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HEAD LAMP DURABILITY PREDICTION

Coutinho, Ismael A.
CNH Industrial: IVECO, Rod. MG 238 Km 73,5, Sete Lagoas - MG
ismael.coutinho@cnhind.com

Toledo, Marcelo A.
Amsted-Maxion Equipamentos Ferroviários: Área Sítio São João, s/n, Jardim São Camilo, Hortolândia - SP
marcelo.toledo@amstedmaxion.com.br

Phd Landre Jr, Jánes
Pontifícia Universidade Católica de Minas Gerais, Av: Dom José Gaspar, 500, Coração Eucaristíco, Belo Horizonte - MG
janes@pucminas.br

Abstract. *Durability is an essential matter at commercial vehicles project development. Reliability is crucial for brand reputation and to minimize financial losses due to inoperative capital good, which is case the case of a truck. Moreover, the adoption of polymeric material continues to grow all over vehicle's subsystems due to its wide range of mechanical properties, relative reduced cost and form design alternatives allowed by the manufacturing process. However, virtual durability prediction of such materials presents, in general, an additional challenge: lack of durability life curves (SN or εN).*
This work is about virtual life prediction of a polypropylene talc filled Front Head Lamp of a light truck from IVECO. Two vibration tests were performed to assess component's durability: a standardized PSD bench input and the accelerations measured at a particular Iveco rough road mission. Moreover, the acquired accelerations were compared to determine the most severe configuration for the Head Lamp among the two possible configurations of the vehicle: cab or van combined with vehicle load condition: GVW or curb. Modal superposition was used at either analysis to calculate component's dynamic response. The damage was accumulated based on S-N approach.

Keywords: *Fatigue, Modal Superposition, Polypropylene, Large Mass Method (LMM)*

1. INTRODUCTION

Reliability is a crucial quality aspect, especially for capital good such as trucks. Any interruption at normal transportation routine not only results in an extraordinary expense but also in a financial loss induced by an inoperative machine. It is a key aspect of the business.

Virtual analysis plays an important role at product development stage since it can assess component's quality and preview failures or fragile points of the design at the beginning phase of project. The goal is to spend time and money for producing a reliable part, ready to withstand several different application conditions.

A Mission Profile comprises several measured load events that are assumed typical of various real-life situations. A ground vehicle, for example, might include events for highway use, town use, light off-road use, cross-country, fully and part loaded vehicle weight, severe pothole and kerb strikes, as well as the initial damage associated with component packaging and transportation. Figure 1 illustrates a typical Rough Road mission. Representative PSDs or time signals are measured for each event along with an estimate of how long the vehicle might be expected to see this event in-service.

Figure 1. Daily Africa mission

To assure product quality and reliability it is mandatory to define representative validation missions according to vehicle operation. Iveco adopt two mandatory validation tests:

- Rough Road: off road circuit, with wide range of variable obstacles along year, since it is a non-controlled pavement.
- Bench test: random vibration input profile, defined by a Power Spectrum Density (PSD), specific for electrical components attached to vehicle cab.

2. HEAD LAMPS

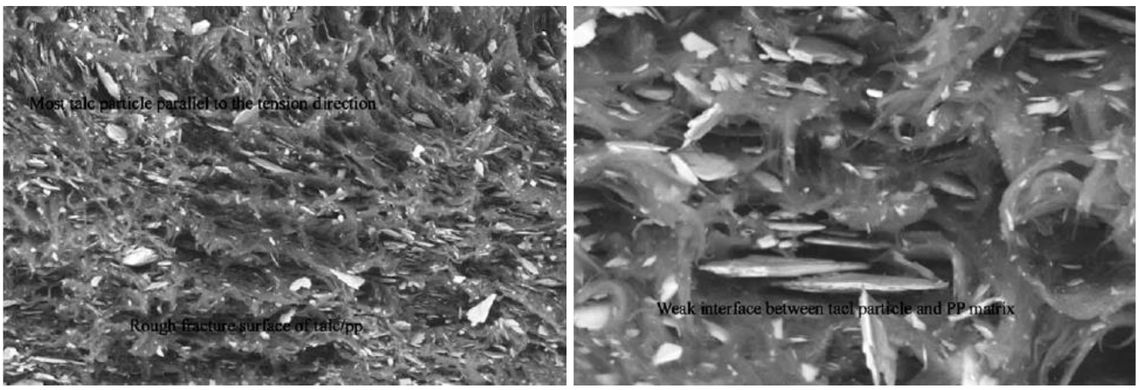
There are two versions of Iveco Light Truck: Cab&Chassis, option for freely implementation, and "Van", closed body shell for goods or passengers transportation. Although each version has its own dynamic characteristics, result from mass distribution, cab stiffness, center of gravity and primary suspension properties, the head lamp is anchored at the same way at both vehicles versions. Figure 2 presents the critical anchorage and the usual reported operation failure.

Figure 2. Head Lamp critical anchorage and most common reported failure

2.1 Polypropylene

The Head Lamp core is made of PP-TD40, polypropylene with 40% talc-filled. (Zhou, Y. *et al.*, 2005) Polypropylene is a semi-crystalline engineering thermoplastic and is known for its balance of strength, modulus and chemical resistance. It have many applications in automobiles, appliances and other commercial products in which creep resistance, stiffness and some toughness are demanded in addition to weight and cost savings. The incorporation of inorganic particulate fillers has been proved to be an effective way of improving the mechanical properties, and in particular the toughness, of polypropylene.

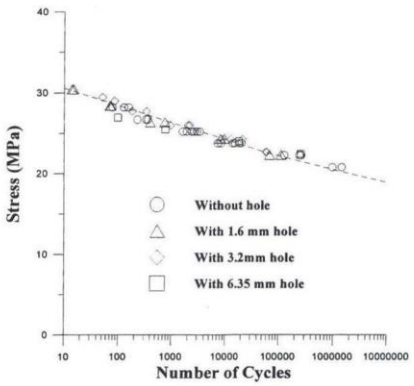
The fracture surface of talc-filled polypropylene shown in Figure 3 is rough, and particles mainly parallel to the flow direction. The damage of specimen is initiated at the location of the highest stress intensity, the particle end. Voids grow from this damage along the particle and coalesce with each other to form crack. No matrix material can be seen adhering to the pulled-out particles (Figure 3 b), indicating it is the weak interface between matrix and particle.



A) B)

Figure 3, Fracture surfaces of talc-filled polypropylene (Zhou, Y. *et al.*, 2005)

Figure 4 presents mechanical properties of head lamp material such as maximum stress, elastic modulus, density, Poisson, as well as one crucial characteristic for virtual durability prediction: S-N curve. Also known as a Wöhler curve, it is a graph of the magnitude of a cyclic stress (S) against the logarithmic scale of cycles resisted before failure (N).



Mechanical Properties	PP-TD40
Tensile Strength [MPa]	50 (min)
Elongation [%]	10
Yield Stress [MPa]	-
Young's Modulus [MPa]	3200 – 3600
Poisson's Ratio	0.28
Density [kg/m^3]	1230-1270

Figure 4. PP-TD40 (40% talc-filled) S-N curve and mechanical properties (Zhou, Y. and Mallick, P. K., 2005)

S-N curves are derived from 10 or more tested samples (Lalanne, 2002) of the material to be characterized subjected to sinusoidal stress of different amplitudes is applied by a testing machine which also counts the number of cycles to failure. The sinusoidal stress amplitude is updated and the cycles counting are reinitiated for other new samples, generating multiple points to characterize material durability (resisted cycles until fracture) to different efforts magnitude (stress amplitude).

Even though the head lamp is anchored by the same 3 points at both truck's version, each one has its own primary suspension and sprung mass stiffness. The first signal treatment task was to determine the most critical condition considering the following variables: vehicle version ("Cab&Chassis" or "Van") summed also with gross weight changes (curb or GVW). An accelerometer was placed near to part's center of gravity and pseudo-damage calculated to compare severity at three axis directions and vehicle condition. Regarding dynamic events, severity is result from three characteristics of the inputted vibration: frequency, amplitude and duration (Bishop, 2014).

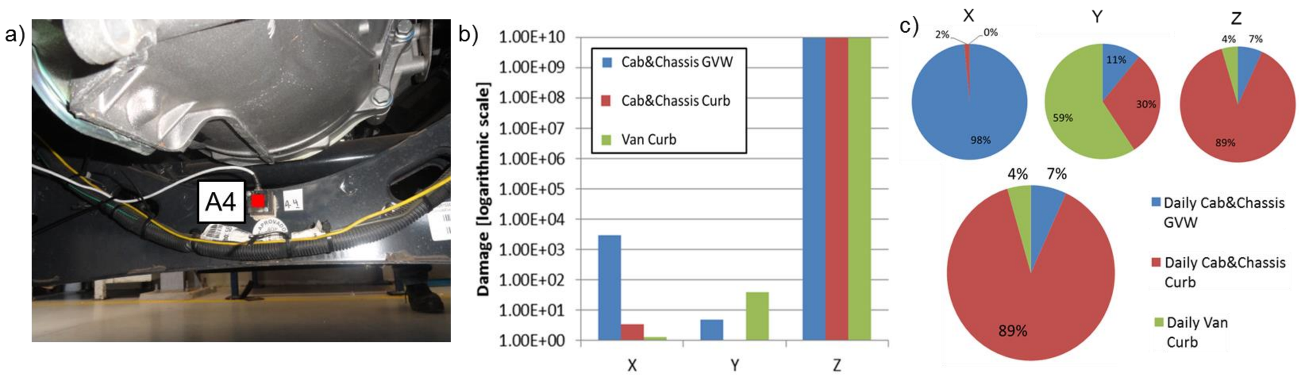


Figure 5. Severity comparison by (a) accelerometer position; (b) vehicle version and loading condition and (c) damage

Figure 5 (a) presents the measurement accelerometer position, at chassis cross member, considered to assess vehicle condition severity. A resultant damage was calculated from the accelerations at each direction. As demonstrated at Figure 04 (b) and (c), even though "Cab&Chassis" at GVW and "Van" at curb had the higher values in X (longitudinal) and Y (lateral) directions respectively, the vertical accelerations, along Z axis, were decisive, and the most critical condition is, in fact, "Cab&Chassis" vehicle version without payload (Curb).

MODAL ANALYSIS: EIGENVALUES, EIGENVECTORS AND PRINCIPAL COORDINATES

Vibrating components are highly susceptible to fatigue failure. Fatigue occurs through long term exposure to time varying loads which although modest in amplitude give rise to microscopic cracks that steadily propagate to failure.

(Thorby, 2008) Eigenvalues and eigenvectors are mathematical concepts, and their use is not confined to vibration theory. The words incidentally are derived from the German word eigen, meaning 'own', so the eigenvalues of a set of equations are its own values, and the eigenvectors are its own vectors. They tend to arise, for example, when the deflection shape of a system depends upon its loading, but the loading, in turn, depends upon the deflection shape. This is obviously true of a vibrating system with more than one degree of freedom, and it will be seen that such a system possesses as many eigenvalues, which are related to its natural frequencies, as there are degrees of freedom, and that each is associated with a shape function known as an eigenvector.

Eigenvalues and eigenvectors can be real or complex, and this is determined entirely by the damping in the system. Systems with no damping have real eigenvalues and eigenvectors. If the damping is of the proportional kind, the eigenvalues are complex, but the eigenvectors remain real. Other than in these special cases, when damping is presented, both: eigenvalues and the eigenvectors become complex and significantly more difficult to deal with. Fortunately, when the damping terms are small, the eigenvalues and the eigenvectors are 'nearly real' (i.e. the imaginary parts are small compared with the real parts), and in practice real eigenvectors can be used as a good approximation for practically all structures, and most systems generally. There are, however, some important exceptions, such as systems with significant gyroscopic effects, such as helicopter rotors, and aircraft flutter or response calculations using equations based on the p method, where aerodynamic damping terms can couple the modes.

Considering a general equation of motion (1) in global coordinates

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \vec{F}(t) \tag{1}$$

It can be transformed into an alternate form in modal coordinates

$$[\boldsymbol{M}]\,\ddot{\vec{q}} + [\boldsymbol{C}]\{\dot{\vec{q}}\} + [\boldsymbol{K}]\vec{q} = \vec{Q}(t) \tag{2}$$

defining a relationship between the global displacements, x, and the modal displacements, q as $[\boldsymbol{X}]$

$$\{x\} = [\boldsymbol{X}]\{q\} \tag{3}$$

As stated by (Thorby, 2008), since eigenvalues and eigenvectors are unaffected by the applied forces, they can be obtained from the 'homogeneous' equations. Assuming there is no damping, and if the system is set in motion at frequency ω, all the elements of the vector $\{q\}$ will vibrate in phase, or exact antiphase, with each other. The responses at all the points q_1, q_2, etc. can be considered as horizontal or vertical projections of a rotating, complex unit vector, $e^{i\omega t}$, multiplied by the real constants $\bar{q}_1$, $\bar{q}_2$, etc. Then the responses of all the points, q_1, q_2, etc., are given by:

$$\{q\} = \begin{Bmatrix} q_1 \\ q_2 \\ \vdots \end{Bmatrix} = \begin{Bmatrix} \bar{q}_1 \\ \bar{q}_2 \\ \vdots \end{Bmatrix} e^{i\omega t} = \{\bar{q}\}e^{i\omega t} \tag{4}$$

The vector of accelerations, $\ddot{q}$, is given by differentiating Eq. (4) twice with respect to time:

$$\{q\} = \begin{Bmatrix} \ddot{q}_1 \\ \ddot{q}_2 \\ \vdots \end{Bmatrix} = -\omega^2 \begin{Bmatrix} q_1 \\ \bar{q}_2 \\ \vdots \end{Bmatrix} e^{i\omega t} = -\omega^2\{\bar{q}\}e^{i\omega t} \tag{5}$$

Substituting Eqs (4) and (5) into 'homogenous' form of Eq. (2) and dividing through by $e^{i\omega t}$ gives

$$(-\omega^2[\boldsymbol{M}] + [\boldsymbol{K}])\{q\} = 0 \;\rightarrow\; (-\lambda[\boldsymbol{M}] + [\boldsymbol{K}])\{q\} = 0 \tag{6}$$

Equation (6) is an eigenvalue problem, although not in standard mathematical form. It can be satisfied by n different values of λ, say λ_1, λ_2, …,λ_n, which are the eigenvalues, and n corresponding vectors, $\{\bar{q}\}$, say $\{\bar{q}\}_1$, $\{\bar{q}\}_2$, …, $\{\bar{q}\}_n$, which are the eigenvectors, where n is the number of degrees of freedom.

When the transformation matrix, $[\boldsymbol{X}]$, in Equation (3) instead of simply being based on assumed modes, is formed by writing its columns as the eigenvectors, normalized by mass, from Eq. (6), something remarkable happens: both the mass and the stiffness matrices of the modal equations then become diagonal, and the system is transformed into a set of completely independent, single-DOF equations. Furthermore, assuming a proportional damping, the damping matrix will also be diagonal.

This transformation into 'normal', or 'principal', coordinates is so useful that it is used in the analysis of almost all multi-DOF systems, enabling the usage of the superposition principle to describe system behavior. It is possible to find the eigenvalues and eigenvectors of a system explicitly only up to two degrees of freedom, (Thorby, 2008). Above that number, the process must be iterative. That's where Lanczos algorithm demonstrates its importance.

LARGE MASS METHOD (LMM)

A common approach to represent base motion acceleration or displacement at virtual models is to consider it as a lumped mass. According to model's purpose it could represent, for example, the vibration bench test anchorages and for a vehicle in motion, the ground imposing vibrations due to its irregularities. As stated by (Brandt, 2010), for such systems, the motion of each mass is a DOF, i.e., there are as many DOF as masses. For either cases: to represent a vibration bench supports or the anchorages of the head lamp at, for example, a truck's cab, it is crucial to make the lumped mass heavy *"enough"*, around $1x10^6$ times greater than the focus excited component, so that it can be considered an inertial reference, cable to withstand the reactions produced by the anchored part and to impose the specified displacement profile. The only limit for the size of the mass is numeric overflow in the computer. Throughout second Newton's law and once the lumped mass is known, it is trivial to calculate the forces, necessary to use at equation (2) to calculate the modal participation coefficients for each mode shape and then proceed to modal superposition to determine the stress history.

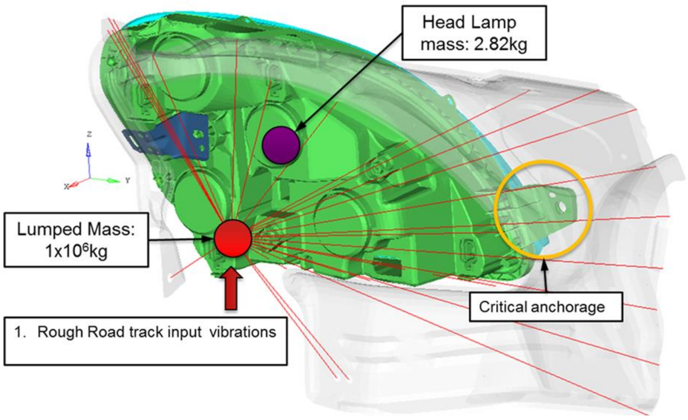


Figure 06. Lumped mass virtual model: Head Lamp

MODAL SUPERPOSITION

(Rao, 2011) Periodic forces of any general waveform can be represented by Fourier series as a superposition of harmonic components of various frequencies. The response of a linear system is then found by superposing the harmonic response to each of the exciting forces.

The key advantage of modal coordinates is that each loading input can be split up into the contribution factors associated with each mode shape for the structure. A CAE program, such as Optistruct, adopted at this work, calculates both modal stresses and modal participation factors for each natural frequency. The linear modal superposition is then calculated according Eq. (7) to find the stress time history, from where the fatigue calculation will start.

$$\sigma(t) = \sum_{i=1}^{n} \sigma_i \cdot q_i(t) \tag{7}$$

Where: $\sigma(t)$=stress time history
σ_i=modal stress for each mode i
$q_i(t)$= modal/principal coordinate for each mode i

RESIDUAL VECTORS

The accuracy of the modal method can be vastly improved by adding the displacement vectors of a static analysis based on the dynamic loading to the matrix of eigenvectors. These vectors are frequently referred to as residual vectors. (Erwins,1995) Assignment of this correction equalizes influence of structure modes outside the frequency measurement range and/or analysis, limited due to practical reasons.

Optistruct generates residual vector based on unit static loads applied at the dynamic degrees of freedom. That is, the static loads for the residuals vectors generation are unit vectors at the degrees of freedom where the dynamic load is applied. The number of residual vectors is equal to the number of loaded degrees of freedom. This method is called unit load and is generally more accurate method.

In the case of excited displacements, the residual vectors are obtained by solving static load cases with unit displacements at the same degrees of freedom as the dynamic excited displacement degrees of freedom.

Aligned with (Erwin, 1995), (Alvarez and Benito, 1992) equation the residual vector (8) to add to the truncated (m) modal basis the effect of neglected modes:

$$\phi_R = \sum_{i=m+1}^{n} q_i(t)\psi_i \tag{8}$$

This formula, however, is not valid for practical use, as it based upon the modes whose computation is intended to avoid. Instead of (8), the following expression (9) is preferable:

$$\phi_R = \chi - \sum_{i=1}^{m} q_i(t)\psi_i \tag{9}$$

X is a vector whose components are the displacements due to an unit load applied at each dynamic degree of freedom.

[Alvarez and Benito] demonstrate equation (10) based on the contribution of all structure modes:

$$\chi = \sum_{i=1}^{n} q_i(t)\psi_i \quad \rightarrow \quad \chi = \sum_{i=1}^{m} q_i(t)\psi_i + \sum_{i=m}^{n} q_i(t)\psi_i \quad \rightarrow \quad \chi = \sum_{i=1}^{m} q_i(t)\psi_i + \phi_R \tag{10}$$

Therefore, to calculate the residual vectors it is only needed to subtract from χ vector each computed eigenvector multiplied by its participation factor.

DESIGN ENHANCEMENT

After the development of a reliable mathematical model, an important advantage is the possibility to make changes and quantitative measure its contribution for part's durability life. Figure 7 presents the proposed modifications at the original head lamp.

1. Increase ribs thickness
2. Increase ribs height
3. Fillet radius

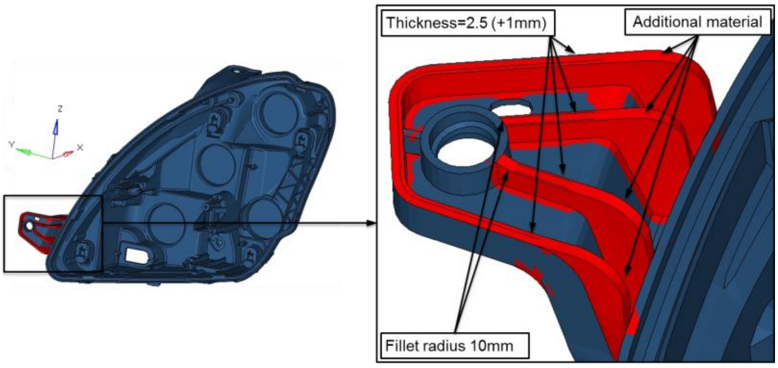


Figure 7. Proposed modifications at original head lamp

FATIGUE AND DAMAGE

Fatigue failures occur due to the application of fluctuating stresses that are much lower than the stress required to cause failure during a single application of stress. It has been estimated that fatigue contributes to approximately 90% of all mechanical service failures (ASM, 2008).

Fatigue damage is the modification of material characteristics, primarily due to the formation of cracks and resulting from the repeated application of stress cycles. This change can lead to a failure. (Lalanne, 2002) Fatigue starts with a plastic deformation, first very much localized, near high stress concentrations, such as persistent slip bands (PSBs), inclusions, porosity, or discontinuities. The effect is extremely weak and negligible for only one cycle. If the stress is repeated, each cycle creates a new localized plasticity. After a number of variable cycles, depending on the level of stress, ultra-microscopic cracks can be formed in the newly plastic area. The plastic deformation extends then from the ends of the crack which increase until becoming visible with the naked eye and lead to failure of the part. Fatigue damage is a cumulative phenomenon.

Cumulative damage theories consider the fatigue process to be one of damage accumulation until the life of the component is exhausted.

As showed at Figure 3, talc filled polypropylene has far more slip bands than so called PSBs of ordinary isotropic materials. Even polished metallic surfaces present preferable shear planes. By the way, it is exactly why they are called persistent slip bands. Slip bands are a result of the systematic buildup of fine slip movements on the order of only 1 nm. However, the plastic strain within the PSB can be as much as 100 times greater than that in the surrounding material. The back-and-forth movement of the slip bands leads to the formation of intrusions and extrusions at the surface, eventually leading to the formation of a crack. The initial crack propagates parallel to the slip bands. At the beginning phase, no structural resistance is compromised. However, as the crack grows, the complete failure of the component is imminent, unless that cycle loading is stopped.

Damage for each cycle is defined as $1/N_f$ where N_f is the calculated number of cycles to failure at that stress level, taking into account all effects including mean stress, surface condition and certainty of survival. Linear damage accumulation is considered, id est, the total damage D for an event or duty cycles is the sum of individual damage values of each cycle in the event or duty cycle, as if they happened separately.

Palmgren-Miner's hypothesis states that fatigue failure occurs when the accumulated damage exceeds one, $D(t) > 1$. Fatigue is usually a complicated mixture of several different mechanisms, and the assumption of linear damage accumulation inherent in Palmgren-Miner's law should be viewed skeptically. Cyclic loads induce strengthening mechanisms such as molecular orientation or crack blunting, the rate of damage accumulation could drop during some part of the material's lifetime. Miner's law ignores such effects, and often fails to capture the essential physics of the fatigue process. For tests with random loading histories at several stress levels, correlation with the Palmgren-Miner rule is generally very good.

DURABILITY MISSION: ROUGH ROAD

Iveco's rough road durability mission is composed by random obstacles, but it is not possible to consider it as stationary signal looking at 10km acquisition performed for this study. Therefore, a damage equivalence study is mandatory to represent this mission profile as a PSD, such as previewed at common standards for electrical components. Figure 09 presents the typical acceleration profile and the correlation with FEM results.

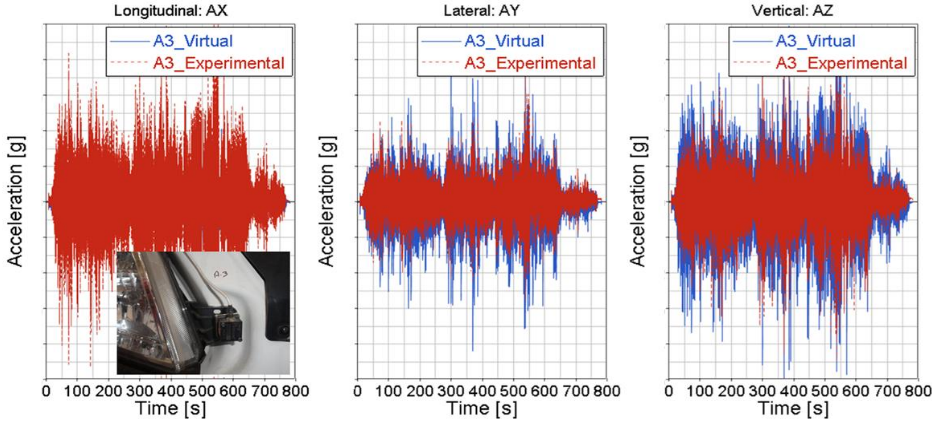


Figure 9. Correlation between virtual x experimental accelerations at Rough Road mission

The calculated damage was extrapolated to compose total mission life. As demonstrated by Figure 10, it was possible to identify critical points, similar to failures reported at the field test and with equivalent life level.

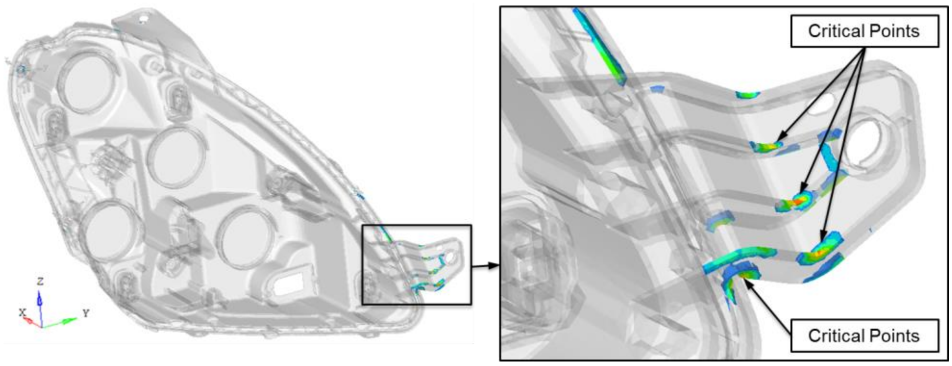


Figure 10. Rough Road mission life prediction, original part.

Once again the life at the enhanced parts was evaluated. The result is presented at Figure 11.

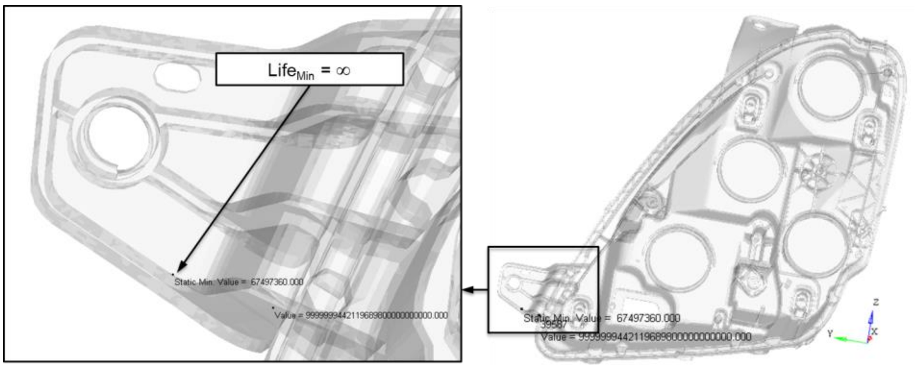


Figure 11. Rough Road mission life prediction, enhanced part.

CONCLUSIONS

Notably for the head lamp, scope of this study, Palmgren-Miner's linear damage accumulation law was able to satisfactory correlate with reported field failure life, even with its heterogeneous, strain rate sensitive material: talc filled polypropylene. The virtual predicted life result was within statistical deviation from field failure reported history.

Finite element model combined with modal superposition principle well described the dynamic empirical behavior. However, key cares must be highlighted:

- Modal calibration and mode shapes inspections for inconsistencies. Only the 10 first mode shapes were considered. Above approximately 50Hz, linearity principle was considered not valid and the mode shapes effect only considered as contribution from residual vectors;
- Reliable data acquisition and signal treatment. Especially for principal coordinates calculation, no drift, steps or discontinuity must exist at the data. Otherwise unreal spectrum will compromise the durability result.
- Consistent S-N curve, even though suppliers do not generally spend resources at this particular essay, it is decisive for converting stress into durability life.
- Appropriate finish factor definition. Manufacturing process is decisive for component's durability. Notch geometry factor concentrate stress and reduce dramatically part's life.

The premature detection of failure points can avoid spending crucial amount of time and money. Matching of virtual and experimental (data acquisition) tools greatly support development process, assuring reliable prototypes and minimizing failures at certification phase.

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