s, there is an instant increase in acceleration response and as time passes the acceleration response continues to increase. As a result, it is concluded that crack started at 2460s instant and continued to propogate with time.

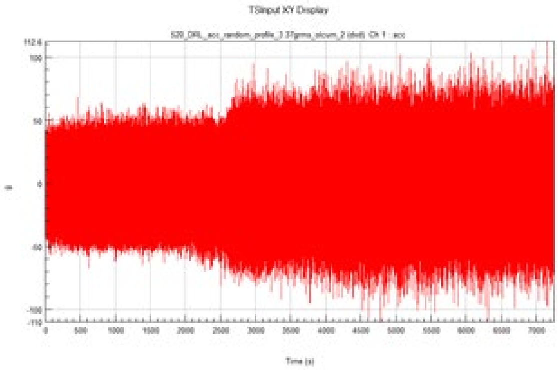


Figure 13. Time History of Acceleration Response

When Fast Fourier Transform (FFT) has been performed from the time history data until 2460s and after 2460s, and when frequency response graphs have been inspected, important difference between frequency responses has been observed. As a result, it is concluded that there was a structural change happened after 2460s. In Fig. 14 both acceleration frequency responses have been plotted for comparison.

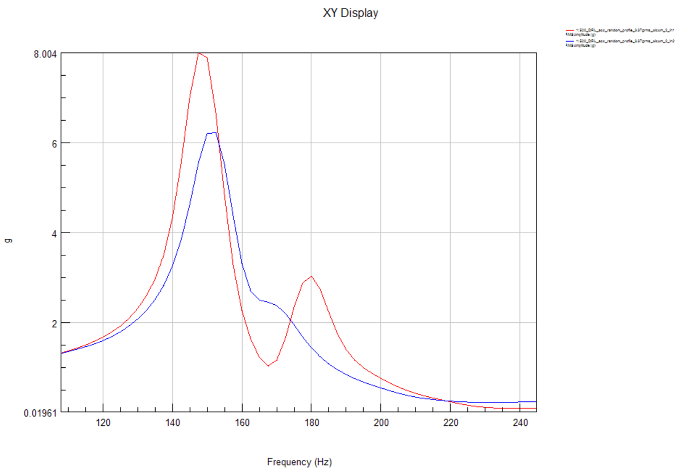


Figure 14. Acceleration Frequency Response Curves before and after crack

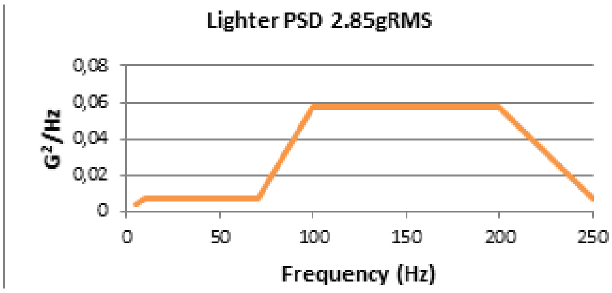


Figure 15. Lighter PSD G

As found before, after test repeated with a lower grms PSD (2.85grms) a crack has been found at the same location. When response acceleration time history shown on Fig. 16 inspected, this time it is concluded that after 7250s crack started and propogated with time. However, because input g amplitude is lower than before, response g levels are also lower and crack propogation is slower. As a result, detection of crack start point is becoming harder as the input g levels are getting lower.

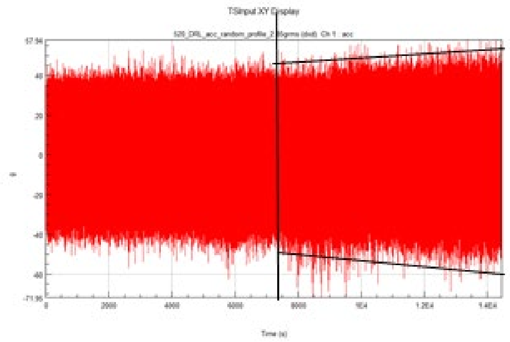

Figure 16. Time History of Acceleration Response

When the acceleration frequency response curves until crack and after crack have been inspected it is decided that a structural change has been happened.

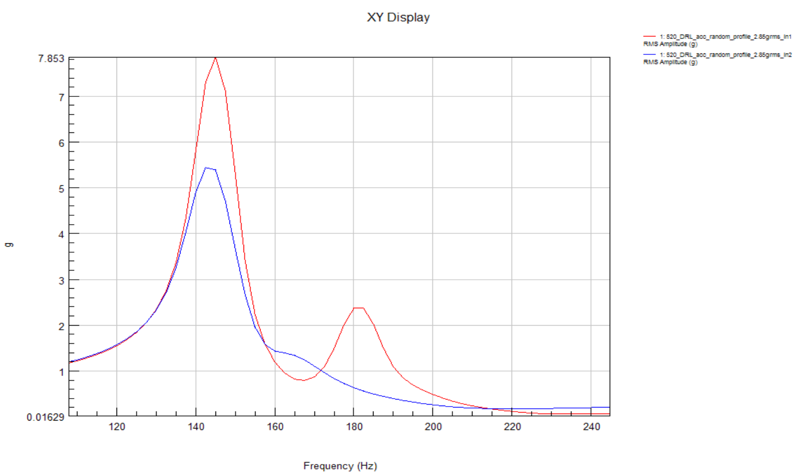


Figure 17. Acceleration Frequency Response Curves before and after crack

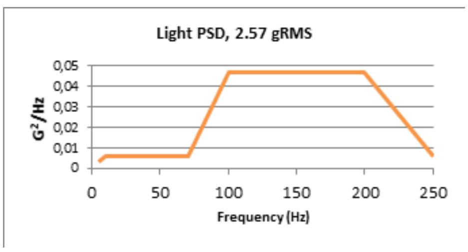


Figure 18. Light PSD G

As found before, after the test repeated with a too low grms level PSD (2.57grms) a crack has been found at the same location. When response acceleration time history shown on Fig. 19 inspected, this time it is concluded that after 14287s crack started and propogated with time. However, because input g amplitude is too low than before, response g levels are also too low and crack propogation is too slow. As a result, detection of crack start is becoming harder as the input g levels are getting lower. Drift was detected on the acceleration time history curve which may be due to longer measurement duration and heating of accelerometer.

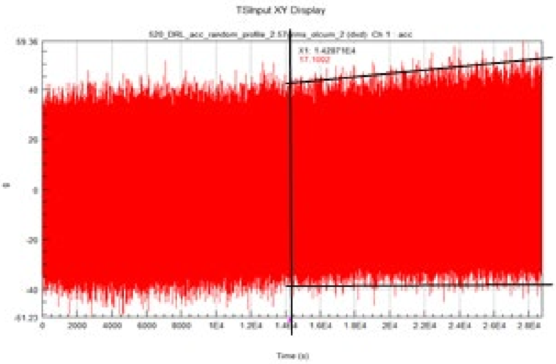


Figure 19. Time History of Acceleration Response

When the acceleration frequency response curves until crack and after crack have been inspected, it is concluded that a structural change has been happened.

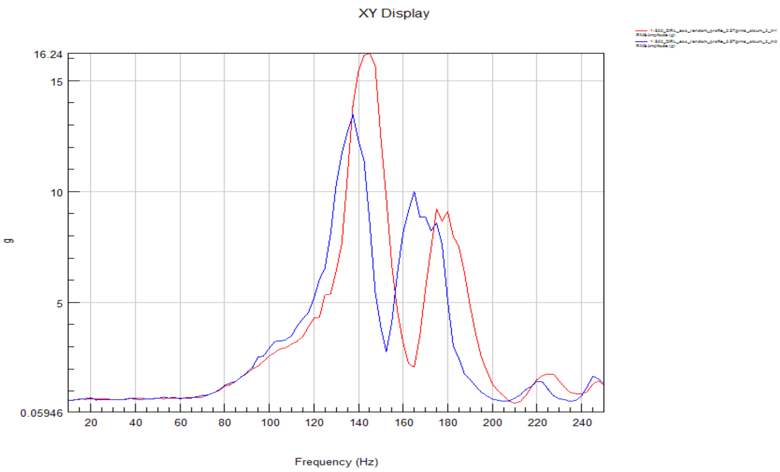


Figure 20. Acceleration Frequency Response Curves before and after crack

When PSD G values and durations until crack start are put in to the formula given in by Lalanne, it is concluded that vibration durability test acceleration formula proposed by LaLanne is applicable for accelerating the vibration durability tests prepared for Automotive Lighting Devices.

$$\frac{G_{Reduced}}{G_{Real}} = \left(\frac{T_{Real}}{T_{Reduced}}\right)^{\frac{2}{b}} \tag{7}$$

$$\frac{0,08}{0.057548} = 1.390141 \quad \left(\frac{7250}{2460}\right)^{\frac{2}{6,64}} = 1.384798 \tag{8}$$

$$\frac{0.08}{0.046705} = 1.712878 \quad \left(\frac{14287}{2460}\right)^{\frac{2}{6,64}} = 1.698722 \tag{9}$$

It should be noted that plastic components mechanical properties change even between different production lots, ambient temperature, molding parameters, etc. In this study, parts from the same production lot have been used and tests are conducted in a short time period. Furthermore, a reliability study can be conducted with high test sample size.

Note: PC-ABS material of the housing part, S-N log-log curve slope b taken as 6.64. If b is not known, it could be calculated by putting the crack start durations and PSD levels in to the test acceleration formula.

## 4. CONCLUSION

In this study it was shown that different type of vibration durability test specifications can be compared by calculation of their FDS's.

It was shown that vibration durability test specifications can be accelerated by the formula proposed by Lalanne. On an updated FEM, random vibration FEA has been performed and critical stress locations were validated by physical tests.

Acceleration response of DRL has been measured by accelerometer and crack start durations have been detected by inspection of acceleration time history data. FFT has been performed on time history acceleration data until crack start and after crack start. By inspection of acceleration frequency response curves, structural changes have been detected

## ABBREVIATION

K=Spring stiffness

b=Slope of S-N curve

C=Basquin coefficient

$n_p^+$=Mean number of maxima per second

T=Duration of vibration

z(t)= Relative response displacement of SDOF mass wrt to base

$z_{rms}$=rms of z(t)

$z_p$=Peak value of z(t)

r=Irregularity factor

Q=Damping Factor

G=PSD G

Γ=Gamma Function

$f_{0=}$Natural Frequency

f=Frequency of excitation

h= Interval ($f/f_0$)

(t)= Absolute acceleration of the base of a single degree-of-freedom system

$\ddot{x}_m$ = Maximum value of (t)

ρ= Logarithmic sweep rate in octaves per minute

$A(f_0)$=Acceleration amplitude in $m/s^2$ at frequency $f_0$ Hz

D=Damage

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