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COMPARATIVE STUDY BETWEEN FRONT CABIN SUSPENSION GEOMETRIES AND THEIR VIBRATION ABSORPTION CAPABILITIES.

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Abstract. *Commercial vehicles present certain peculiarities in relation to passenger vehicles. It must be considered that they are vehicles whose use is for long periods of exposure and with variable loads, capable of completely changing the dynamics of the product. The product development scenario always seeks to obtain the best dynamic performance linked to the lowest cost options. Therefore, this study shows a comparative study of vibration absorption between two front cabin padded suspensions, with different geometries and stiffness: 1- Baseline stiffness 600 N/mm; 2-Proposal stiffness 414 N/mm. The proposed suspensions show differences not only in the stiffness but also in the geometry of the rubber, presenting a low-cost solution. The analyzes are based on the transmissibility (0Hz-20Hz) between the chassis and the cab, the vibration absorption at the base of the driver's seat (0Hz-20Hz) and the steering wheel (0Hz-20Hz) for an input signal made on a rough road track at 50km/h. It could be seen that, although not the perfect one as the transmissibility results are still high, the proposed solution managed to reduce both the levels of Hand-Arm Vibration (HAV), as well, as the levels of Whole-Body Vibration (WBV), compared with the solution used in the industry (Baseline).*

Keywords: *cabin suspension, geometries, Hand-Arm Vibration, Whole-Body Vibration.*

1. INTRODUCTION

The vertical dynamics deals with the behavior of the vehicle and its occupants when subjected to excitations from the floor on which the vehicle travels (external excitations), or from the engine, wheel, transmission (internal excitations).

According to Ghosh and Dinavahi (2005), the suspension system provides a fundamental role in the absorption of the mentioned vibrations, and it has as main objectives:
- the improvement of the passenger comfort;
- the increase of the security;
- the maintenance of the physical integrity of the load transported.

The proposed analysis focuses not only on the truck vibrations itself but also on the influence of the declared vibrations by the users. So, it is important to know that the human body has some natural frequencies that when in resonance with the frequencies present in the vehicle, might cause symptoms that goes from a small discomfort to dizziness and nausea (Griffin, 1996), (Mansfield, 2005).

According to Schmitt and Leingang (1975), due to the ability to meet the requirements for automotive applications, elastomeric materials have become a standard for vibration isolation systems. Elastomeric cushions are generally compact, low cost and maintenance-free. They can be easily designed to obtain the necessary elastic constants for a given application, they are compact and durable.

According to Swanson (1993), the mathematical model of an elastomeric cushion is illustrated in Figure 1. Also known as the Voigt model, it consists of a spring and a damper mounted in parallel.

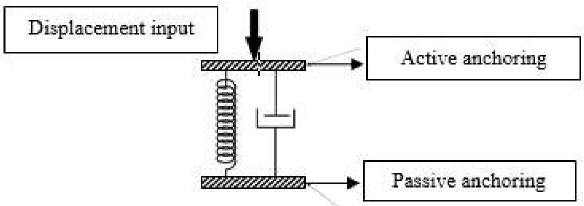

Figure 1. Mechanical model of an elastomeric cushion.
Adapted from (Swanson, 1993)

According to Yu, Naganathan and Dukkipati (1999), elastomeric cushions are also capable of providing dynamic stiffness as shown in Figure 2. The elastic constant or stiffness of the cushion increases as the excitation frequency increases due to the cushioning present in the cushion.

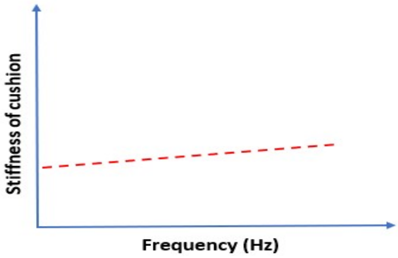

Figure 2. Dynamic stiffness of an elastomeric pad. Adapted from (Yu, Naganathan and Dukkipati, 1999)

As they concluded, based on this rigidity characteristic, it is possible to infer that the elastomeric pad can provide an acceptable compromise between vibration isolation and static displacement.

2. VEHICLE CONFIGURATION

The vehicle used in the tests (Figure 3) was a medium truck with traction 6x2, motorization 280 CV, 6 cylinders, a total gross height of 24 tons and automatic transmission of 6 speeds.

Note: The air suspension of the driver's seat was disabled for all evaluations.

Figure 3. Vehicle 6x2 (Iveco Latin America)

According to Guo et al (2021) the dynamic model of the three-axle heavy-duty commercial vehicle consists of the cab, the carriage, the chassis, the suspensions and mounts, the axles, and the tires. The cab and the carriage are supported on the chassis by elastic mounts, and the axles are connected to the chassis with suspensions. All suspensions and elastic mounts are modelled with parallel springs and dampers with equivalent stiffness and damping coefficients. The schematic of the vehicle model is shown in Figure 4.

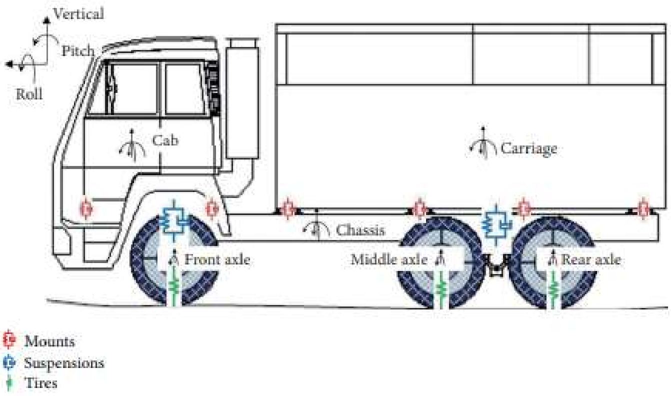

Figure 4. Schematic of the heavy-duty vehicle model (Guo et al, 2021)

3. CABIN PADDED SUSPENSION

In trucks, the damping of the vibration originating from the primary suspension and chassis to the cabin can be made by the cabin padded suspension, as presented in Figure 5. It is a component made of metal-rubber that besides offering support, has elastic properties that makes it able to isolate and mitigate the vibrations generated by the track. Figure 5 shows the geometry of the front cabin suspension for the baseline configuration. Since it is a commercial solution, it will not be possible to disclose the cabin suspension geometry of the proposed configuration for reasons of industrial secret.

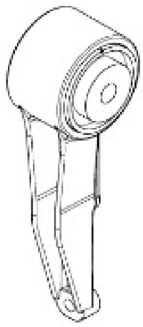

Figure 5. Front cabin padded suspension - baseline (Iveco Latin America)

4. METHODOLOGY

The resonances in the frequency range of interest are obtained by accelerometers installed at main points in the vehicle, allowing analyzing if the levels of the front cabin suspension are acceptable or beyond the limit (Mrad et al, 2018).

For this analysis, only the accelerations measured on the Z axis will be considered, as the geometry of the suspensions has a purely vertical displacement (Z), with the other axes (X, Y) having negligible displacements.

The aim is to compare the effect of the rigidity of the front cabin suspension pad in two configurations: Baseline *versus* Proposal in the transmissibility of the vibration of the chassis *versus* cabin. The configuration of each front cab pad analyzed is given below:

Baseline:
Rz – 600 N/mm
Proposal:
Rz – 414 N/mm

The chosen method for testing the vibration exposure is based on the quantity required among the three axes (in this case, just Z axis). As an alternative to the use of the W_h filter , the RMS acceleration values from one-third-octave band analysis can be used to obtain the corresponding frequency-weighted acceleration.The RMS frequency-weighted acceleration a_{hw} can be calculated, in accordance with equation (1) (ISO5349-1, 2001), (EU HAV, 2006).

$$a_{hv} = \sqrt{\sum_i (W_{hi} a_{hi})^2} \qquad (1)$$

Where W_{hi} is the weighting factor for the i^{th} one-third-octave band and a_{hi} is the RMS acceleration measured in the i^{th} one-third-octave band, in meter per second squared (m/s^2).

The daily exposure value vibration, *A (8),* for a reference period of 8 hours, where T is the time of the total exposure associated with a_{hv} is given by equation (2):

$$a(T_0) = a_{hv}\sqrt{\frac{T}{T_0}} \qquad (2)$$

Where $a(T_0)$, a_{hv}, T, T_0 are the total daily exposure for a period of 8h, the total weighted acceleration m/s^2 according to eq. (1), time in hours of equipment usage, time for a period of 8h, respectively.

For whole-body vibration the calculation is based on the following formula: (ISO2631-1, 1997)

$$a_w = \left[\frac{1}{T}\int_0^T a_w^2(t)dt\right]^{\frac{1}{2}} \qquad (3)$$

Where $a_w(t)$ is the weighted acceleration as a function of time (time history), in meters per second squared (m/s^2) and T is the duration of the measurement, in seconds.

To calculate $a_{w,}$ one uses Wk (Z axis) and Wd (X and Y axis) as the weighting filters for WBV comfort levels.

5. INSTRUMENTATION

Figure 6 shows an sketch of the instrumented points. It was used four tri-axial accelerometers (Figure 7), being those at the left front suspension cab (A1), at the left front chassis (A2), at the top of the steering wheel (A3) and at the driver seat base (A4) and one seat pad (that has also a triaxial accelerometer - Figure 9) on the driver seat (A5). The accelerometers were used in the analysis of transmissibility between cab *versus* chassis and in the comparisons of absorption in the steering wheel and driver's seat.

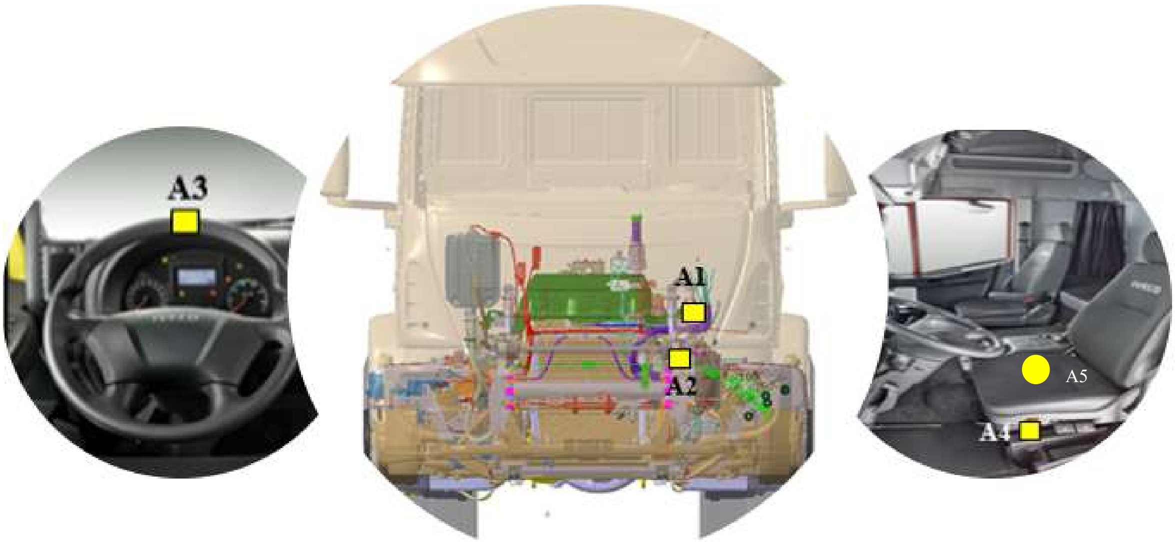

Figure 6: Sketch of instrumentation

Figure 7: Silicon Designs tri-axial Accelerometers 2480-050

The softwares used for signals analysis were Catman Professional 6.0 (HBM, 2009) and Diadem 2012 (NI, 2012) and the hardware was the MGC Plus (HBM,2004) (Figure 8).

Figure 8: MGC PLUS HBM

The Seat Pad Larson Davis with the vibration meter HVM100 were used in the analysis of the whole body vibration calculation (Figure 9)

Figure 9: Seat Pad Larson Davis

6. RESULTS

Figure 10 presents the filter capacity of the front cabin suspension baseline and proposal in relation to the inputs coming from the track.

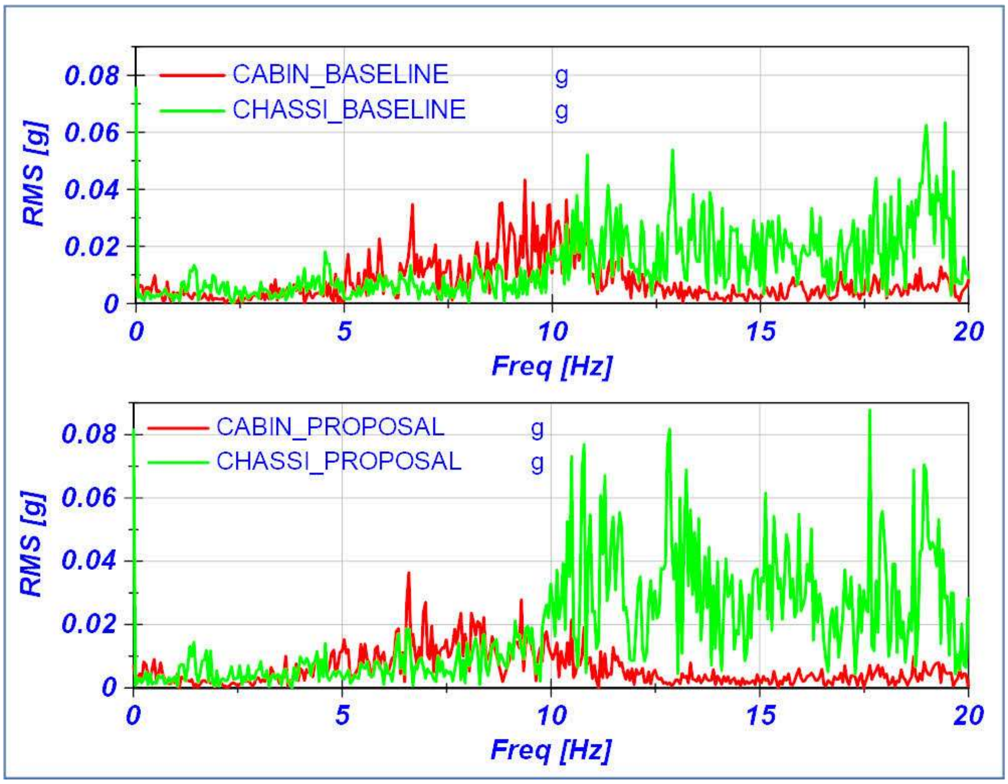

Figure 10: chassis *versus* cab filtering efficiency

The graph shows that the absorption capacity of both configurations of rubbers, in the frequency range from 5Hz to 10Hz, does not present efficient absorption results, because the relation of attenuation of vibration between input (chassi) versus output (cabin) was not obeyed. That may be explained by the fact that the dynamic stiffness of the evaluated elastomers present resonance frequency in this range.,It is important to be clear that the quality of the insulation of a rubber pad artifact is linked, in addition to the deformation frequency to which the material will be submitted, to the amount of deformation and the working temperature of this material. However, analyzing the overall efficiency between the baseline versus proposed configuration, it was observed that the proposed rubber pad, obtained much more satisfactory results, because it was able to more efficiently attenuate the acceleration peaks received by the cabin.

Figure 11 shows a comparison of filtering performance between the two configurations, at the steering wheel position. Analysing this graph, it is clear that for the vibrations measured at the steering wheel, the proposed rubber pad has managed to considerably attenuate the measured amplitudes, reducing the level of discomfort observed in this location.

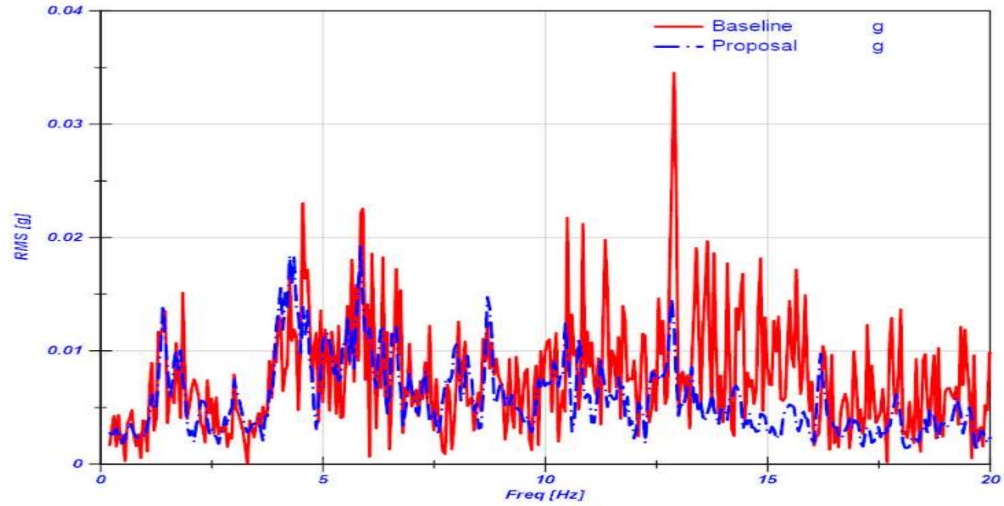

Figure 11: Steering wheel accelerations: Baseline *versus* Proposal

Figure 12 shows a comparison of filtering performance between the two configurations, at the driver seat base position. This graph shows, once again, a good vibration absorption carried out by the rubber pad proposed at the base of the driver's seat, proving a better efficiency of this configuration in the filtering relation for the cabin.

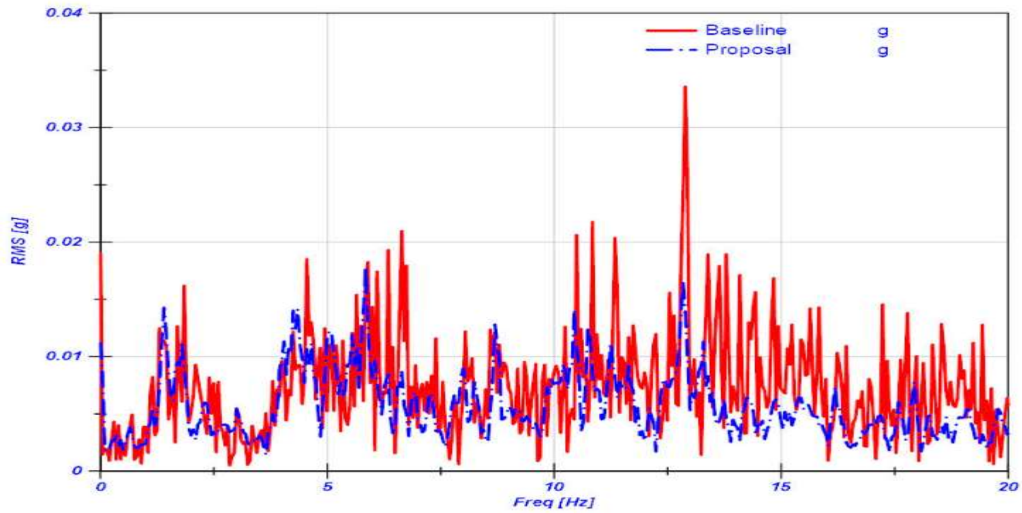

Figure 12: Driver seat accelerations: Baseline *versus* Proposal

Table 1 shows the subjective analysis performed with the presence of 11 participants obtained according to the SAE scale presented at Table 2.
Note: Due to the company's internal rules, the real overall values could not be disclosed. So, a mathematical relationship was established, in which the evaluated proposal was 37.5% more effective than the baseline in relation to the calculated acceleration.

At this stage, 11 top experienced drivers were considered, without knowing any information regarding the changes made to the vehicle. All performed the assessment on a scale of 1 to 10, whose overall was considered in accordance with the values in the SAE table presented in Table 2.

Itens evaluated	Baseline	Proposal
Steeringwheel	100%	137.50%
Driver seat	100%	137.50%
	Overall	Overall

Table 1: Overall subjective analysis according to Iveco Standard 16-1026 scale presented in Table 2

		NOt ACCEPTABLE				ACCEPTABLE					
CUSTOMER STANDPOINT AS TO PARAMETER TO EVALUATE	DETECTED BY	ALL CUST.	AVERAGE CUSTOMERS			CRITICAL CUSTOMERS		EXPERIENCED CUSTOMERS			
	PERCEPTIBILITY	HIGH			MODERATE			SMALL		VERY SMALL	NULL
	SENSATION	INTOLERABILITY		STRONG NUISANCE	NUISANCE	SLIGHT NUISANCE	SMALL INCONVENIENCE		SLIGHT INCONVENIENCE	NO INCONVENIENCE	
	REACTION	REJECTION		REMONSTRATION		CLAIM	ACCEPTANCE		APPRECIATION		ENTHUSIASM
ALPHA–NUMERIC SCALE	EVALUATION	VERY POOR	POOR	LESS THAN MEDIOCRE	MEDIOCRE	ACCEPTABLE LIMIT	ACCEPTABLE	SUFFICIENT	GOOD	VERY GOOD	EXCELLENT
	VOTE	1	2	3	4	5	6	7	8	9	10
REQUIRED CORRECTIVE ACTIONS		RADICAL AND IMMEDIATE ACTION		IMPROVING AND TIMELY ACTION			TO IMPROVE	CARE REQUIRED		OK	

Table 2: Subjective analysis (IS16-1026)

6.1 HAND-ARM VIBRATION ANALYSIS (at the steering wheel)

The vibration total value obtained according to eq. (1) at the steering weel are given below:

BASELINE a_{hv} = X m/s^2
PROPOSAL a_{hv} = 0,74*X m/s^2

Due to the company's internal rules, the real acceleration values could not be disclosed. So, a mathematical relationship was established, in which the evaluated proposal was 74% more effective than the baseline in relation to the calculated acceleration.

Considering the standards for assessing hand and arm vibrations, the hours of exposure to reach the Exposure Action Value (EAV) related to the signals measured on the steering wheel were calculated according to eq. (2). These are presented in Figure 13. The EAV is a value that when achieved for an 8h period, the employer has to take action to reduce the levels. At present, it is 2.5 m/s^2 for Hand-Arm Vibration (Directive EU/44/2002, 2002).

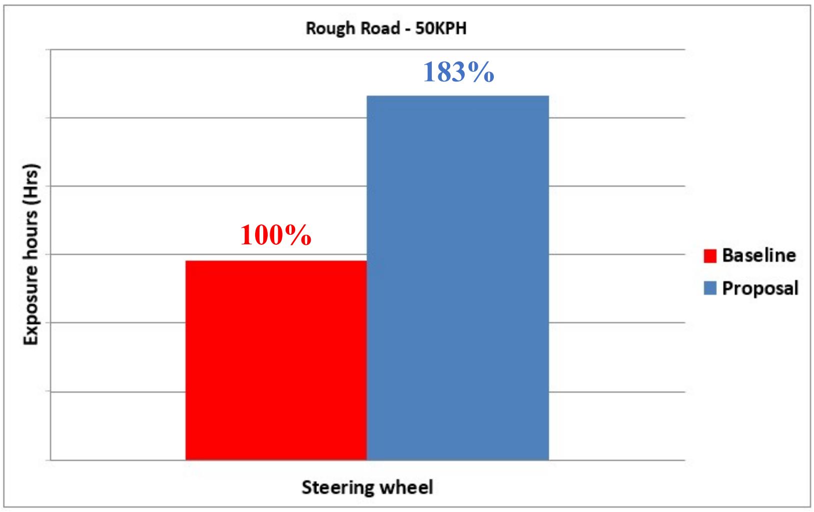

Figure 13: Graph of exposure hours to reach the EAV for hand-arm vibration

The results were made available in a bar graph that shows the considerable gain obtained in the proposed rubber configuration, with an improvement of approximately 83% in the levels of vibrations filtering at the steering wheel.

6.2 WHOLE BODY VIBRATION ANALYSIS

For the evaluation of whole body vibration, the seat pad installed in the driver's seat was used to acquire the vibration the driver was exposed according to equation (3). However, since only z direction was relevant for the study, the other axes were supressed. Then, that was used to calculate the hours one is exposed to reach the EAV, that for WBV has the value of 1.15m/s^2 (Directive EU/44/2002, 2002). For that, eq. (2) can be used, making the equivalent considerations where a_{hv} is changed to a_w.

From the measurement made at the driver's seat the values obtained were the ones presented in Figure 14.

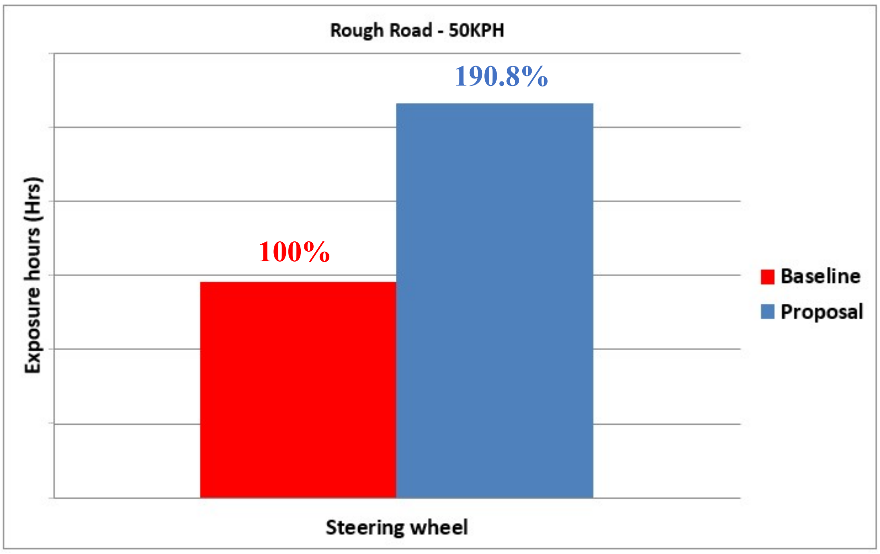

Figure 14: Graph of exposure hours of to reach the EAV for whole body vibration

The bar graph reflects the gain obtained in filtering the vibrations of the driver's seat obtained by the proposed rubber configuration, proving the best damping efficiency compared to baseline rubber (improvememt of 90,80%).

It is important to recall that, for this evaluation, the air suspension of the seat was deactivated.

7. CONCLUSION

One can see in the graphs presented that the vibration between 5Hz to 10Hz in both configurations (baseline and proposed) obtained a poor behavior, having an output with levels higher than the input, that is, there was an acceleration amplification in this frequency range due to issues of dynamic stiffness of the material..

However, the proposed solution, managed to improve this input x output relationship, better filtering the vibrations in this frequency range, while the baseline obtained higher values, presenting a poorer filtering efficiency.

In the individualized analysis by systems, one noticed that both the vibrations that arrive at the steering wheel and at the base of the driver's seat, the proposal configuration managed to reduce the amplitude levels and obtained better absorption performance compared to the baseline.

One knows that padded suspensions are not able to obtain the same efficiency and impact absorption performance as spring suspensions plus shock absorbers. However, this study shows that, working well with the concept of rubber pad geometry, making it less blocked in the main axes movement (X, Y, Z) and calculating a correct rigidity, based on the mass of the cabin on which it will be dependent on the absorption capacity of this elastomer, a good dynamic comfort behavior is achieved with a low development cost.

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