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Optimization of Suspension Characteristics for Increasing Expected Daily Exposure Durations in Vehicles According to ISO 2631-1 Standard using Genetic Algorithms

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ABSTRACT

Human perception related vehicle comfort problems have been an important topic for researchers as evident from a large number of scientific papers available in the literature. Increasing duration of expected daily exposure in vehicles means that the vehicle is comfortably driven for longer periods of time. The aim of this study is to increase the comfortable driving time of a bus by optimizing the suspension system characteristics using genetic algorithm approach. Thus a full car model of the corresponding vehicle is built in order to improve its driving comfort. In the optimization procedure, maximum allowed vertical acceleration of the seat and suspension deflection parameters are used as the cost functions. Furthermore, genetic algorithm parameters and weighting factors are defined in accordance with the problem and applied to solve the optimization problem. Comparison between optimized and current suspension systems is made by using health guidance zones in the International Organization for Standardization 2631-1 to evaluate comfort. As a result, duration of the expected daily exposure time is increased for different types roads. This method can be applied to different types of vehicles and can be a guidance tool for vehicle manufacturers to produce comfortable vehicles.

Keywords: Human perception, vehicle comfort, duration of the expected daily exposure time
I-INCE Classification of Subject Number: 49

1. INTRODUCTION

Vehicle vibrations has anatomical and mental effects such as motion sickness on human health and comfort. Vibration comfort is an important issue in terms of human health. Thus minimizing the tiredness and driver's discomfort feeling in long journeys is an significant problem that automotive manufacturers should overcome (1). Human perception related vehicle comfort problems have been an important topic for researchers as evident from a large number of scientific papers available in the literature. A number of tests have been carried out to measure the comfort of the vehicles at different road roughness. According to the results of tests made, it is tried to produce vehicles with ideal vehicle vibration for human health (2). The vehicles in motion vibrate mainly with

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excitation from the road (13). These vibrations are generally in the frequency range of 0 Hz to 25 Hz (3). The excitation that the vehicle gets from the road pass through the wheel system and the axle body via the spring and damper elements (4).

An increased expected daily exposure duration in vehicles corresponds to an increase in the period of time that the vehicle is driven. The aim of this study is to increase the comfortable driving time of a bus by optimizing the suspension system characteristics using genetic algorithm approach. A genetic algorithm is applied to define optimum parameters of the seat in order to minimize bus suspension deflection and driver's body vertical acceleration to achieve the best comfort of the driver. Vehicle suspension systems aim to improve the ride comfort, handling, road holding, suspension deflection, and static deflection (8). In the optimization procedure, maximum allowed vertical acceleration of the seat and suspension deflection parameters are used as the cost functions. Furthermore, genetic algorithm parameters and weighting factors are defined in accordance with the problem and applied to solve the optimization problem.

Many of mathematical models are suitable to explore the bus driving comfort. A full vehicle model of the corresponding vehicle is built in order to improve its driving comfort. The full car model of the bus, which is intended to be improved in comfort, is created with Matlab software (14). The generated vibration system has eight degrees of freedom. As road roughness, Power spectral densities (PSD) depending on reference angular spatial frequency measured in different classes of road is used for full vehicle model. Experimentally measured PSD depending on reference angular spatial frequency were converted to amplitude.

ISO 2631-1 Standard is to describe quantitative whole body vibration methods: human health and comfort, possibility of vibration detection, incidence of illness caused vibration. According to International Organization for Standardization 2631-1, the root mean square (RMS) value of the frequency-weighted acceleration is obtained using with seat accelerations (5).

In this paper, comparison between optimized with genetic algorithm and current suspension (previous) systems is made by using health guidance zones in the International Organization for Standardization 2631-1 to evaluate duration of the expected daily exposure. The results showed that an active suspension using genetic algorithms successfully increased the duration of the expected daily exposure time.

2. METHOD

2.1 Mathematical Model of the Bus

In order to evaluate comfort more suitable, the full vehicle model of the bus, which is intended to be improved in comfort, is created with Matlab software. The full vehicle model with passive suspension systems is shown in Figure 1. The bus vibration model are consisting of mass, spring and damping elements. The vehicle body have pitch and roll motion. The full car model vibration model has eight degrees of freedom. These degrees of freedom are one of the vertical motion of the seat; one of the vertical and two rotational movements of the body; four of the vertical movements of the wheels; the degree of freedom of one's movements. To be simple, the rolling and pitch angle have been assumed to be small. The suspensions between the sprung mass and unsprung mass are assumed linear elements. The tires are modeled as linear springs without dampers (12). Newton's second law of motion is used with free-body diagram concept, the following eight equations. Full vehicle model is shown in Figure 1 and definitions of mathematical model's symbols are given in Table 1. The differential equations of vehicle

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body, suspension motions and seat are derived by using Newton’s laws of motion as follows:

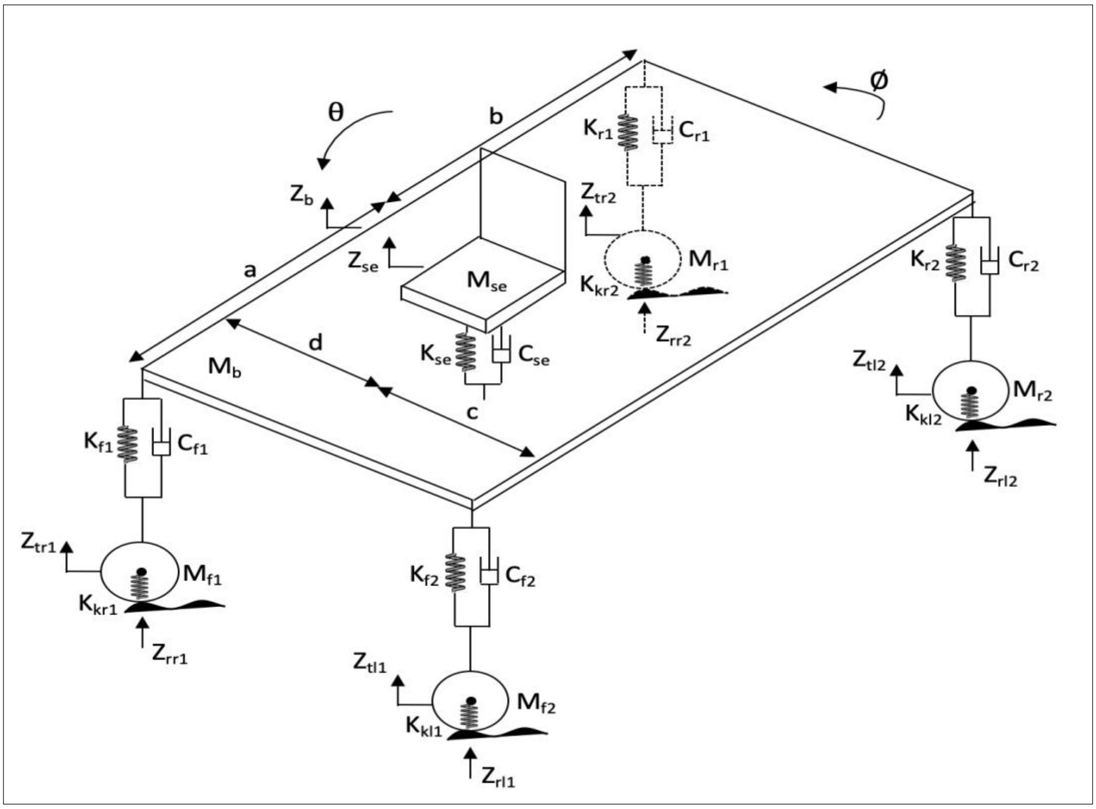

Figure 1. Full vehicle model

Table 1. Symbols and definitions of model

Symbols	**Definitions**
M_b	Mass of vehicle body (kg)
$M_{f1} M_{f2}$	Mass of front right and left wheel (kg)
$M_{r1} M_{r2}$	Mass of rear right and left wheel (kg)
M_{se}	Mass of seat (kg)
Z_b	Displacement of vehicle body (m)
$Z_{rr1} Z_{rl1}$	Road roughness of front right and left (m)
$Z_{rr2} Z_{rl2}$	Road roughness of rear right and left (m)
Z_{se}	Displacement of seat (m)
$Z_{tr1} Z_{tl1}$	Displacement of front right and left wheel (m)
$Z_{tr2} Z_{tl2}$	Displacement of rear right and left wheel (m)
$K_{kr1} K_{kl1}$	Spring stiffness of front right and left wheel (N/m)
$K_{kr2} K_{kl2}$	Spring stiffness of rear right and left wheel (N/m)
$K_{f1} K_{f2}$	Spring stiffness of front right and left suspension (N/m)
$K_{r1} K_{r2}$	Spring stiffness of rear right and left suspension (N/m)
$C_{f1} C_{f2}$	Damping coefficient of front right and left suspension (Ns/m)
$C_{r1} C_{r2}$	Damping coefficient of rear right and left suspension (Ns/m)
$K_{se} C_{se}$	Spring stiffness and damping coefficient of seat
θ	Pitch angle of body
Ø	Roll angle of body

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$$
\begin{aligned}
&M_b\ddot{Z_b} + Z_b(K_{f1} + K_{f2} + K_{r1} + K_{r2}) + (C_{f1} + C_{f2} + C_{r1} + C_{r2})\dot{Z_b} \\
&\quad - (K_{f1}a + K_{f2}a - K_{r1}b - K_{r2}b)\theta - (C_{f1}a + C_{f2}a - C_{r1}b + C_{sl2}b)\dot{\theta} \\
&\quad - (K_{f1}c - K_{f2}d - K_{r1}c + K_{r2}d)\emptyset - (C_{f1}c - C_{f2}d + C_{r1}c - C_{r2}d)\dot{\emptyset} \\
&\quad + K_{se}(Z_b - Z_{se} + \theta c - \emptyset a) + C_{se}\left(\dot{Z_b} - \dot{Z_{se}} + \dot{\theta}c - \dot{\emptyset}a\right) - K_{f1}Z_{wr1} \\
&\quad - K_{f2}Z_{tl1} - K_{r1}Z_{tr2} - K_{r2}Z_{tl2} - C_{f1}\dot{Z_{tr1}} - C_{f2}\dot{Z_{tl1}} - C_{r1}\dot{Z_{tr2}} - C_{r2}\dot{Z_{tl2}} \\
&\quad = 0 \qquad 1
\end{aligned}
$$

$$
\begin{aligned}
&I_{yy}\ddot{\theta} - K_{f1}(Z_b - \theta a - \emptyset c + Z_{tr1})a - K_{f2}(Z_b - \theta a + \emptyset d - Z_{tl1})a \\
&\quad + K_{r1}(Z_b + \theta b - \emptyset c - Z_{tr2})b + K_{r2}(Z_b + \theta b + \emptyset d - Z_{tl2})b \\
&\quad + C_{f1}\left(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{tr1}}\right)a - C_{f2}\left(\dot{Z_b} - \dot{\theta}a + \dot{\emptyset}d - \dot{Z_{tl1}}\right)a \\
&\quad + C_{r1}\left(\dot{Z_b} + \dot{\theta}b - \dot{\emptyset}c - \dot{Z_{tr2}}\right) + C_{r2}\left(\dot{Z_b} + \dot{\theta}b + \dot{\emptyset}d - \dot{Z_{tl2}}\right)b \\
&\quad + K_{se}(Z_b - Z_{se} + \theta c - \emptyset a) + C_{se}\left(\dot{Z_b} - \dot{Z_{se}} + \dot{\theta}c - \dot{\emptyset}a\right) = 0 \qquad 2
\end{aligned}
$$

$$
\begin{aligned}
&I_{xx}\ddot{\emptyset} - -K_{f1}(Z_b - \theta a - \emptyset c + Z_{tr1})c + K_{f2}(Z_b - \theta a + \emptyset d - Z_{tl1})d \\
&\quad - K_{r1}(Z_b + \theta b - \emptyset c - Z_{tr2})c + K_{r2}(Z_b + \theta b + \emptyset d - Z_{tl2})d \\
&\quad - C_{f1}\left(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{tr1}}\right)c + C_{f2}\left(\dot{Z_b} - \dot{\theta}a + \dot{\emptyset}d - \dot{Z_{tl1}}\right)d \\
&\quad - C_{r1}\left(\dot{Z_b} + \dot{\theta}b - \dot{\emptyset}c - Z_{tr2}\right)c + C_{r2}\left(\dot{Z_b} + \dot{\theta}b + \dot{\emptyset}d - \dot{Z_{tl2}}\right)d \\
&\quad - K_{se}(Z_b - Z_{se} + \theta c - \emptyset a) - C_{se}\left(\dot{Z_b} - \dot{Z_{se}} + \dot{\theta}c - \dot{\emptyset}a\right) = 0 \qquad 3
\end{aligned}
$$

$$
\begin{aligned}
&M_{f1}\ddot{Z_{wr1}} - K_{f1}(Z_b - \theta a - \emptyset c - Z_{tr1}) + K_{kr1}(Z_{tr1} - Z_{rr1}) \\
&\quad - C_{f1}\left(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{wr1}}\right) = 0 \qquad 4
\end{aligned}
$$

$$
\begin{aligned}
&M_{f2}\ddot{Z_{wl1}} - K_{f2}(Z_b - \theta a - \emptyset c - Z_{tl1}) + K_{kl1}\ddot{(}Z_{tl1} - Z_{rl1}) - C_{f2}(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{tl1}}) \\
&\quad = 0 \qquad 5
\end{aligned}
$$

$$
\begin{aligned}
&M_{r1}\ddot{Z_{wr2}} - K_{r1}(Z_b - \theta a - \emptyset c - Z_{tr2}) + K_{kr2}(Z_{tr2} - Z_{rr2}) - C_{r1}\left(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{tr2}}\right) \\
&\quad = 0 \qquad 6
\end{aligned}
$$

$$
\begin{aligned}
&M_{r2}\ddot{Z_{wl2}} - K_{r2}(Z_b - \theta a - \emptyset c - Z_{tl2}) + K_{kl2}(Z_{tl2} - Z_{rl2}) - C_{r2}\left(\dot{Z_b} - \dot{\theta}a - \dot{\emptyset}c - \dot{Z_{tl2}}\right) \\
&\quad = 0 \qquad 7
\end{aligned}
$$

$$
M_{se}\ddot{Z_{se}} - K_{se}(Z_b - Z_{se} + \theta c - \emptyset a) - C_{se}\left(\dot{Z_b} - \dot{Z_{se}} + \dot{\theta}c - \dot{\emptyset}a\right) = 0 \qquad 8
$$

2.2 Road Roughness

Power spectral density (PSD) is the limiting mean-square value of one signal per unit frequency bandwidth. The road profile can be explained by the vertical displacement by the PSD (6). It is necessary to define road roughness from the road in terms of amplitude so that it can be calculated real and imaginary amplitude values from the transfer function of the seat and road roughness of full car model. Road roughness for the different types of roads in the measurements made are given in terms of the displacement PSD ($\Phi_h(\Omega_0)[cm^3]$).However, for using the $\hat{z}/\hat{h}$ transfer function from the model of the full vehicle model, we need road roughness in amplitude $\hat{h}$ so that we can find the seat vertical acceleration rms values. Because of this reason, PSD depending on reference angular spatial frequency (1 rad/m) ($\Phi_h(\Omega_0)$) is converted to amplitude. The amplitude

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was used instead of the PSD depending on angular spatial frequency, conversion is as follows equations 9,10,11;

$$\hat{h} = \int_0^\infty \Phi_h(\Omega)d\Omega = \int_0^\infty \Phi_h(L)dL \qquad 9$$

Derivative of $L = 2\pi/\Omega$ is $dL = -2\pi/\Omega^2 d\Omega$ equation

$$\Phi_h(L) = \frac{\Omega^2}{2\pi}\Phi_h(\Omega) = \frac{\Omega^2}{2\pi}\left(\frac{\Omega_0}{\Omega}\right)^w \Phi_h(\Omega_0) \qquad 10$$

For this calculation w is 2:

$$\Phi_h(L) = \frac{\Omega^2}{2\pi}\Phi_h(\Omega_0) = \text{constant} \qquad 11$$

Ω : Angular spatial frequency
Ω_0: Reference angular spatial frequency
L : Average wavelength
$\Phi_h(\Omega)$: Displacement PSD depending on angular spatial frequency
$\Phi_h(\Omega_0)$: PSD depending on reference angular spatial frequency
$\Phi_h(L)$: PSD function of wavelength

$\Phi_h(L)$ is equal to the constant value. The r.m.s value of the excitation function (Δh) can be generated in the wavelength range (ΔL) and a single harmonic function can be generated by the $\hat{b}_{abs}$amplitude to be generated for the average wavelength (L). The factor of $\sqrt{2}$ for the conversion between the amplitude and the effective value of the harmonic function should be considered (7).

$$\widetilde{\Delta h} = \sqrt{\int_{\bar{L}-\Delta L/2}^{\bar{L}+\Delta L/2} \Phi_h(L)dL} = \sqrt{\Phi_h(\bar{L})\Delta L} = \frac{\hat{b}_{abs}}{\sqrt{2}} \qquad 12$$

$$\hat{b}_{abs} = \sqrt{\frac{\Omega^2}{\pi}\Phi_h(\Omega_0)\Delta L} \qquad 13$$

The $\hat{b}_{abs}$ amplitude equals a constant value in a constant wavelength range (ΔL). Amplitude transformations of the $\Phi_h(\Omega)$ measurements made for good concrete, medium asphalt and very bad parquet types of roads given in Table 2 were analyzed for this study.

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Table 2. PSD depending on reference angular spatial frequency for different types of roads

Road Class	Road Type	w	$\Phi_h(\Omega_0)$
Concrete	Very good	2.29	0.6
	Good	**1.97**	**4.5**
	Medium	1.97	8.7
	Bad	1.72	56.0
Asphalt	Very good	2.20	1.3
	Good	2.18	6.0
	Medium	**2.18**	**23.0**
Parquet	Good	1.75	14
	Medium	1.75	23
	Bad	1.81	123
	Very bad	**1.81**	**602**

2.3 ISO 2631-1 Frequency Weighting Curve

People's sensitivity to whole body vibrations varies with frequencies (5). In the study, ISO 2631-1 W_k frequency weighting curves was used. Because W_k weighting factor represents vertical direction of seat. The frequency weighting curve in the 1/3 octave band in ISO 2631-1 is shown in Figure 2a. The frequency weighting curve for principal weightings shown in Figure 6a was used in vehicle comfort calculations. However, in order to use this curve in our study, it was necessary to convert the values from the decibel unit to the amplitude unit. It is shown in Figure 2b.

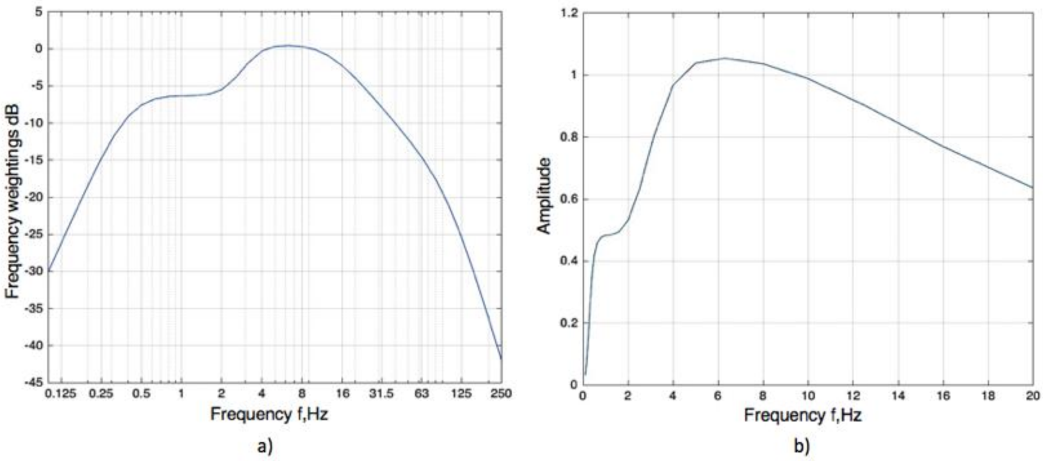

Figure 2. *a) Frequency weighting curves function of dB b) Frequency weighting curves function of amplitude*

2.4 Optimization with Genetic Algorithms

A genetic algorithm, which is inspired by Charles Darwin's theory of natural evolution, is intuitive. This algorithm reflects the natural selection process in which the next generation is selected for reproduction of the next generation's offspring (9).

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The genetic algorithm optimization is applied to optimize the linear suspension system characteristics which are damping coefficients (c_1, c_2, c_3, c_4) and spring constants (k_1, k_2, k_3, k_4) to find the best comfort for the driver. It is minimized car suspension working space and driver's seat acceleration to define the best comfort with this optimization method. Furthermore, genetic algorithm parameters and weighting factors are defined in accordance with the problem and applied to solve the optimization problem. In the optimization process parameters which are given in Table 3 were used. The simulations were carried out in simulink package of MATLAB.

Table 3. Genetic Algorithm Parameters

Genetic Algorithm Input Parameters
50 as population size
Uniform mutation technique
1200 as number of generations
0.84 as generation gap
700-15000 as lower boundary
2000-23000 as upper boundary
Heuristic crossover technique
0.81 as probability of crossover
1e −15 as objective function accuracy

Health of the driver is as important as the stability of the bus so that the seat acceleration is as valuable as the suspension working area(11). For this study, the desired objective is proposed as the minimization of a multi objective function (OBF) consisting with the combination of car suspension working spaces$((x - x_1), (x - x_2), (x - x_3), (x - x_4)$ for 4 suspensions and the seat acceleration $(\ddot{x_s})$. To solve this genetic algorithm optimization problem, the objective function with the weighting factor approach was created. Appropriate weighting factors are calculated and given in the Table 4.

$$OBF = w_1(x - x_1) + w_2(x - x_2) + w_3(x - x_3) + w_4(x - x_4) + w_5\ddot{x_s} \qquad 14$$

Table 4. Weighting Factors

Weighting Factors	w_1	w_1	w_1	w_1	w_1
Input Value	0.5	0.3	0.2	0.4	0.25

3. RESULTS AND DISCUSSION

As a result of the optimization of the genetic algorithm using the current (passive) vehicle parameters, the new suspension system characteristics which are damping coefficients (c_1, c_2, c_3, c_4) and spring constants (k_1, k_2, k_3, k_4) were obtained. The design results from the passive and optimal suspensions are given in Table 5. According to the calculations, the natural frequency of the vehicle body is between 1-2 Hz, the natural frequency of the seat is between 2.5-3.5 Hz, the natural frequency of the axle is 10-12 Hz.

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Table 5. Genetic Algorithm (GA) Optimization results for suspension characteristics

Suspension Characteristics	Currently used (passive)	GA Optimization
k_1	298000 N/m	183072 N/m
k_2	298000 N/m	183072 N/m
k_3	273000 N/m	152031 N/m
k_4	273000 N/m	152031 N/m
c_1	17500 Ns/m	15381 Ns/m
c_2	17500 Ns/m	15381 Ns/m
c_3	18100 Ns/m	13928 Ns/m
c_4	18100 Ns/m	13928 Ns/m

According to International Organization for Standardization 2631-1, the root mean square (RMS) value of the frequency-weighted acceleration is obtained using with seat accelerations. Optimized seat acceleration value obtained by using optimized suspension characteristics. In case of different measured road excitation, results showed that RMS frequency-weighted acceleration of the seat and suspension deflection are reduced after the genetic algorithm optimization. As shown in figure 3, comparison between GA optimized and current suspension (passive) systems is made by using health guidance zones in the International Organization for Standardization 2631-1 to evaluate comfort. The duration of the expected daily exposure of the driver was calculated with using the health guidance zones.

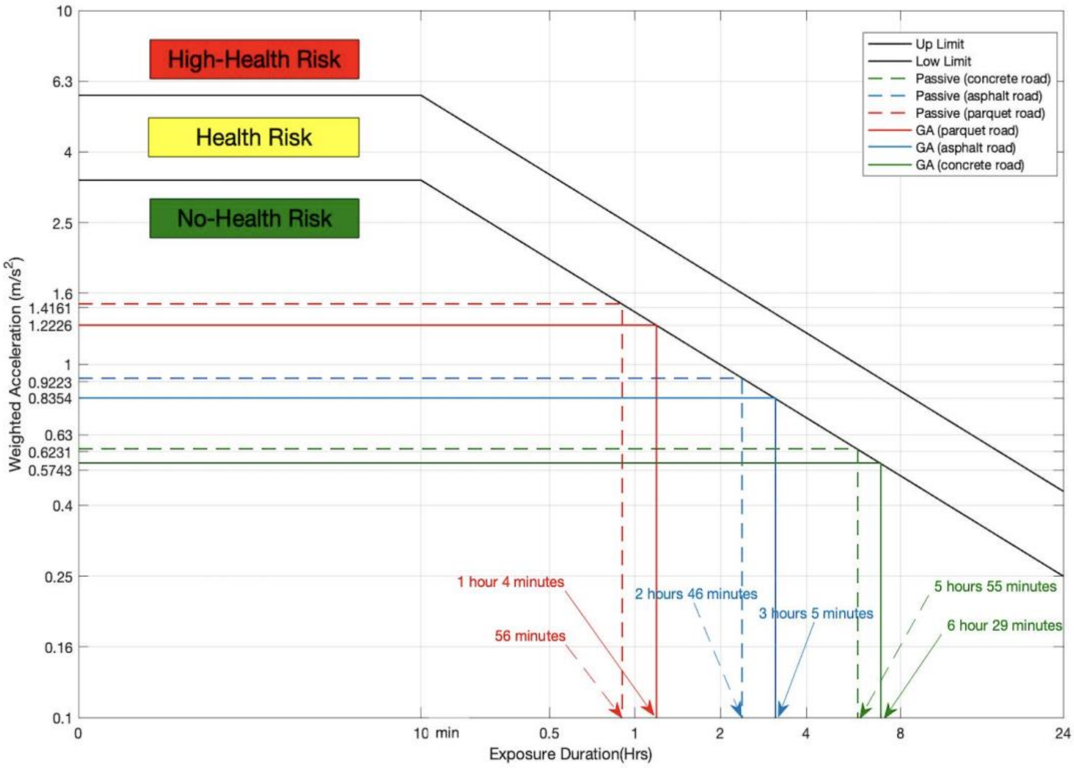

Figure 3. Comparison between GA optimized and current suspension (passive) systems by using health guidance zones

As a result, comfortable driving time and duration of the expected daily exposure time are increased. Table 6 shows that reduction of seat RMS frequency-weighted acceleration for concrete, asphalt and parquet types roads are 7.83%, 9.42%, 13.66%, respectively. Increasing of duration of the expected daily exposure for concrete, asphalt

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and parquet roads are 9.65%, 11.45%, 14.28%, respectively. This method can be applied to different types of vehicles and can be a guidance tool for vehicle manufacturers to produce comfortable vehicles.

Table 6. Comparison of previous and optimized comfort characteristics for different types of roads.

Road types	PSD $\Phi_h(\Omega_0)$	Seat RMS frequency-weighted Acceleration			Duration of the Expected Daily Exposure		
		Previous	Optimized	**Improvement**	Previous	Optimized	**Improvement**
Concrete	$4.5x10^{-6}m^3$	0.6231 m/s^2	0.5743 m/s^2	**7.83%**	5 hours 55 minutes	6 hours 29 minutes	**9.65%**
Asphalt	$23x10^{-6}m^3$	0.9223 m/s^2	0.8354 m/s^2	**9.42%**	2 hours 46 minutes	3 hours 5 minutes	**11.45%**
Parquet	$602x10^{-6}m^3$	1.4161 m/s^2	1.2226 m/s^2	**13.66%**	56 minutes	1 hours 4 minutes	**14.28%**

4. CONCLUSIONS

The new optimized damping coefficients and spring constants were obtained by optimizing current suspension characteristics on the vehicle. Vehicle manufacturers can model their vehicles at the production stage and achieve the best driving time for drivers by optimizing suspension system with genetic algorithm. Through the genetic algorithm optimization, the new suspension characteristic increases the duration of the expected daily exposure and driving time according to ISO 2631-1. Drivers can drive for longer and more comfortable with the increase in these times,

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