PROBABILISTIC STRUCTURAL ANALYSIS APPLIED TO SPRING LEAF SUSPENSION ASSEMBLY OF SEMI-TRAILER TANK VEHICLE

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Abstract*. The present paper proposes and presents a structural analysis procedure for suspension parts of a semi-trailer tank vehicle by applying FEM (Finite Element Method) programs and PSD (Power Spectral Density) analysis in frequency domain. The three main concepts adopted are: the modeling of all suspension's components, the simplification of the tank's modeling and the probabilistic approach. This procedure leads to fatigue prediction of suspension's components. Furthermore, the results allow concluding that the road condition is much more relevant to the reduction of the components' lifetime than the vehicle's velocity.*

Keywords*. Suspension, truck, PSD, FEM, fatigue*

1. Introduction

The increase of computer processing capability, associated to new and more efficient mathematical methods - introduced into programs of structural analysis based on Finite Element Method (FEM) - allows putting into practice complex structural analyses, faster and efficiently. Besides, deterministic design procedure is been substituted by a stochastic approach, especially in situations where evidentially the loads are random. The proposed procedure for lifetime prediction of semi-trailer tank FEM modeling is based on three main concepts:

- Instead of modeling suspension parts independently, after determining the loads acting in each part, it is suggested modeling the whole suspension assembly respecting the correct linkage between the parts. This approach permits taking into account more effectively the parts interactions and flexibilities. Nonetheless, it increases significantly the model degrees of freedom and represents a huge obstacle to the analysis viability.
- In order to minimize this problem for the suspension assembly modeling, once the aim it is not to obtain stress values in the whole structure but in the suspension parts only, a consistent simplification of tank modeling allows reducing the degrees of freedom. This simplification consists in replacing the shell structure of the tank by a beam structure with equivalent properties – mass, moment of inertia, polar moment of inertia, centroid position – similarly to the hull girder approach in ship structures' design (Hughes, 1983).
- Finally, since displacements imposed on the tires by soil irregularities are random, stochastic treatment is justified. Therefore, it is intended to apply the PSD of the wheels displacement as the principle of a stochastic approach. The use of PSD analysis depends on some modeling criteria that will be described further.

Hence, applying the described concepts to a suspension assembly of a semi-trailer tank vehicle, this paper describes and discusses the suspension assembly modeling, the validation of the hull girder simplification by comparison with the full tank results as well as the PSD FEM process calculation.

2. Brief Theory Review

Some theoretical concepts are necessary to guaranty the paper's comprehension and, therefore, must be at least presented. The main concepts are: PSD analysis and Dirlik's method for determining the Probability Density Function (pdf) of stress Rainflow cycles.

2.1. PSD Analysis

A PSD analysis consumes less CPU time processing when it is performed in the frequency domain. Hence, the first step of this approach, that also applies mode superposition, is determining the modes of vibration and natural circular frequencies of the structure that correspond, respectively, to the eigenvectors and eigenvalues obtained by solving the structure's eigenproblem. Equation (1) presents the eigenproblem.

$$[K]\{\Phi\}=\lambda[M]\{\Phi\} \tag{1}$$

where $[K]$ is the stiffness matrix, $[M]$ is the mass matrix, $\{\Phi\}$ is an eigenvector of mode shapes and λ is an eigenvalue equal to squared natural circular frequency. The description of methods to solve this eigenproblem is available at Bathe (1996).

In the specific situation of the semi-trailer tank, there are no nodal force excitations but only nodal displacement excitations and, consequently, the mode superposition theory will be presented for the base excitation case.

2.1.1.Mode Superposition

The mode superposition is a linear operation that is based on summing the contribution of each mode of vibration to obtain the structure's dynamics response. Equation (2) presents the equation of equilibrium governing the linear dynamic response of a system.

$$[M]\{\ddot{Q}\}+[C]\{\dot{Q}\}+[K]\{Q\}=\{P\} \tag{2}$$

being $\{Q\}$, $\{\dot{Q}\}$ and $\{\ddot{Q}\}$ the displacement, velocity and acceleration vectors of the finite element nodes; $\{P\}$ is the vector of applied loads; and $[C]$ is the damping matrix. All matrixes and vectors in Eq. (2) can be reorganized in order to segregate free Degrees Of Freedom (DOF), $\{Q_f\}$, and restrained DOF, $\{Q_r\}$, as:

$$\begin{bmatrix}[M_{ff}] & [M_{fr}]\\ [M_{rf}] & [M_{rr}]\end{bmatrix}\begin{Bmatrix}\{\ddot{Q}_f\}\\ \{\ddot{Q}_r\}\end{Bmatrix}+\begin{bmatrix}[C_{ff}] & [C_{fr}]\\ [C_{rf}] & [C_{rr}]\end{bmatrix}\begin{Bmatrix}\{\dot{Q}_f\}\\ \{\dot{Q}_r\}\end{Bmatrix}+\begin{bmatrix}[K_{ff}] & [K_{fr}]\\ [K_{rf}] & [K_{rr}]\end{bmatrix}\begin{Bmatrix}\{Q_f\}\\ \{Q_r\}\end{Bmatrix}=\begin{Bmatrix}\{0\}\\ \{P_r\}\end{Bmatrix} \tag{3}$$

In Eq. (3), $\{P_r\}$ is the unknown force vector at restrained DOF due to the imposed displacements; and the vector null appears because it is assumed that there are no nodal forces acting at free nodes. From Eq. (3) and assuming light dumping, we have:

$$[M_{ff}]\{\ddot{Q}_f\}+[C_{ff}]\{\dot{Q}_f\}+[K_{ff}]\{Q_f\}=\{P_{eff}\}=-[M_{fr}]\{\ddot{Q}_r\}-[K_{fr}]\{Q_r\} \tag{4}$$

where $\{P_{eff}\}$ is the vector representing effective force due to the imposed displacements; $\lfloor K_{ff}\rfloor$ and $\lfloor M_{ff}\rfloor$ are used in Eq. (1) to obtain mode shapes and natural circular frequencies. By Rayleight approach, we may write:

$$\{Q\}=\sum_{n=1}^{N}\{\Phi_n\}\cdot y_n \tag{5}$$

being $\{\Phi_n\}=\{\phi_{1n},\phi_{2n},\ldots,\phi_{jn},\ldots,\phi_{XN}\}^T$ the eigenvector corresponding to the nth-mode of vibration; X is the total number of free DOF; N is the total number of calculated circular natural frequencies; and $y_n(t)$ is the nth term of the generalized displacements. Equation (6) can be written for each term of $\{Q_{ff}\}$:

$$q_j=\sum_{n=1}^{N}\phi_{jn}\cdot y_n \tag{6}$$

where $j=1,2,\ldots,X$; and y_n is obtained from the second order differential equation equivalent a single degree of freedom:

$$m_n\ddot{y}_n+c_n\dot{y}_n+k_n y_n=L_n \tag{7}$$

In Eq. (7), $n=1,2,\ldots,N$; $m_n=\{\Phi_n\}^T[M]\{\Phi_n\}$ and $c_n=\{\Phi_n\}^T[C]\{\Phi_n\}$ are, respectively, the generalized mass and damping for nth-mode of vibration. The generalized stiffness k_n is given by:

$$k_n=m_n\omega_n^2 \tag{8}$$

In Eq. (8), $\omega_n=\sqrt{\lambda_n}$ is the natural circular frequency of the nth-mode shape; and λ_n is the nth eigenvalue solution of Eq. (1). Finally, the generalized loads L_n for nth-mode shape is:

$$L_n=\{\Phi_n\}^T\cdot\{P_{eff}\}=\sum_{l=1}^{X}\phi_{\ln}p_l \tag{9}$$

being $\{P_{eff}\}=\{P_1,P_2,\ldots,P_X\}^T$ and $\{\Phi_n\}=\{\phi_{1n},\phi_{2n},\ldots,\phi_{\ln},\ldots,\phi_{XN}\}^T$.

2.1.2. Power Spectral Density

It is important to note that the PSD is the Fourrier transform of the autocorrelation function (Newland, 1984; Meirovitch, 1986) as shown by the following equation:

$$S_{q_j}(\omega)=F\left[R_{q_j}(t)\right]=F\left[\int_{-\infty}^{+\infty}q_j(t+\tau)q_j(\tau)d\tau\right]=F\left[\int_{-\infty}^{+\infty}\sum_{n=1}^{N}\phi_{jn}y_n(t+\tau)\sum_{m=1}^{N}\phi_{jm}y_m(\tau)d\tau\right]=\sum_{n=1}^{N}\sum_{m=1}^{N}\phi_{jn}\phi_{jm}F\left[\int_{-\infty}^{+\infty}y_n(t+\tau)y_m(\tau)d\tau\right] \tag{10}$$

where $S_{q_j}(\omega)$ is the PSD of the jth term of the displacement vector $\{Q\}$. The last term in Eq. (10) can be reached by using Eq. (6) and knowing that $\{\Phi_n\}$ is not time dependent. By applying PSD definition, we have:

$$S_{q_j}(\omega) = \sum_{n=1}^{N}\sum_{m=1}^{N} \phi_{jn}\phi_{jm} S_{ynm}(\omega) \tag{11}$$

The spectrum of generalized displacements $S_{ynm}(\omega)$ - output - is related to the spectrum of generalized loads – effective loads, input - as follows:

$$S_{ynm}(\omega) = H_n^*(\omega) H_m(\omega) S_{Lnm}(\omega) \tag{12}$$

being $H_m(\omega)$ the transfer function presented in Eq. (13); $H_n^*(\omega)$ is the complex conjugate of $H_n(\omega)$, with:

$$H_n(\omega) = \left(k_n\left((1-\beta_n^2) + i(2\zeta_n\beta_n)\right)\right)^{-1} \tag{13}$$

where ζ_n is the damping ratio; β_n is the ratio of the frequency of the externally applied excitation to the natural frequency; and $i^2 = -1$. The spectrum of generalized loads $S_{Lnm}(\omega)$ is function of the spectrum of effective loads:

$$S_{Lnm}(\omega) = F\left[\int_{-\infty}^{+\infty} L_n(t+\tau) L_m(\tau) d\tau\right] = F\left[\int_{-\infty}^{+\infty}\sum_{h=1}^{X}\phi_{hn}P_n(t+\tau)\sum_{k=1}^{X}\phi_{km}P_m(\tau)d\tau\right] = \sum_{h=1}^{X}\sum_{k=1}^{X}\phi_{hn}\phi_{km}S_{pnm}(\omega) \tag{14}$$

Equation (14) is obtained by using Eq. (9) and the fact that $\{\Phi_n\}$ is not time dependent. Since the effective load depends on the applied acceleration and displacement – Eq. (4) - and its PSD's are correlated as follows:

$$S_{\ddot{Q}r}(\omega) = \omega^4 S_{Qr}(\omega) \tag{15}$$

it is possible to obtain the displacement PSD of a DOF from the acceleration PSD of the applied displacements. Furthermore, having the PSD of nodal displacements implies in being able to calculate the nodal stress PSD because in FEM analysis the stress results are directly obtained from displacement results.

2.1.3. Wave Propagation

When applying more then one PSD simultaneously, the use of Cross-Spectral Density function (CSD) allows inputting the phase difference between excitations. There are many different types of CSD and their choice depends on the studied phenomenon. Once the model of semi-trailer tank, used in this paper, takes into account symmetry simplification – even thought structure's torsion effects will be lost - all wheels are submitted to the same path and, therefore, only the time lag due to the vehicle's velocity have to be applied (Cebon, 1999). Equation (16) shows expression that defines the time lag CSD:

$$S_{jk}(\omega) = S_{jj}(\omega) e^{-i\omega T_{jk}} \tag{16}$$

where $S_{jj}(\omega)$ is the PSD input; $T_{jk} = d_{jk}/V$ is the time lag; d_{jk} is the distance between excitations j and k; V is the vehicle's velocity; and $i^2 = -1$. From the definition of the time lag, we can affirm that $T_{kj} = -T_{jk}$ and, consequently, $S_{kj}(\omega) = S_{jk}^*(\omega)$. Hereby, we can write the full matrix of spectrum excitations including PSD's and CSD's:

$$[S(\omega)] = \begin{bmatrix} S_{11}(\omega) & S_{12}(\omega) & \dots & S_{1X}(\omega) \\ S_{21}(\omega) & S_{22}(\omega) & \dots & S_{2X}(\omega) \\ \vdots & \vdots & \ddots & \vdots \\ S_{X1}(\omega) & S_{X2}(\omega) & \dots & S_{XX}(\omega) \end{bmatrix} = \begin{bmatrix} S_{11}(\omega) & S_{11}(\omega)e^{-i\omega T_{12}} & \dots & S_{11}(\omega)e^{-i\omega T_{1X}} \\ S_{22}(\omega)e^{-i\omega T_{21}} & S_{22}(\omega) & \dots & S_{22}(\omega)e^{-i\omega T_{2X}} \\ \vdots & \vdots & \ddots & \vdots \\ S_{XX}(\omega)e^{-i\omega T_{X1}} & S_{XX}(\omega)e^{-i\omega T_{X2}} & \dots & S_{XX}(\omega) \end{bmatrix} \tag{17}$$

It is important to notice that, in the case of the semi-trailer tank, there are four positions of excitations – three wheels under the tank and one in the tractor – and all wheels are submitted to the same PSD.

2.2. Dirlik's Method

The fatigue prediction can be done by using the stress PSD, which is an output of the PSD analysis, to determine the probability density function (pdf) of stress Rainflow cycles. This step can be performed by a fatigue modeler proposed by Dirlik in 1985, which is widely applicable (Halfpenny, 1999; Faber et all 1999; Bishop, 1999). Dirlik proposed an empirical closed form for pdf following extensive computer simulation using the Monte Carlo technique (Halfpenny, 1999; Bishop, 1999), as follows:

$$p(S) = \frac{\frac{D_1}{Q} e^{\frac{-Z}{Q}} + \frac{D_2 Z}{R^2} e^{\frac{-Z^2}{2R^2}} + D_3 Z e^{\frac{-Z^2}{2}}}{2\sqrt{M_0}} \tag{18}$$

where variables are written in function of stress PSD output moments:

$$M_n = \int_0^{+\infty} \omega^n S(\omega) d\omega \tag{19}$$

the spectral irregularity factor is given by:

$$\gamma = \frac{M_2}{\sqrt{M_0 M_4}} \tag{20}$$

the expected average frequency of the stress response is:

$$x_m = \frac{M_1}{M_0}\sqrt{\frac{M_2}{M_4}} \tag{21}$$

the other variables are defined as follows:

$$D_1 = \frac{2\left(x_m - \gamma^2\right)}{1+\gamma^2} \qquad D_2 = \frac{1-\gamma-D_1+D_1^2}{1-R} \qquad D_3 = 1 - D_1 - D_2$$

$$Q = \frac{1.25\left(\gamma - D_3 - D_2 R\right)}{D_1} \qquad R = \frac{\gamma - x_m - D_1^2}{1-\gamma-D_1+D_1^2} \qquad Z = \frac{S}{2\sqrt{M_0}}$$

2.3. S-N Relationship

Once the pdf - Eq. (18) - is calculated, the lifetime prediction can be found. Nonetheless, the S-N curve of the material must be defined and adjusted for the influence of equivalent static mean stress value. In the present paper, the S-N curve approach adopted is the elementary rule and the correction rule is Goodman's modification (Fuchs et all, 1980). Equation (22) presents the number of cycles of lifetime in function of stress amplitudes:

$$N(\sigma_c) = N_f^{\beta} \tag{22}$$

where $\beta = \dfrac{\log\left(\sigma_c / \alpha\right) - \log(\sigma_u)}{\log(\sigma_f) - \log(\sigma_u)}$

and, from Goodman's correction rule:

$$\alpha = 1 - \frac{\sigma_m}{\sigma_u} \tag{23}$$

being σ_m the equivalent static mean stress value; σ_c is the stress amplitude; σ_u is the material's ultimate strength stress; σ_f is the material's fatigue strength; and N_f is the lifetime for fatigue strength.

The damage generated by known pdf, applying equivalent static mean stress correction, is given by:

$$\frac{D}{T} = \frac{1}{T_0}\int_0^{+\infty} \frac{p(\sigma_c)}{N(\sigma_c)} d\sigma_c \tag{24}$$

with D being total damage – fatigue failure occurs when $D=1$; T is the total duration time of the excitation; and T_0 is the fundamental period defined as:

$$T_0 = \sqrt{\frac{M_2}{M_4}} \tag{25}$$

It have to be made clear that the Von Mises stress values, obtained by the FEM analysis, are used as input of the equations that depends on stress results. Consequently, even compressive stress values provoke fatigue damage what is a conservative assumption.

2.4. Formulation for Road Roughness

There is an ISO formulation for Road Roughness that dates from 1972. Once its expression is not mathematically useful, the expression presented in Eq. (26) is an approximation of the ISO formulation for the PSD, $S_u(\kappa)$, of the road height u (Cebon, 1999);

$$S_u(\kappa) = \frac{S_0}{(2\pi\kappa + 0.1)^3 (2\pi\kappa + 1)^{-0.75}} \tag{26}$$

where κ is the wavenumber in cycles/m that is related with the circular frequency as shown by eq. (27). The constant, S_0, depends on the road class and the classification is available in Tab. (1).

$$\omega = 2\pi V \kappa \tag{27}$$

Table 1 - S_0 Values in Function of the Road Class

Road Class	Very Good	Good	Average	Poor	Very Poor
S_0 (1E-6m³/cycle)	2 up to 8	8 up to 32	32 up to 128	128 up to 512	512 up to 2048

Since, for the FEM PSD analysis, the PSD must be a function of the circular frequency and - in this case – it is the vertical acceleration's PSD, Eq. (28) presents the relation between the used PSD and $S_u(\kappa)$ (Cebon, 1999; Newland, 1984).

$$S_a(\omega) = \frac{\omega^4}{2\pi V} S_u(\kappa) \tag{28}$$

Each excitation positions are submitted to the PSD defined in Eq. (28). Hence, using Eq. (17) is possible to construct the full matrix of spectrum excitations. The vertical acceleration PSD defines the effective force PSD – see Eq. (4) and Eq. (15) – and, thus, the response PSD can be calculated as shown above using Eq. (11), Eq. (12) and Eq. (14). These equations, and the fact that the road class depends on changing the constant S_0, allow one to conclude that it is not necessary to make analyses using all different road classes. By making the analysis for one road class, it is possible to obtain the results for the others upon multiplying the calculated stress PSD moments by the ratio of the constant S_0 of the new road class results to be obtained to the constant S_0 of the road class used in the FEM PSD analysis.

3. Truck Modeling

The first concept of the present approach is to model the whole suspension assembly respecting the correct linkage between the parts. This was established by coupling the adequate degrees of freedom of linked parts. Figure (1) illustrates the suspension modeling that was used for the Full Tank approach and for Tank Hull Girder approach. Also important is that the adopted model takes into account symmetry simplification although this approach provokes the lost of torsion results.

The second concept is the reduction of the model's number of elements by substituting the tank's shell modeling for the tank's beam modeling. This is possible because we are not interested in obtaining stress values in the tank but in the suspension's components. Figure (1) illustrates both modeling. It must be emphasized that the beam modeling properties – mass, moment of inertia, polar moment of inertia, centroid position – are equivalent of the shell modeling properties. However, in the case of the shell modeling, the weight of the liquid is computed by the hydrostatic pressure effects.

FEM models are not created at the static equilibrium state. Besides, before reaching equilibrium state – only under gravity load - it is observed great displacements of the whole tank especially of spring leaves. Hereby, the PSD analysis, which assumes a linear behavior of the structure, can be performed only after a non-linear analysis that leads the model to the static equilibrium state. In order to evaluate the consistence of the Hull Girder approach, Fig. (1) presents the stress distribution in the suspension components for both approaches after reaching the static equilibrium due the gravity effects with geometric non-linearity. The results presented in Fig. (1) allow affirming that the Hull Girder approach is a suitable and consistent approach to reduce the model's number of elements.

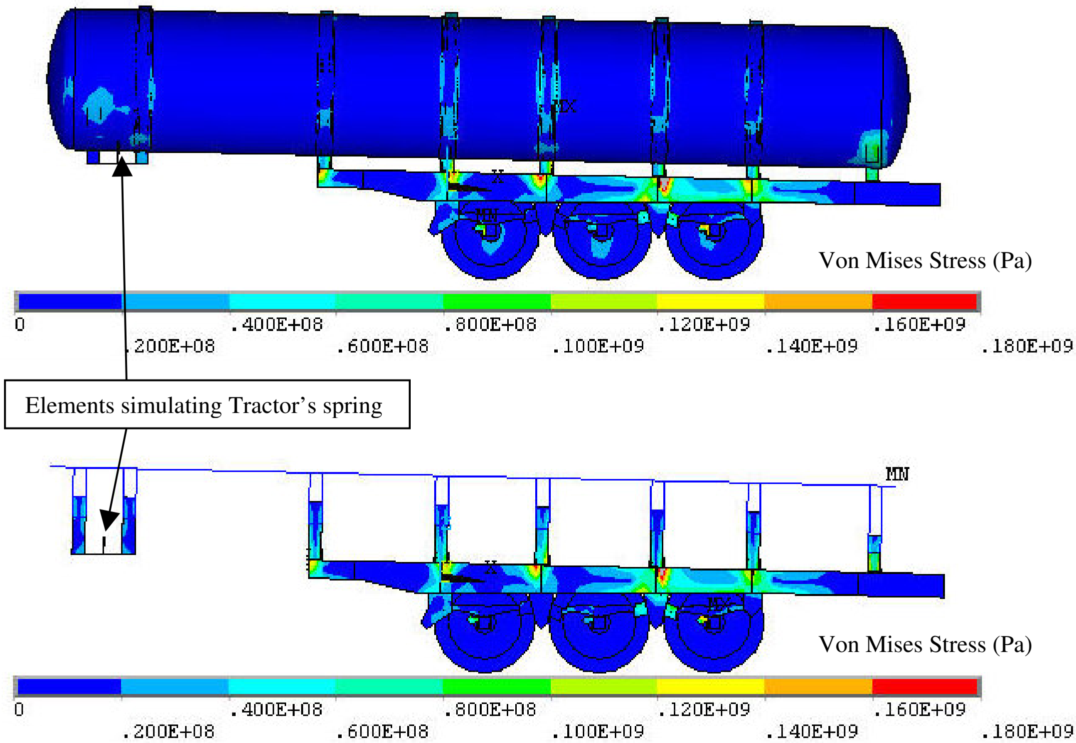

Figure 1. Von Mises Stress Distribution for Both Modeling Approaches for Static Equilibrium after Nonlinear Analysis.

PSD is a more accurate representation of input excitation than an equivalent time history (Bishop, 1999). Furthermore, the frequency domain PSD analysis is more indicated than the time domain PSD analysis because it consumes less CPU time. So that, modal analysis must be performed necessarily and it is absolutely important to notice that for modal analysis the hull girder approach is a better choice than shell modeling. This is due to the fact that in the first case the inertial effects of the liquid are taken into account by the mass associated to the beam elements and, on the order hand, the hydrostatic pressure applied in the shell modeling do not consider inertial effects what leads to incorrect calculation of natural circular frequencies of the whole structure.

4. Analysis Results

The acceleration PSD's representing the road roughness defined by Eq. (28) were applied on the wheels and the extremity of elements simulating tractor's spring – see Fig. (1). The path is the same for all excitation points and there is only the time lag between them, which represents the wave propagation effect, the CSD is given by Eq. (16) (Cebon, 1999). One road class was tested for four different vehicle's velocities – 25, 50, 75 and 100 km/h – and the results of stress PSD moments for the other road classes were obtained from the one calculated what saved CPU time consuming. The probability of encountering each road class was defined by applying the results of the Survey 2002 of the Brazilian roads condition carried out by The Brazilian Confederation of Transportation (Confederação Brasileira dos Transportes, CNT). On the order hand, the probabilities of traveling at each different velocity were arbitrary defined by the paper's authors. This step deserves more adequate approach in future studies. The definition of the occurrence probability of each load condition is absolutely essential for applying the lifetime-weighted method (Hughes, 1983). This approach is based on summing the damage of each load condition multiplied by its occurrence probability. Table (2) summarizes the results of calculating the lifetime prediction by using Dirlik's method for only one hotspot in the longitudinal stiffener, with mean stress $\sigma_m = 154MPa$.

In order to evaluate which parameter – velocity or road class – affects more the components lifetime, let us observe that for a given road class, the ratio of lifetime at lower velocity to lifetime at higher velocity is approximately 7E3. Besides, for a given velocity, the ratio of lifetime at best road class to lifetime at worst road class is approximately

5E17. Therefore, it is possible to state that the road class influences more the components lifetime than the vehicle's velocity. This statement can be graphically verified by watching Fig. (2).

Table 2. Lifetime Prediction for One Hotspot Node on the Longitudinal Stiffener ($\sigma_m = 154 MPa$)

S_0 (m³/cycle)	V (km/h)	Spectrum Moments M_0 (Pa^2)	M_1 $\left(\frac{Pa^2}{s}\right)$	M_2 $\left(\frac{Pa^2}{s^2}\right)$	M_4 $\left(\frac{Pa^2}{s^4}\right)$	Lifetime (s)	Damage (1/s)	Velocity Probab.	Path Probab.	Occurrence Probability	Load Case Damage (1/s)
4E-6	25	4,19E+11	1,41E+12	1,24E+13	3,00E+16	3,22E+27	3,10E-28	0,3	0,395	0,1185	3,68E-29
	50	8,07E+11	4,16E+12	7,92E+13	3,69E+17	2,13E+25	4,70E-26	0,4	0,395	0,158	7,43E-27
	75	1,10E+12	6,67E+12	1,58E+14	9,82E+17	1,98E+24	5,05E-25	0,2	0,395	0,079	3,99E-26
	100	1,32E+12	8,95E+12	2,31E+14	1,51E+18	4,62E+23	2,17E-24	0,1	0,395	0,0395	8,55E-26
16E-6	25	1,67E+12	5,64E+12	4,98E+13	1,20E+17	1,24E+23	8,09E-24	0,3	0,217	0,0651	5,27E-25
	50	3,23E+12	1,66E+13	3,17E+14	1,48E+18	8,15E+20	1,23E-21	0,4	0,217	0,0868	1,06E-22
	75	4,40E+12	2,67E+13	6,31E+14	3,93E+18	7,59E+19	1,32E-20	0,2	0,217	0,0434	5,72E-22
	100	5,28E+12	3,58E+13	9,25E+14	6,03E+18	1,77E+19	5,65E-20	0,1	0,217	0,0217	1,23E-21
64E-6	25	6,70E+12	2,26E+13	1,99E+14	4,79E+17	4,73E+18	2,11E-19	0,3	0,282	0,0846	1,79E-20
	50	1,29E+13	6,66E+13	1,27E+15	5,91E+18	3,13E+16	3,20E-17	0,4	0,282	0,1128	3,61E-18
	75	1,76E+13	1,07E+14	2,52E+15	1,57E+19	2,91E+15	3,44E-16	0,2	0,282	0,0564	1,94E-17
	100	2,11E+13	1,43E+14	3,70E+15	2,41E+19	6,78E+14	1,47E-15	0,1	0,282	0,0282	4,16E-17
256E-6	25	2,68E+13	9,02E+13	7,97E+14	1,92E+18	1,81E+14	5,51E-15	0,3	0,098	0,0294	1,62E-16
	50	5,17E+13	2,66E+14	5,07E+15	2,36E+19	1,20E+12	8,35E-13	0,4	0,098	0,0392	3,27E-14
	75	7,04E+13	4,27E+14	1,01E+16	6,29E+19	1,11E+11	8,97E-12	0,2	0,098	0,0196	1,76E-13
	100	8,45E+13	5,73E+14	1,48E+16	9,65E+19	2,60E+10	3,85E-11	0,1	0,098	0,0098	3,77E-13
1024E-6	25	1,07E+14	3,61E+14	3,19E+15	7,67E+18	6,95E+09	1,44E-10	0,3	0,008	0,0024	3,45E-13
	50	2,07E+14	1,07E+15	2,03E+16	9,45E+19	4,59E+07	2,18E-08	0,4	0,008	0,0032	6,97E-11
	75	2,81E+14	1,71E+15	4,04E+16	2,51E+20	4,27E+06	2,34E-07	0,2	0,008	0,0016	3,74E-10
	100	3,38E+14	2,29E+15	5,92E+16	3,86E+20	9,96E+05	1,00E-06	0,1	0,008	0,0008	8,03E-10
										Damage (1/s)	1,25E-09
										Total Lifetime (s)	8,01E+08
										Lifetime (years)	25,41

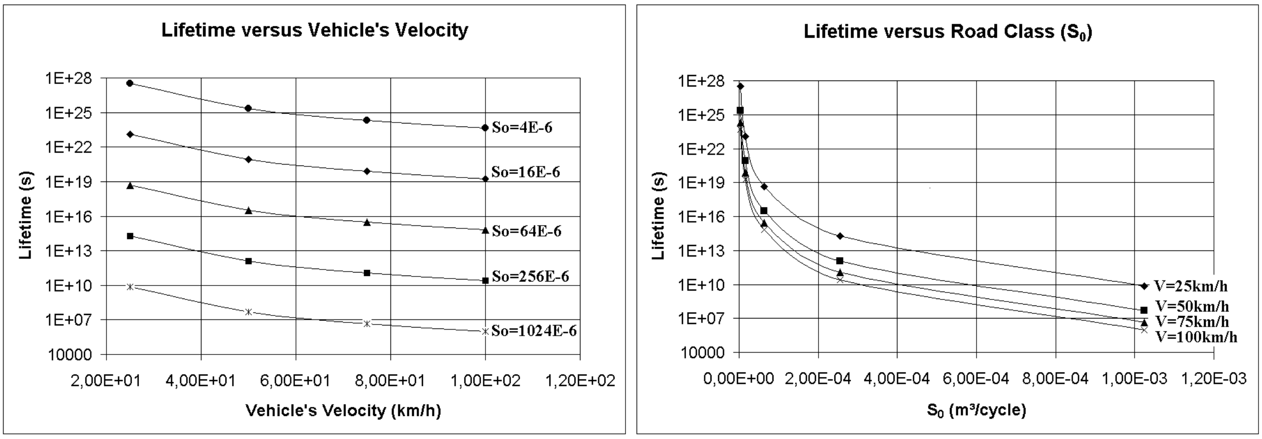

Figure 2. Influence of Vehicle's Velocity and Road Class in the Fatigue lifetime

5. Conclusion

Through the present study, the following conclusions can be drawn when implementing probabilistic structural analysis for the suspension assembly of semi-trailer tank vehicle:

1) The modeling of all components of the suspension assembly is viable once even a non-linear analysis for the full model could be performed without consuming too much CPU time.

2) The Hull Girder approach is a consistent simplification of the shell modeling of the tank. This approach is more adequate for a frequency domain PSD analysis because allows taking into account the inertial effects of the carried liquid more easily.

3) The PSD analysis of the semi-trailer tank was performed successfully without consuming too much CPU time.

4) The PSD analysis can be performed for only one road class and the results of the others are obtained by performing the multiplication of the stress PSD moments - calculated - by relation between constant values.

5) The components lifetime is much more affected by the road class than by the vehicle's velocity.

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7. Acknowledgements

Brazilian research funding agency, CAPES (Coordenação de Aperfeiçoamento de Pessoal de Nível Superior), supported this research. The authors are gratefully acknowledged.