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Design of Active Tyre-Suspension-Seat System Through Multibody Model and Genetic Algorithms

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Abstract – *The tyre-suspension-seat dynamic system, driveline and engine vibrations are generally considered in the vibrational field as the main factors that influence the particular feeling of comfort perceived by passengers on a vehicle. Hence, the development of several criteria and models for the optimal estimation of the design parameters of such systems. Among these parameters, the most detrimental impacting on the passenger comfort are undoubtedly acceleration and its variation. The two types of suspension systems (conventional passive suspension system and active suspension system) differ as the first foresees the spring-damper characteristics to be adjusted so that only one of several conflicting objectives (such as passenger comfort, road holding, and suspension deflection) is followed. In active suspension systems, instead, these objectives are balanced by the designer in a more efficient manner thanks to the feedback-controller actuator assembly. However, this approach presents some limitations linked to the extremely wide spectrum of magnitude and frequency of external forces that the tyre-suspension-seat system has to efficiently control and mitigate. It remains that in the existing optimisation models and systems time exposure limits established by unification agencies and road authorities are not generally considered. This paper illustrates the development of an active tyre-suspension-seat system control for passenger cars, using both a non-linear multibody model and Genetic Algorithm (GA) controls. A benefit of the proposed active tyre-suspension-seat system control is also to consider various time exposure limits and an active damping element. The main innovative element introduced by this work consists in having coupled an active control to passive mechanical parameters in order to minimize the seat acceleration. The 3 DoF multibody model, applied to a quarter body for symmetry reasons, treated road roughness as an input variable in the GA control so as to determine the vertical component of acceleration. The numerical and experimental applications of the proposed model to a specific case study allowed to validate the effectiveness of the active system towards the vibrations transmitted to the passenger.* ***Copyright © 2021 The Authors.***

Keywords*:* *Passenger Comfort, Acceleration Variation, Active Suspension System, Genetic Algorithm, Vibrations*

Nomenclature

ω	Forcing frequency
λ	Spectrum of forcing frequency
$\{\boldsymbol{x}\}$	Design variables
C	Damping matrix
c_{se}	Seat damping
C_{sp}	Skyhook proportionality constant dampers
c_{sp}	Sprung damping
c_t	Tire viscoelastic damping
DoF	Degrees-of-Freedom
$f(t)$	Forcing function
GA	Genetic Algorithm
h	Road profile sinusoidal shape height
h	Road profile sinusoidal shape length
IRI	International Roughness Index
ISO	International Standard Organization
K	Stiffness matrix
k_{se}	Seat stiffness
k_{sp}	Sprung stiffness
k_t	Tire stiffness
M	Mass matrix
m_p	Mass passenger
m_{sp}	Sprung mass
m_u	Unsprung mass
$q(t)$	Road disturbance
RMS	Root Mean Square
T	Time interval analysed
V	Veicle longitudinal velocity
WBV	Whole Body Vibration
x	Longitudinal displacement
y	Transversal displacement
z	Vertical displacement

I. Introduction

One of the greatest sources of ride comfort is represented by the level of in-vehicle vibration

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experienced, which is produced by road roughness. The International Roughness Index (IRI) constitutes the most prevalent road roughness measurement; the ISO 2631-1 standard indicates instead the requirements for the evaluation of human exposure to whole-body vibration (WBV) percepted by drivers and passengers on the ride with respect to health and comfort. The definition of IRI is established through a quarter car model. The quarter car model was introduced by Karamihas and Sayers in 1996 [1], and has become the most widely used numerical model adopted by researchers and regulatory bodies ever since [2]-[4].

The vibration due to road roughness that is transmitted to passengers can be evaluated principally by measuring acceleration on the seat at different frequencies. Acceleration intensity can only be assessed and optimised, but can never be completely eliminated in the entire frequency range [5].

Optimal control theory is recognized as one of the most applied theories for the assessment of feedback gains for active suspension systems [6], [7].

In order to improve the suspension performance, instead, Pan and Sun propose an output feedback finite-time control method for stabilizing the perturbed vehicle active suspension system [8]. Fei, Wang and al. propose an adaptive hybrid event-triggered scheme for the cloud-aided quarter-car suspension framework [9].

Equally extensive is the application of numerical optimisation methods to passive suspension design, which is characterized by mechanical parameters (spring and damping values) as the design variables [10].

In many recent applications the vehicle suspension system is controlled using fuzzy and PID controls for a quarter vehicle model. The active suspension systems performance is compared with the passive one to show the improvements. The performance of the Fuzzy and PID controlled suspension systems are generally evaluated mathematically using MATLAB and/or SIMULINK toolbox [11].

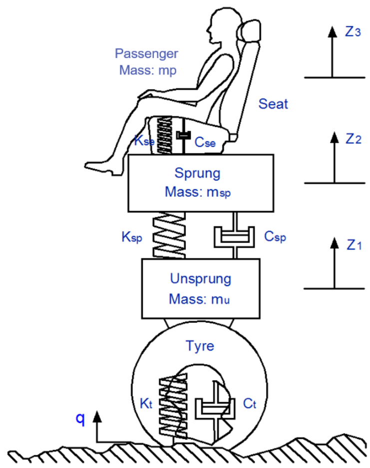
In this paper, an active tyre-suspension-seat system control for passenger cars that uses both active Genetic Algorithm (GA) control and a non-linear multibody model is introduced. The optimization of seat acceleration in the entire frequency spectrum, as recently suggested by Bestle [12] and Schiehlen [13] is gained by adopting a 3 degrees-of-freedom (DoF) multibody model of a quarter car body and wheel assembly used both in an active control and passive mechanical parameters as design variables. Seat vertical acceleration is used as a parameter in order to evaluate WBV and, thus, also passenger comfort. Among the control system also considers time exposure limits established by unification agencies and road authorities.

A multibody model places road roughness, vehicle speeds, suspension values of elasticity, damping, and inertial and seat vertical displacement and acceleration in relation. The use of parametric suspension 3D models allows the exact determination of variability in inertia and structural values, creating an interactive model that is applicable to a wide range of vehicles.

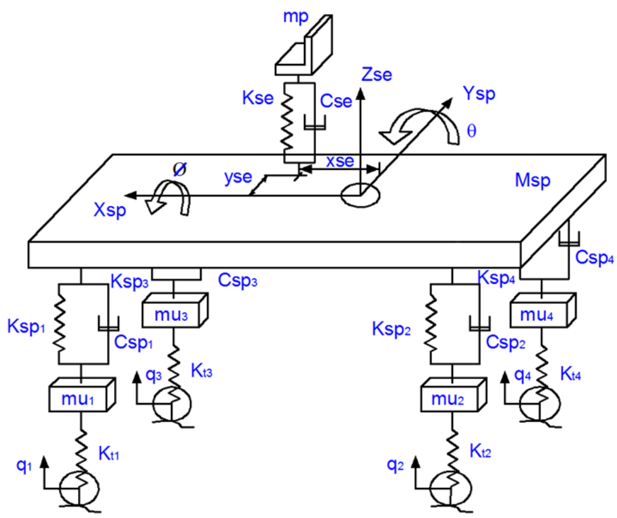
The paper is divided in this introductory session and other five main sections. Section 2 defines the tyre-suspension-seat dynamic multibody model and the GA control system. Section 3 deals experimental setup and describes the multibody model validation. In the Section 4 the optimisation problem and GA algorithm procedure were discussed. Considerations and results interpretations were reported in section 5. Conclusions and future works were drawn in Section 6.

II. Active Tyre-Suspension-Seat Model

By viewing road roughness as an acceleration input the effect of travel speed on seat displacement and acceleration can be studied and optimized with a GA control of an equivalent non-linear multibody model. The schematic representation of the quarter car multibody model with the 3 DoF multibody model adopted in this study can be seen in Fig. 1(a).



(a)



(b)

Figs. 1. (a) Schematic 3 DoF multibody model of a quarter car body; (b) schematic 8 DoF multibody model of entire car

The 3 DoF multibody model (Fig. 1(a)) was selected for the analysis of a quarter car body vehicle dynamics, instead of a more detailed 8 DoF model (Fig. 1(b)) which would also allow longitudinal and transversal load displacement to be considered. The complex calculation appears to be unnecessary for the most frequent applications, since the 8 DoF model provides substantially different values from the 3 DoF model only in cases of considerable longitudinal and transversal load displacement [14].

Table I outlines the design variables considered for the model, their initial values, and the range of variation. The initial values correspond to those of a medium-sized sedan for which the experimental tests are available to validate the model. The variability was set in such a way that most of the medium-sized vehicles currently on the market were included in the study.

TABLE I
DESIGN VARIABLE: INITIAL VALUES AND VARIATION RANGE

	mu [kg]	*msp* [kg]	*mp* [kg]	*kt* [kN/m]	*ct* [Ns/m]
Initial value	25	250	80	100	800
Upper bound	25	300	80	100	800
Lower bound	25	230	80	100	800

	ksp [kN/m]	*csp* [Ns/m]	*kse* [kN/m]	*cse* [Ns/m]
Initial value	25	2150	4	1500
Upper bound	40	2500	6.5	3000
Lower bound	10	1800	3	1000

The 3 DoF multibody model can be applied to optimise all vehicles designated for the transport of persons, the only exception being modern heavy vehicles, since these may exhibit features such as additional suspension between the chassis and the cab.

The multibody model equations of motion were obtained using road roughness, vehicle speeds, suspension values of elasticity, damping, and inertial and seat vertical displacement and acceleration. The equations of motion in matrix form are:

$$[\mathbf{M}]\{\ddot{\boldsymbol{z}}\} + [\mathbf{C}]\{\dot{\boldsymbol{z}}\} + [\mathbf{K}]\{\boldsymbol{z}\} = \{f(t)\} \tag{1}$$

where $[\mathbf{M}]$, $[\mathbf{C}]$ and $[\mathbf{K}]$ are the mass, damping and stiffness matrices, respectively.

The initial mass properties of the system used to validate the model are defined by: the mass passenger m_p= 80 kg, the car-body sprung mass m_{sp}= 250 kg, and the unsprung mass m_u= 25 kg. An active element foresees a force proportional to the absolute vertical velocity between sprung mass and unsprung mass. This device (named skyhook dampers), characterised by proportionality constants C_{sp}, is able to reduce effectively car body motions than passive dampers [13]. As far as the forcing function $\{f(t)\}$ is concerned, it is dependent on the spring-damper model of the tyre, with k_t= 100 kN/m and c_t = 800 N·s/m, and the road disturbance ($q(t)$). Fig. 2 shows the sinusoidal shape of the road profile characterised by height h and length l. The choice of this profile, indicated by the ISO standards, allows to simulate the decomposition into harmonics of any generic road irregularity profile.

In dynamic equations road disturbance is given by:

$$q(t) = \frac{h}{2}\left(1-\cos\left(\omega\ t\right)\right) \tag{2}$$

where the forcing frequency is given by:

$$\omega = \frac{2\pi V}{l} \tag{3}$$

and where height is the vehicle velocity. The remaining six unknown parameters comprise the set of design variables:

$$\{x\} = \{k_t,\ C_t,\ k_{sp},\ C_{sp},\ k_{se},\ C_{se}\}^T \tag{4}$$

As previously stated, seat acceleration is considered to be the most detrimental parameter that impacts the physical comfort of passengers. This is expressed in an objective function as:

$$\min \int_0^T \frac{|\ddot{z}_3(t)|}{|\ddot{q}(t)|}dt \tag{5}$$

where T is the time interval analysed.

Specifically, we are interested in finding the values of the design variables that minimise the ratio of seat acceleration to the input acceleration in the time interval T, throughout the forcing frequency range ($0 \div \omega_{max}$). The spectrum of the forcing frequency can be considered dimensionless as a function of the first frequency of the system itself:

$$\lambda = \frac{\omega}{\omega_{\text{first}}} \tag{6}$$

and can be linked to the speed of the vehicle and to the irregularities of the road surface. In particular, the values of $q(t)$ are easily linked to the power spectrum IRI of the shape represented in Fig. 3 [15]-[16].

The constraints during optimisation represent the required road-holding ability and suspension working space. For the case studied, these constraints can be expressed as:

$$|q(t) - z_1(t)| - 0.051\ m \leq 0 \tag{7}$$

$$|z_2(t) - z_1(t)| - 0.125\ m \leq 0 \tag{8}$$

In particular, the equation (7) allows the road-holding ability or safety to be checked constraining tyre compression, while the equation (8) allows the suspension working space to be checked constraining the excursion of suspension (Figs. 4).

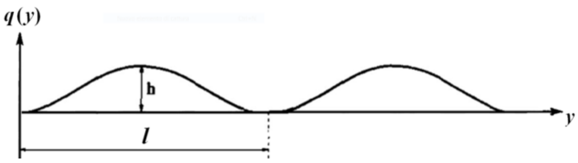


Fig. 2. Road surface profile

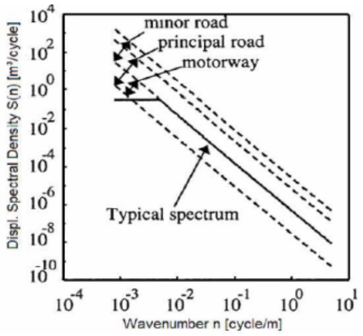


Fig. 3. Power spectrum for different road surface

III. Experimental Validation

The validation of the 3 DoF multibody model was performed by experimental data.

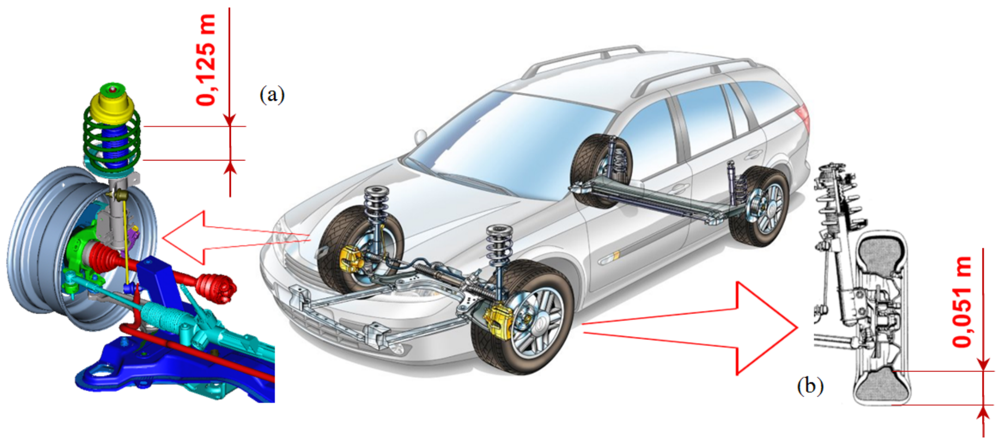
In particular the seat vertical acceleration was measured using 4 PCB® tri axial accelerometers, type ICP®, sensitivity 10 mV/g (+/- 500 g) by passing over a crossbar with a height h= 0.05 m and length l=0.05m (Figs. 5) at velocity of V_1= 27 km/h ± 1 km/h and at velocity of V_2= 40 km/h ± 2 km/h.

Inertia, sprung and unsprung mass data and design variables of the vehicle used for the test were reported in the first row of Table I.

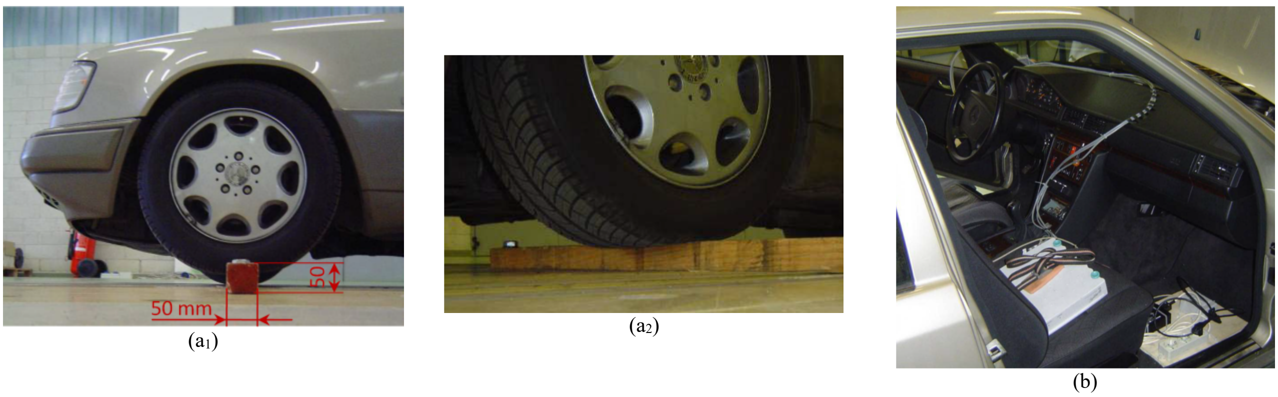
The values of height h= 0.05 m and length l=0.05 m produces the following forcing function:

$$q_{1,2}(t) = \frac{0.05}{2}\left(1 - \cos\left(\frac{2\,\pi\,V_{1,2}}{2.5 \times 0.05}t\right)\right) \quad (9)$$

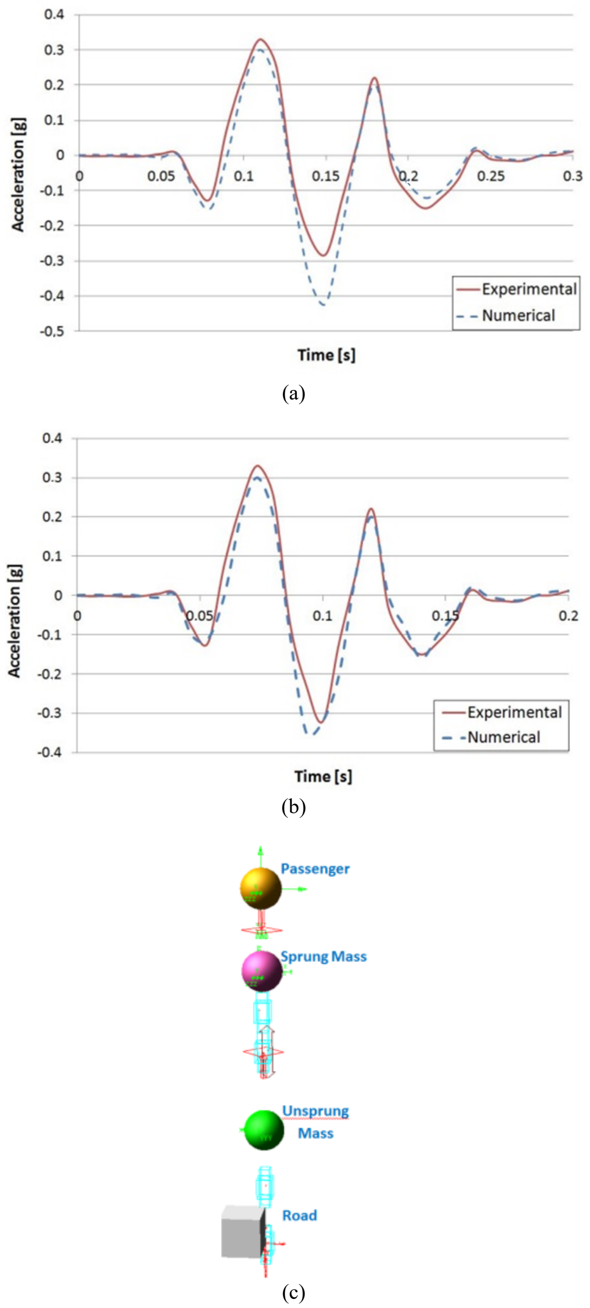
where 2.5 is the factor of correction for transforming the rectangular input into a sinusoidal shape; V_1= 27 km/h ± 1 km/h and V_2= 40 km/h ± 2 km/h. The seat vertical acceleration was experimentally measured using 4 PCB® tri axial accelerometers, type ICP®, sensitivity 10 mV/g (+/- 500 g). (Fig. 5(b)). In Fig. 6(a) and in Fig. 6(b) the experimental and numerical seat vertical acceleration was compared.



Figs. 4. Constraints: (a) suspension working space; (b) road-holding ability or safety



Figs. 5. (a1) Dimensions of transversal section of crossbar; (a2) crushing of the tire during crossbar crossing; (b) n°2 accelerometers on the front panel in cushion seat and n°2 accelerometers on platform for feet



Figs. 6. (a) Numerical and experimental seat acceleration measures in cross-beam at V1=27 km/h and (b) V2= 40 km/h; (c) 3 DoF multibody model (MSC ADAMS code)

The numerical-experimental comparison for V_1=27 km/h and V_2=40 km/h were carried out for a time windows of 0.3 s and 0.2 s respectively. The system is, in fact, considerably damped and these time windows are sufficient to evaluate the goodness of the numerical-experimental comparison.

For both crossbar crossing speeds only, a numerical central peak is higher of about 20% than the experimental. Overall, however, the standard deviation obtained by repeating the measurements and the mean value over the time windows analysed provide an error of less than 2%.

IV. Genetic Algorithm Optimization Procedure

The GA of Goldberg [17] and Haftka and Gurdal [18] was adopted to realize active tyre-suspension-seat system control.

The optimization procedure was based on the random choice of projects, having a uniform probability distribution, which constitute the so-called first generation.

More specifically, the single project was defined by means of a finite length binary string made up of smaller strings each representing a single project variable. Therefore, from the six strings of the design variables considered to obtain as many binary sites, 2^6 = 64 possibilities for each variable and $(2^6)^6 = 6.9\times10^{10}$ possible designs were achieved.

The first generation was then encoded in chromosomes to create a binary representation of a solution made up of the genes in the GA (components of decision variables).

A specific evaluation function is responsible for evaluating solutions according to their suitability. In this case, fitness is a numerical value that describes the probability of survival and reproduction of a solution (called genome). Through the "crossover" operator, capable of extending and enlarging only the surviving genes, a new population was created. However, in order to introduce a stochastic perturbation, the genetic operator "mutation" was used.

The new generation thus created will be subjected to the steps mentioned above until convergence criteria are met, such as running time or fitness.

Implementing the genetic algorithm for the optimal suspension design problem implies that the required precision is equal to the zero-decimal place for each design variable. Therefore, since the domain of the variable k_{sp} has length 30000, the precision requirement implies that the interval [10000; 30000] must be divided into at least 30000 intervals of equal size, making 15 bits necessary for the first part of the gene (30000 between 2^14 and 2^15). Similarly, since the domain of the variable c_{sp} is 7000 in length, the precision requirement imposes that the interval [1800; 2500] should be divided into at least 7000 intervals of equal size, requiring 13 bits in the second part of the gene. Likewise, 12 bits should be assigned to k_{se} and 11 bits should be assigned to c_{se}.

As only four design variables (k_{sp}, c_{sp}, k_{se} and c_{se}) are used to minimize the objective function, as shown in Table I, 51 bits in all chromosomes are initialized randomly. During the evaluation phase, the decoding of each chromosome takes place. Some chromosomes are better in terms of reducing the value of the objective function than others. Therefore, to these chromosomes a greater weight is assigned, which equates to a greater opportunity to be involved in the production of the next generation. The crossover is carried out by exchanging the same segment of two chromosomes (in this case the last 10 bits), while the mutation is performed by inverting a bit in a chromosome (from 0 to 1 or vice versa). Once the new generation is obtained, the repetition of the previous steps is stopped when the convergence criterion is met. Since the iteration leads to lower values of the objective function, the best chromosome will correspond to the minimum objective function.

The minimum objective function was weighted according to the values of Table II (World Health Organization) to consider the time exposure limits.

TABLE II
VIBRATION EXPOSURE TABLE

Vibration magnitude m/s²	Daily exposure time 15m	30m	1h	2h	3h	4h	5h	6h	8h
30	450	900							
25	315	625	1250						
20	200	400	800						
19	180	360	720	1450					
18	160	325	650	1300					
17	145	290	580	1150					
16	130	255	510	1000					
15	115	225	450	900	1350				
14	98	195	390	785	1200				
13	85	170	340	675	1000	1350			
12	72	145	290	575	865	1150	1450		
11	61	120	240	485	725	970	1200	1450	
10	50	100	200	400	600	800	1000	1200	
9	41	81	160	325	485	650	810	970	1300
8	32	64	130	255	385	510	640	770	1000
7	25	49	98	195	295	390	490	590	785
6	18	36	72	145	215	290	360	430	575
5,5	15	30	61	120	180	240	305	365	485
5	13	25	50	100	150	200	250	300	400
4,5	10	20	41	81	120	160	205	245	325
4	8	16	32	64	96	130	160	190	255
3,5	6	12	25	49	74	98	125	145	195
3	5	9	18	36	54	72	90	110	145
2,5	3	6	13	25	38	50	63	75	100
2	2	4	8	16	24	32	40	48	64
1,5	1	2	5	9	14	18	23	27	36
1	1	1	2	4	6	8	10	12	16

red   yellow   green

In this way the time interval analysed (T) becomes a new design variable in GA optimization procedure. The three colors (green, yellow and red) shown in Table II refer to the three main areas of the exposure limits. The first (green) at low risk, the second (yellow) at medium risk, the third (red) at high risk.

In a preliminary phase, it seems to be convenient to fix the number of generations and the size of the population, making changes to the mutation rate and the crossover rate. By imposing a mutation rate of 1%, good convergence to the minimum value of the objective function was found for a variety of crossover rates. Furthermore, a population size of 100 and a crossover rate of 25% seem to ensure better performance in terms of minimizing the objective function.

With reference to the results of the preliminary analysis, 1%, 25% and 100% identify the appropriate mutation rate, crossover rate and population size, respectively. Moreover, the only quantity that changes in this context is constituted by the number of generations, which will be calculated by means of the convergence criterion during the iteration. Fig. 7 illustrates the average value of the objective function at each generation. The results of four simulation runs are so defined.

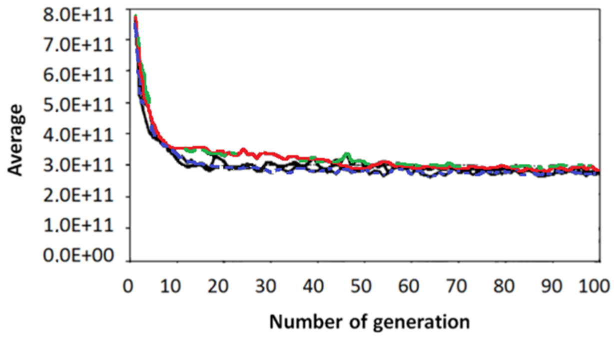


Fig. 7. Average values for one hundred populations

It is through the average value of the population that the convergence is estimated. The design variables corresponding to the optimum value of the objective function are: k_{sp} = 34 kN/m, c_{sp} = 2010 Ns/m, k_{se} = 3 kN/m and c_{se} = 1000 Ns/m respectively.

V. Results and Discussion

In Table III it is possible to compare the values of the design variables for four independent runs of the GA. In addition, we can find in this table the peak absolute acceleration value of the seat for the vehicle (bold value), subjected to road profile.

TABLE III
RESULT FOR ACTIVE SUSPENSION DESIGN

Design variable	*Run 1*	*Run 2*	*Run 3*	*Run 4*
msp [kg]	250	250	250	250
kt [kN/m]	100	100	100	100
ct [Ns/m]	800	800	800	800
ksp [kN/m]	32	34	35	30
csp [Ns/m]	2150	2100	2200	2300
kse [kN/m]	4	3	3,5	4,5
cse [Ns/m]	1000	1000	1100	1200
$max\lvert\ddot{q}(t)\rvert$ [m/s²]	4.5	3.8	4.2	4.4

It derives that the seat suspension values, k_{se} and c_{se}, are consistently at or near their lower bounds, forming a ‘soft suspension’, with the forces applied to the seat and its acceleration kept low. The best design, whose measurement was defined by the lowest objective function value, was obtained from run 2.

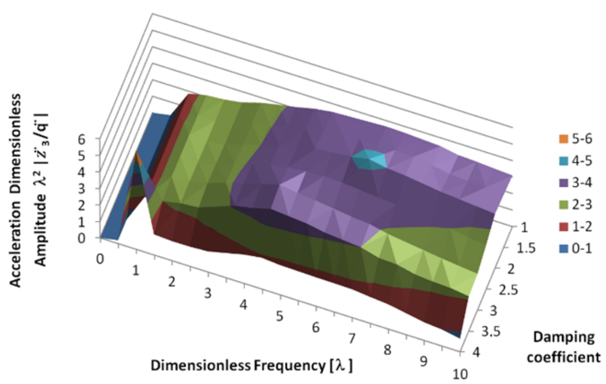


Fig. 8. Acceleration amplitude Vs frequency and damping coefficient

Fig. 8 indicates the optimized vertical acceleration value (response gain) on the seat subjected to variable frequency and suspension damping coefficient. Fig. 9 outlines optimum damping values for shock absorber obtained from the optimization process.

It is easy to determine that the obtained values, evaluated in terms of RMS, apply to the second zone of the diagram "*confort reaction to vibration*" established by ISO 2631- (1997) (Fig. 10).

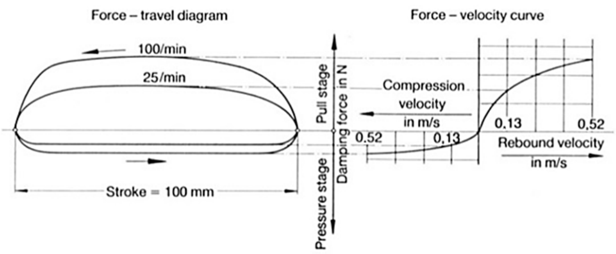


Fig. 9. Optimum damping values for shock absorber

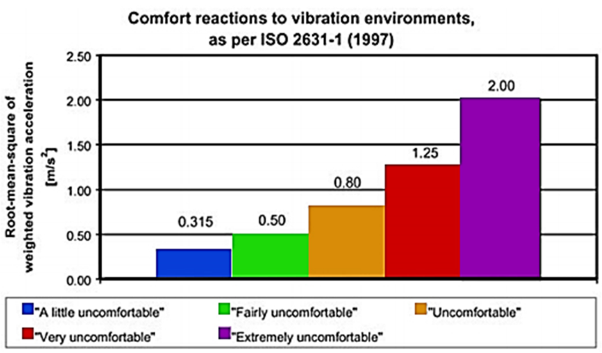


Fig. 10. Comfort reaction to vibration

VI. Conclusion and Future Works

This paper proposes the development of an active suspension system for passenger cars, applying genetic algorithm control to a 3 DoF multibody model. The goal was achieved by optimizing the best vertical acceleration on the seat for different road profiles in the entire possible frequency range considering also time exposure limits outlined by unification agencies and road authorities. Road irregularities have been determined by means of the International Roughness Index (IRI), whereas the acceleration values obtained were compared to the "comfort reaction to vibration" values established by ISO 2631 (1997). The results proved that the studied active suspension system leads to a measurable improvement for vehicle passenger comfort conditions. Moreover, it has been observed that the methods applied lead to stable and efficient optimization procedures which make them suitable to be used as design tools during the development of new vehicle concepts.

The developed method lends itself to further interesting developments. Among the most interesting was the possibility of implementing into the multibody vehicle model also different boundary conditions: various vehicle velocity, different road profiles, vehicle trajectories, deformability of the frame. Moreover, also a more complex multibody model could be implemented in future works, considering that in recent years there was a significant development of numerical multibody human models able to replicate different morphologies.

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