Modeling of Passenger Ride-Comfort Enhancement through Designing Seat Cushion with Scilab-Xcos

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***Abstract*—This paper demonstrates a modeling of vehicle ride comfort using 11 degrees of freedom (DoF) quarter-car approximation with a free and open-source Scilab-Xcos software. The simulation is aimed to enhance the ride quality of passengers by optimizing parameters of the seat cushion in the seat structure of the vehicle. From the simulation results, the seat cushion plays an important role in improving the ride comfort of the passenger. A significantly lower vertical displacement and acceleration were observed on the occupant's head relative to the model without seat-cushion. Moreover, the ride comfort can be further increased by optimizing the parameters in the seat cushion. The lowest vibration peaks were attained from the combination of the stiffness of 16 N/m and damping of 25 Ns/m. However, variations of the mass of the seat cushion are found to give insignificant effect on the comfort improvement.**

***Index Terms*—ride quality modeling, seat cushion, 11 DoF quarter car, human model, Scilab-Xcos.**

I. INTRODUCTION

Bumpiness of the road surface is the main cause of shock and vibrations in vehicles while driving. Controlling the vibration from the road irregularity is crucial to attaining the safety and comfort of the passenger. As the wheels stepped on obstacles, the efficiency of drive transmission, braking and steering systems will decrease as tires vertically apart from the road. This affects the ride handling and safety. Furthermore, the vibrations from the road are also transmitted to the tire and then to the vehicle site structure, which eventually arrives in the passenger seat. The vibrations sensed by the occupant defines ride quality of the vehicle. Despite the comfort issue, long periods of exposure to vibration at certain frequencies can be harmful to the health of the occupants. Thus, decreasing amplitude of the whole body vibration below the human tolerance is highly important, as agreed in the International standard ISO 2631 [1, 2].

The vibration dynamics can be reduced by imposing a hard suspension, but mostly it sacrifices the comfort of the passenger. In fact, the vibration control is not intended to eliminate the entire vibration, but only the unwanted vibrations so that the unpleasant effects are kept within acceptable limits [3]. Conventionally, ways to properly isolate the unpleasant vibration inputs are by modifying the suspension system, either by adopting adaptive-passive, semi-active or fully active controls [4, 5]. However, apart from the suspension designs, the comfort of the passenger can be enhanced further by involving the other components, such as chassis structure, seat structure, seat cushion, and isolators. Particularly, the seat cushion is an important component to be considered in regard to the passenger senses [6]. The cushion may determine the overall vehicle ride quality since this is the closest component in contact with the occupant [7]. The ability of the seat cushion to isolate the road disturbances will improve the ride quality. This enhancement can be achieved by optimizing the stiffness and damping parameters of the seat cushion.

In this paper, we demonstrate a simple analysis to improve ride quality through parameters optimization on a seat-cushion using a free and open-source numerical platform by Scilab-Xcos software. The passenger comfort is evaluated from the peaks of displacement and acceleration on the occupant's head against the change of seat cushion parameters.

II. MATHEMATICAL MODEL AND SIMULATIONS

The dynamics of the human in the vehicle were modeled by 11 degrees of freedom (DoF) of lumped mass-spring-damper systems included 7 DoF for the human body and 4 DoF of a quarter car for the seat structure and suspension system

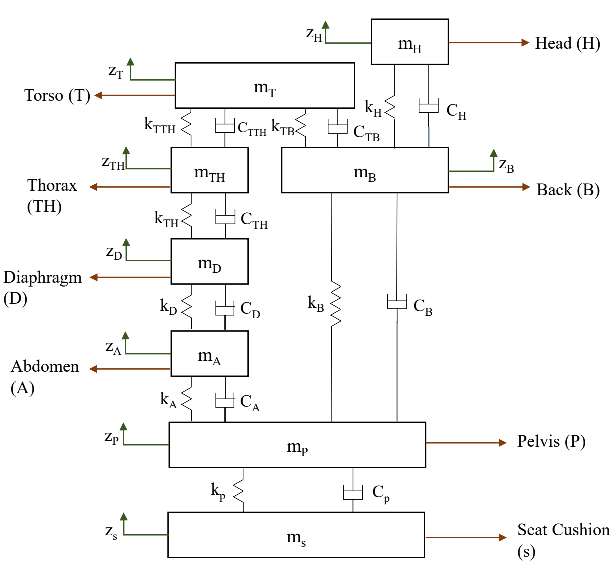

Fig. 1. 7 DoF of human model including 1 DoF of seat cushion

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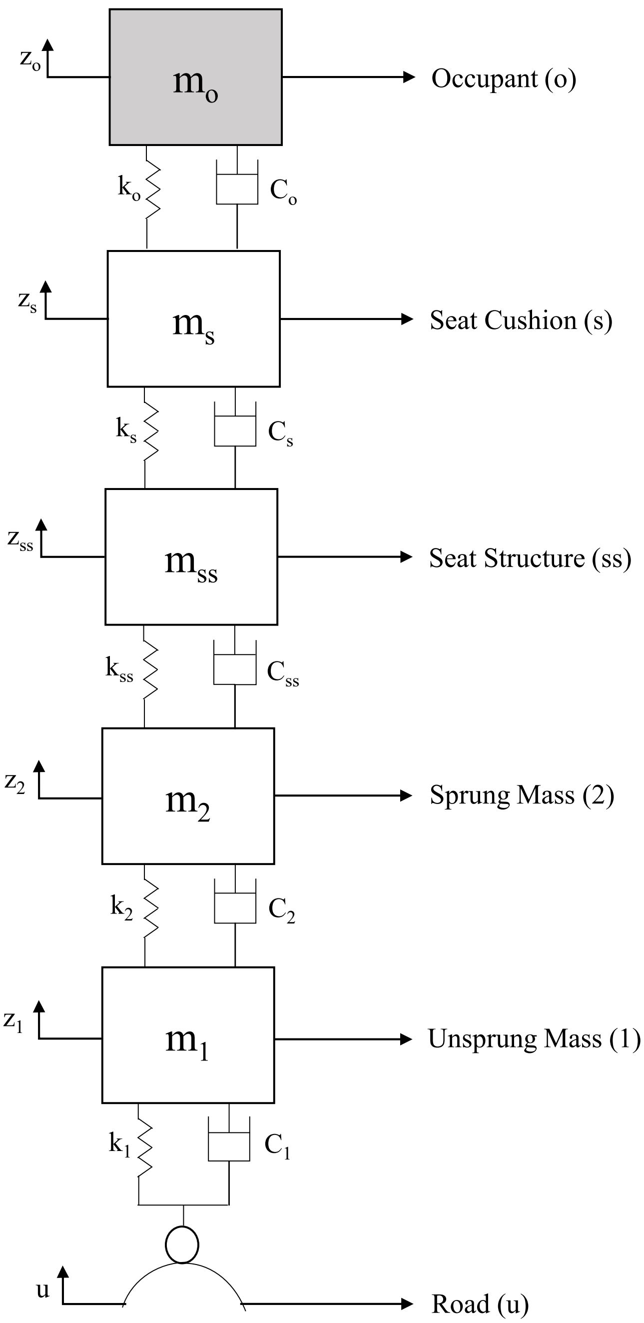

Fig. 2. Quarter-car model with seat and occupant

as depicted in Fig. 1 and 2, respectively. For the sake of simplification, the lumped parameters in the model are all assumed to be linear [8].

The dynamic structure of human body is very complex and sophisticated. Fortunately, as shown in many literature recently, human models can be well-developed based on lumped-parameter models [6, 7, 9]. In this work, the human body is modeled using 7 DoF of spring-damper-mass systems aligned in series and/or parallel as depicted in Fig. 1. The model structure follows the work of Banerjee *et.al.* [7, 8]. As shown in the figure, parts of the human body consist of pelvis (P), abdomen (A), diaphragm (D), thorax (TH),

TABLE I
HUMAN MODEL PARAMETERS

Model Parameters	Mass m (kg)	Stiffness k (N/m)	Damping Coefficient C (Ns/m)
Head (H)	5.45	52600	3580
Back (B)	6.82	52600	3580
Torso-Thorax (TTH)	32.762	877	3580
Torso-Back (TB)	-	52600	3580
Thorax (TH)	1.362	877	292
Diaphragm (D)	0.455	877	292
Abdomen (A)	5.921	877	292
Pelvis (P)	27.3	23500	371

torso (T), back (B), and a human head (H). By employing Newton's law of motion and linear Hooke-like spring and damper expressions for each lumped parameter [10, 11], the body components can be expressed in the following equation of motions,

$$
\begin{aligned}
\ddot{z}_P &= \frac{1}{m_P}(Fe_P + Fd_P - Fe_A - Fd_A - Fe_B - Fd_B),\\
\ddot{z}_B &= \frac{1}{m_B}(Fe_B + Fd_B - Fe_H - Fd_H\\
&\quad -Fe_{T2} - Fd_{T2}),\\
\ddot{z}_A &= \frac{1}{m_A}(Fe_A + Fd_A - Fe_D - Fd_D),\\
\ddot{z}_D &= \frac{1}{m_D}(Fe_D + Fd_D - Fe_{TH} - Fd_{TH}),\\
\ddot{z}_{TH} &= \frac{1}{m_{TH}}(Fe_{TH} + Fd_{TH} - Fe_{T1} - Fd_{T1}),\\
\ddot{z}_T &= \frac{1}{m_T}(Fe_{T1} + Fd_{T1} + Fe_{T2} + Fd_{T2}),\\
\ddot{z}_H &= \frac{1}{m_H}(Fe_H + Fd_H).
\end{aligned}
\tag{1}
$$

where $\ddot{z} = \frac{d^2z}{dt^2}$, $\dot{z} = \frac{dz}{dt}$, and z stand for the vertical acceleration, velocity, and displacement of each body, respectively. The

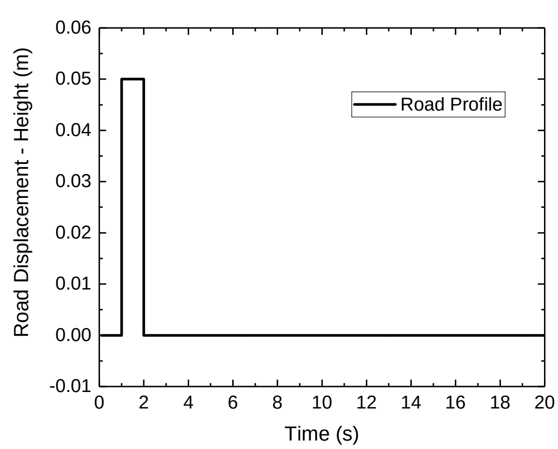

Fig. 3. Step function in respect to the time evolution as the input road profile for the simulation

TABLE II
QUARTER-CAR MODEL PARAMETERS

Model Parameters	Mass m (kg)	Stiffness k (N/m)	Damping Coefficient C(Ns/m)
Unsprung mass(1)	60	190000	1000
Sprung mass(2)	1500	1700	1100
Seat Structure(ss)	15	1200	200
Seat Cushion(s)	2	16	50

masses, stiffness, and damping coefficients in each component are provided in Table I, which were obtained from [7].

Instead of considering a full-car vehicle suspension system as proposed in [7], a quarter-car model is proposed in this work. The quarter-car system provides much more simple but rather focus analysis to optimize the seat-cushion parameters. In this way, a simple analysis to produce an optimized seat-cushion parameter for a better ride quality can be achieved. The quarter-car model contains of a 4 DoF system included of the input disturbance from the road (u), unsprung mass (m_2), sprung mass (m_1), seat structure ss, and seat cushion (s), as shown in Fig. 2. Following the above approach, the equation of motions of the 4 DoF suspension system can be expressed as follow,

$$\begin{aligned}
\ddot{z}_2 &= \frac{1}{m_2}(Fe_2 + Fd_2 - Fe_1 - Fd_1), \\
\ddot{z}_1 &= \frac{1}{m_1}(Fe_1 + Fd_1 - Fe_{ss} - Fd_{ss}), \\
\ddot{z}_{ss} &= \frac{1}{m_{ss}}(Fe_{ss} + Fd_{ss} - Fe_s - Fd_s), \\
\ddot{z}_s &= \frac{1}{m_s}(Fe_s + Fd_s - Fe_P - Fd_P),
\end{aligned} \tag{2}$$

The lumped parameters in each component are provided in Table II.

All the forces involved in Eqs. 1 and 2 are defined as,

$$\begin{aligned}
Fe_1 &= k_1(u - z_1), \\
Fd_1 &= C_1(\dot{u} - \dot{z}_1), \\
Fe_2 &= k_2(z_1 - z_2), \\
Fd_2 &= C_2(\dot{z}_1 - \dot{z}_2), \\
Fe_{ss} &= k_{ss}(z_2 - z_{ss}), \\
Fd_{ss} &= C_{ss}(\dot{z}_2 - \dot{z}_{ss}), \\
Fe_s &= k_s(z_{ss} - z_s), \\
Fd_s &= C_s(\dot{z}_{ss} - \dot{z}_s), \\
Fe_P &= k_P(z_s - z_P), \\
Fd_P &= C_P(\dot{z}_s - \dot{z}_P), \\
Fe_A &= k_A(z_P - z_A), \\
Fd_A &= C_A(\dot{z}_P - \dot{z}_A), \\
Fe_D &= k_D(z_A - z_D), \\
Fd_D &= C_D(\dot{z}_A - \dot{z}_D), \\
Fe_{TH} &= k_{TH}(z_D - z_{TH}), \\
Fd_{TH} &= C_{TH}(\dot{z}_D - \dot{z}_{TH}), \\
Fe_{T1} &= k_{TTH}(z_{TH} - z_T), \\
Fd_{T1} &= C_{TTH}(\dot{z}_{TH} - \dot{z}_T), \\
Fe_{T2} &= k_{TB}(z_B - z_T), \\
Fd_{T2} &= C_{TB}(\dot{z}_B - \dot{z}_T), \\
Fe_B &= k_B(z_P - z_B), \\
Fd_B &= C_B(\dot{z}_P - \dot{z}_B), \\
Fe_H &= k_H(z_B - z_H), \\
Fd_H &= C_H(\dot{z}_B - \dot{z}_H).
\end{aligned} \tag{3}$$

In the simulation, the vehicle is imagined to travel with a constant horizontal speed over a road profile with a bump, as shown in Fig. 3. The profile was correlated to a typical real-life road conditions with a 0.05 m of bump height [7]. The bump width is represented in terms of a time length of a step input signal. The tire encounters the bump at one second. This value was chosen to count for 1 Hz of frequency of road excitation. The simulation was performed by using a block-diagram method under Xcos module in Scilab software. Runge-Kutta 4(5) solver was used for all the calculations.

In accordance with Eqs. 1, 2, and 3, the dynamics can be calculated using four kinds of Xcos block, as shown in Fig. 4 and 5. Firstly, unsprung, sprung, seat structure, seat cushion, abdomen, diaphragm, and thorax can be obtained by one form of Xcos block. Those components have the same equations because each mass has two contacts with other

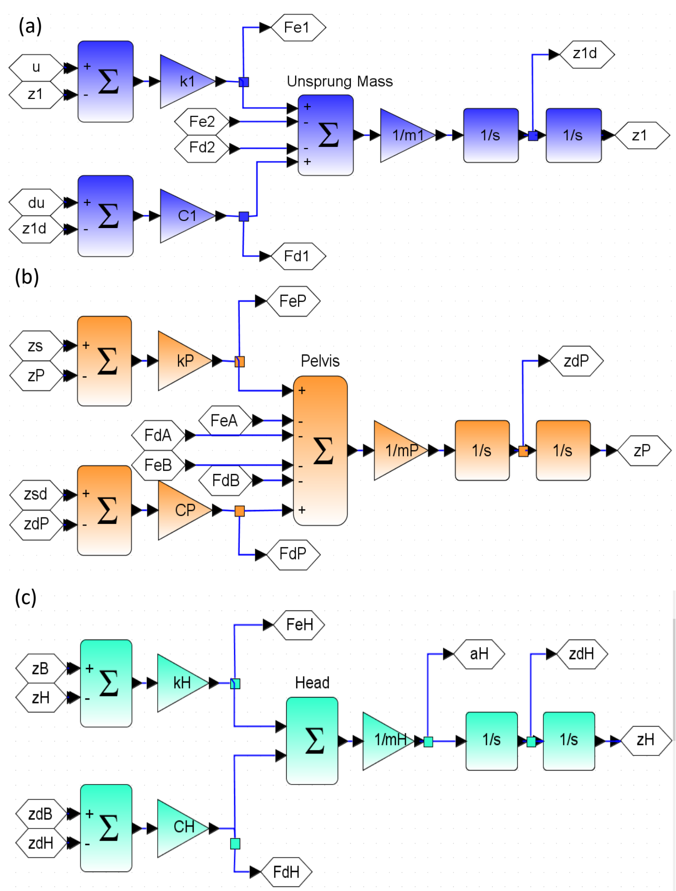

Fig. 4. Xcos-Scilab block diagram for (a) unsprung mass, (b) pelvis, and (c) head modeling

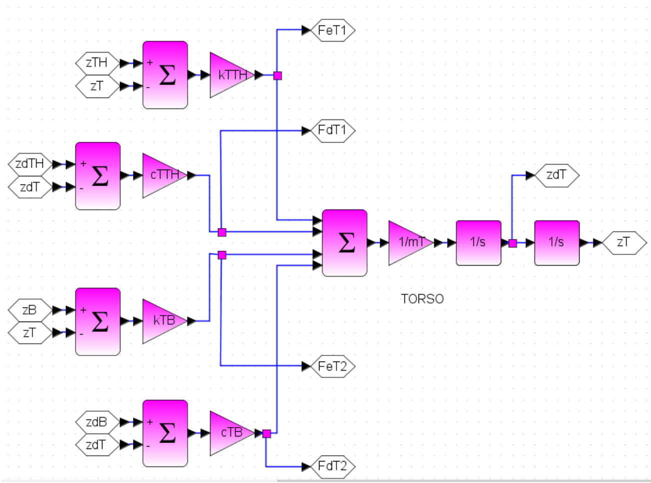

Fig. 5. Xcos-Scilab block diagram for torso modeling

components. For example, unsprung mass has two contacts, *i.e.* with sprung mass and road input. The difference between them is only on the input parameters. Secondly, pelvis and back components have a different mathematical equation. Both pelvis and back have contacts with 3 other masses. Pelvis has a contact with the back, abdomen, and seat cushion, while back has a contact with the torso, pelvis, and head. Furthermore, the torso model has a different block formation compare to the other, as shown in Fig. 5. Both torso and head are on the top of the structure, which does not have any further contact. The difference between both is that the torso has a contact with 2 other masses, while head has only one contact with the back.

III. Results and Discussion

Fig. 6 shows the vertical displacement and acceleration of the whole body of the vehicle caused by the bump from the road conditions (see Fig. 3). The investigated vertical response includes the sprung mass, seat structure, seat cushion, and head. As shown in the figure, the input signal applied to the bottom of the vehicle is transferred to the passenger's head. This confirms that 1 Hz frequency of the road excitation affects the passenger comfort. Nevertheless, lower disturbance amplitudes in both acceleration and displacement are observed on the head in comparison to the vibration on the sprung component. It may be related to the structure between the tire and the occupant's head. As characterized from the figure, both vertical displacement and acceleration on the head are similar with the seat cushion. This result is in agreement with the previously reported works [7, 9]. This implies that the seat cushion plays a significant role to reduce the vertical disturbances from the bottom at this frequency. A further

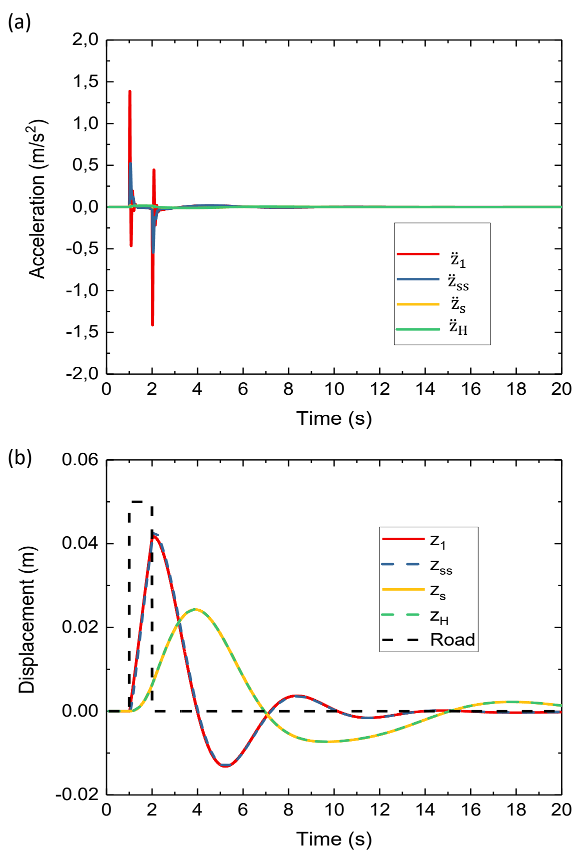

Fig. 6. Vertical (a) acceleration and (b) displacement on sprung mass (m_1), seat structure (ss), seat cushion (s), and head (H)

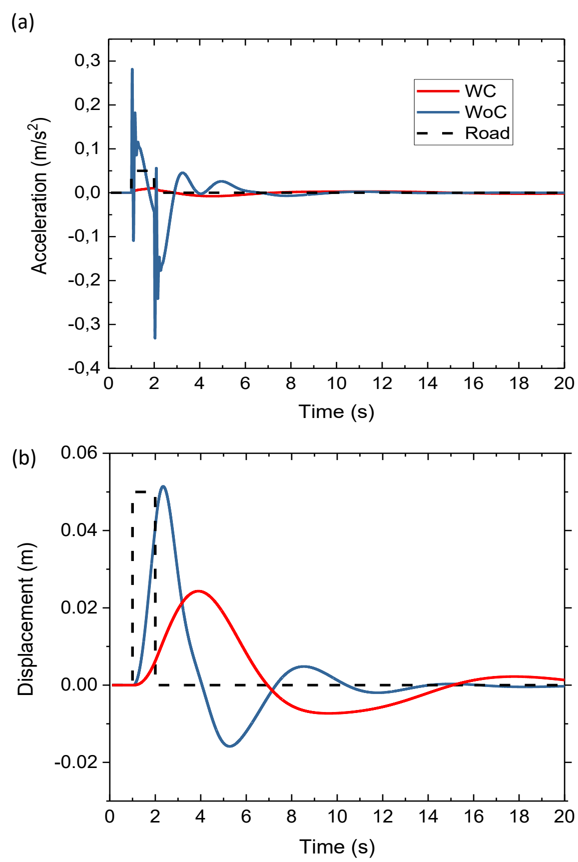

Fig. 7. Vertical (a) acceleration and (b) displacement of occupant's head with (WC) and without (WoC) a seat cushion

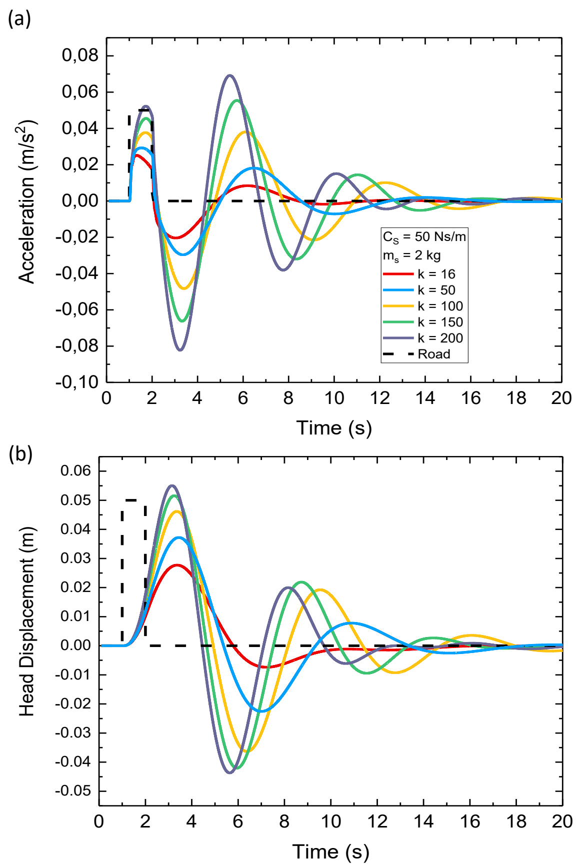

Fig. 8. Vertical (a) acceleration and (b) displacement of occupant's head with the seat-cushion stiffness coefficient variation

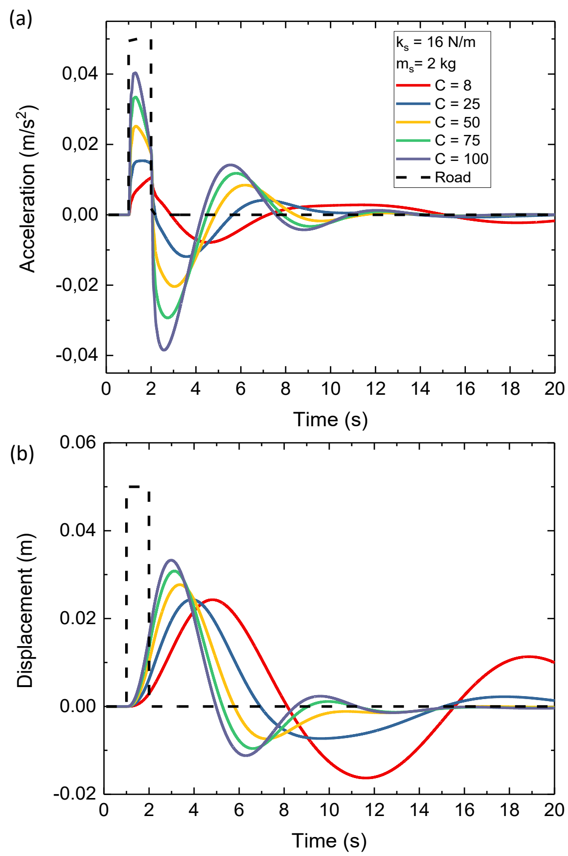

Fig. 9. Vertical (a) acceleration and (b) displacement of occupant's head with the seat-cushion damping coefficient variation

analysis of the seat cushion becomes more important to improve the passenger ride comfort.

To elaborate the importance of the seat cushion, observations have been made on the vertical displacement and acceleration at the passenger's head with and without a seat cushion. Fig. 7 shows the response on the occupant's head with and without a seat cushion. From the figure, the vibration peaks on the head of the model with a seat cushion are much smaller than without seat cushion. The largest peak reduces about 3 cm by introducing the seat cushion on the seat structure. Moreover, the frequency of the oscillation is also lower when the seat cushion is applied, which increases the ride comfort of the occupant [12].

The ride comfort can be enhanced further by optimizing the seat-cushion parameters. Seat cushion parameters, such as stiffness, damping coefficient, and mass, could have a high impact to reduce the vertical displacement and acceleration. The optimization involves three parts of the simulation, *i.e.* a variation of stiffness parameters with constant mass and damping coefficients, damping coefficient variations with constant mass and stiffness, and a variation of seat-cushion mass. The optimizations are based on the minimum peaks of vertical displacement and acceleration observed on the passengers head.

The dynamic response on the occupant's head for the stiffness variation is plotted in Fig. 8. The damping coefficient and the mass of the seat cushion are assumed to be 50 Ns/m and 2 kg, respectively. The others parameters are the same with the parameters provided in Tabel I. As shown in the figure, the amplitude of vibrations is found to reduce with the decreases of stiffness. The lowest reduction in both displacement and acceleration is observed in the seat-cushion as stiffness value set to 16 N/m. Lower peaks of displacement of 0.23 m and acceleration of 0,24 m/s^2 are achieved using this parameter. These values are twice lower than the used of 200 N/m of stiffness value.

The optimized stiffness coefficient of 16 N/m and mass of 2 kg were used as the seat-cushion parameters for the second observation with the damper variations. Similar to the case of stiffness, the vibration peaks on the occupant's head are reduced along with the decreases of damping parameter. The results are shown in Fig. 9. However, in contrast to the stiffness case, the reduction of the displacement peak is saturated on 25 Ns/m of damping value. There are no further decreases observed in the displacement peak. A further reduction in damping constant to 8 Ns/m only decreases the frequency of the vibration. Although the 8 Ns/m of damping value gives a lower peak in the acceleration counterpart, but a larger secondary peak of displacement is observed in this parameter. This peak induces more vibration on the occupant's head, which deteriorates the comfort. Thus, the 25 Ns/m offers the

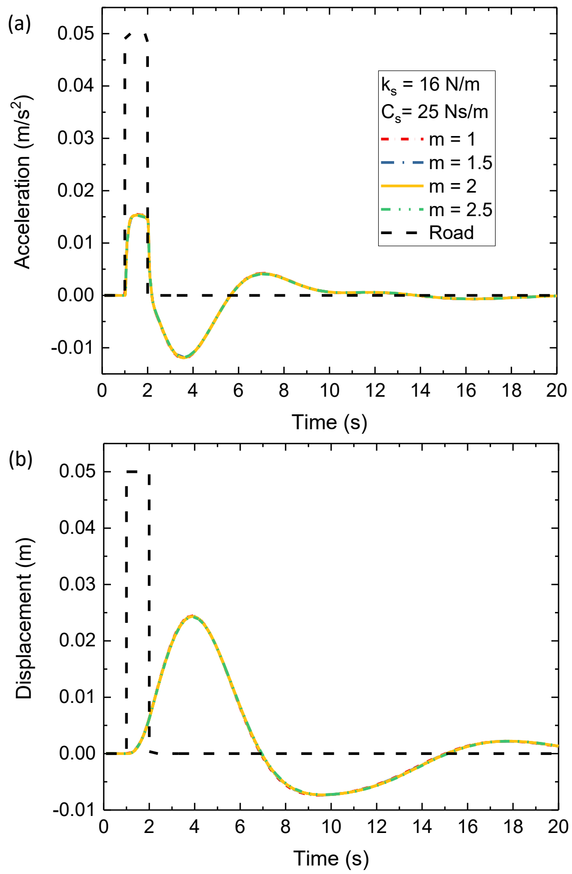

Fig. 10. Vertical (a) acceleration and (b) displacement of head with the seat-cushion mass variation

optimized condition to enhance the ride comfort, showing a better performance in reducing the vertical displacement.

The variation of the mass value in the seat cushion is the third factor we need to consider. The optimized coefficients of 16 N/m of stiffness and 25 Ns/m of damping were used as the constant parameters. The mass parameters were varied from 1 kg to 2.5 kg with 0.5 kg as the step interval. The result is shown in Fig.10. From the figure, the vibration peaks are found to be the same in both acceleration and displacement for all the mass changing. This observation implies that the mass variation has no significant impact in changing both the vertical displacement and acceleration on the occupant's head at 1 Hz of road-input frequencies.

IV. Conclusion

A modeling of 11 DoF quarter-car model was performed under a free Scilab-Xcos program. The main aim of the simulation is to enhance the occupant ride comfort through optimization of seat-cushion parameters. The peaks of vertical displacement and acceleration of the passengers head were used to characterize the ride comfort. Models with and without a seat cushion were compared to see the impact of the seat cushion. As a result, the presence of a seat cushion in the model can significantly reduce the peaks of vertical displacement and acceleration on the passenger's head. The stiffness and damping coefficients on the seat cushion show a pronounced effect to further enhance the driving comfort. However, the mass of the seat cushion does not give a significant improvement to reduce the vertical displacement and acceleration. From the optimization results, the best stiffness parameter is 16 N/m and the best damping coefficient is 25 Ns/m, which can give the lowest peaks of vibration in the head.

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