Dynamic Behavior and Force Analysis of the Full Vehicle Model using Newmark Average Acceleration Method

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Abstract—In this study, the dynamic interaction between road and vehicle is modeled. For this purpose, a full vehicle model with eight degrees of freedom is considered. The equations of motion of the whole system are derived by the Newmark average acceleration method. Due to varying road roughness, the forces affecting the driver and the vehicle-components are analyzed in detail. Also, vertical and rotational displacements, velocities, and accelerations are examined, and results graphs are given. Two different pre-defined road profiles, created as non-random road excitation, and five different vehicle speeds are presented and analyzed.

Keywords-vehicle road interaction; full-vehicle model; passenger comfort; dynamic forces on vehicle components

I. INTRODUCTION

Vehicle dynamics and forces affecting vehicle components and driving comfort with increasing vehicle speed are an emerging research topic. Many vehicle models have been proposed to examine vehicle-road interaction. Generally, these models can be classified in three types, quarter-car, half-car, and full-car modeling. Using Taguchi L16 array and SNR analysis, measurements for the performance of the semi-active suspension system with MR damper were taken in [1] and according to the results, damper cylinder material is the key parameter to the design of the magnetorheological (MR) damper. By using speed bump as road disturbance, overshoot and settling time with passive suspension system were analyzed in [2] for a quarter-car model with 2 degrees of freedom. Adding an inerter on the suspension system, the performance of vibration absorbing was investigated in [3] and the vibration was considerably decreased. A robust quarter-car control scheme was created in [4] along with a road disturbance profile with a sliding mode controller. According to the variable damping coefficient limit, semi-active suspension systems were created in Matlab/Simulink in [5] and system’s damping coefficient limit of 4000Ns/m was performed best when considering ride comfort. Theoretical and experimental analyses of rail vehicles were modeled in [6] using electro-mechanics similarity theory. A quarter-car model with a PID controller used to minimize the vertical body acceleration was prepared in [7]. A method was developed in [8] for the performance optimization of an eCAR considering the impact of road dynamics, acceleration rate, mass changing and gear ratio. According to the results, the proposed method was efficient and simple and could be applied to any eCAR model. An active suspension system which used sliding mode control was created and compared with the passive suspension system in [9]. The results show that this system has better effect on vibration isolation compared to the passive suspension system.

A Grey Fuzzy Sliding Mode controller was proposed in [10] for improving ride comfort. The results show that this controller provides robustness to the system under the presence of uncertainties. Under three different road roughness classes classified by the ISO-8608 standard, Newmark Beta method was used in [11] for quarter-car analysis. Under step road roughness, the suspension system of a quarter-car model was analyzed in [12]. The results show that ride comfort was not suitable according to the ISO2631-1 standard. For half-car suspension system analysis, ADAMS/CAR program was used and B class road and pulse input were utilized as road disturbance in [13]. A nonlinear half-car model was created in [14] and for reducing the vibration levels PID control, Fuzzy Logic Control (FLC), Hybrid Fuzzy-PID control (HFPPID), and Hybrid Fuzzy Logic controller with Coupled Rules (HFPIIDCR) have been used. According to the simulation results the HFPIIDCR shows good performance in reducing the vibration levels under nonlinear system parameters. A half-car model with road adaptive nonlinear control was created in [15] with road adaptive algorithm schemes, showing adaptation potential for different road types. Active and passive suspension systems of a half-car model were analyzed in [16] using nonlinear suspension stiffness and damping. The results showed that the nonlinear active suspension system performed better when compared with the passive one. A full-car model with seven degrees of freedom with nonlinear suspension springs and dampers under sinusoidal road disturbances was studied in [17] and according to the results the car response...
could be chaotic. The suspension system of a full-car model was examined in [18] using three different road assumptions, while a full-car model was analyzing in [19] for passive and active suspension systems under a sinusoidal road excitation with optimization technique.

Using a bump road excitation as a road roughness model, a semi active control algorithm was used in [20] for a full-car model with an MR damper. The results show that this control algorithm decreases displacements and accelerations. For optimization of ride comfort and handling, a semi-active control was created in [21] using a full-car model. According to the results, this algorithm achieved the intended goals. A quasi-LPV approach was studied in [22] as a semi-active controller for seven degrees of freedom of a full-car model. For a semi-active full-car model, LQ controller and observer were compared in [23] with a real vehicle with a skyhook controller. An H∞ observer was designed in [24] to decrease the effects of unknown ground disturbances on the full-car model with seven degrees of freedom. Ride comfort and frequency response up to 18Hz of a full-car model with ten degrees of freedom were examined in [25]. For road profile irregularities estimation a new technique called Independent Component Analysis (ICA) was proposed in [26]. The obtained results show that the proposed technique is adequate for identifying road disturbances. Optimization of the passive suspension system was examined in [27], and for this purpose, a numerical-computational program was developed. Driver seat vertical acceleration was reduced approximately by 21.14% of weighted RMS value. A robust finite-frequency H∞ controller was developed and studied in [28]. The fundamentals of full-car dynamics and the vibration influence to the human body in order to design better active suspension systems for suitable comfort level were investigated in [29]. A full-car model with seven degrees of freedom and an MR damper was analyzed in [30] for suspension system control. The control algorithm included optimal control algorithm and Fuzzy Logic, Linear Quadratic Regulator (LQR) and Fuzzy controller.

In this study an accurate modeling of vehicle road interaction using a full-car model which can be used for the investigation of vertical, pitch, and roll movements of the car body, driver’s seat and other suspension components, is presented. In addition, as an issue which has not been discussed in detail in the literature, the analysis of the forces applied to both the vehicle body and the driver's seat was also conducted. Newmark average acceleration method was used as a different method than the ones generally used for these studies. In addition, in order to examine the effects of vehicle speed on vehicle dynamics, analysis was performed at 5 different vehicle speeds. For this purpose, analysis was performed for two different proposed road roughness models.

II. MATHEMATICAL MODELING

To study vehicle dynamics and dynamic forces affecting vehicle components, the vehicle could be classified in one of the three different models which are (from the least to the most complex): quarter-car, half-car, and full-car. In this study the full-car model was used because it is the nearest to real-life vehicles (Figure 1). In this model, roll and pitch rotation movements can be examined along with vertical movements and forces.

The full-car model has eight degrees of freedom and includes passenger mass \( m_p \), car body mass \( m_s \), front left wheel mass \( m_{f1w} \), rear left wheel mass \( m_{r1w} \), front right wheel mass \( m_{f2w} \), rear right wheel mass \( m_{r2w} \), moment of inertia of pitch \( I_p \), moment of inertia of roll \( I_r \), and connecting elements which comprise of dampers and springs with the same properties and linear characteristics. In this system, springs are named as: passenger seat stiffness \( k_p \), front left suspension stiffness \( k_{f1} \), rear left suspension stiffness \( k_{r1} \), front right suspension stiffness \( k_{f2} \), rear right suspension stiffness \( k_{r2} \), front left wheel stiffness
represent the widths of the road defects. In the second model

\( f - M - f - \) equations are given below:

\[
\begin{align*}
\mathbf{m}_{x}\ddot{z}_p + c_p (\dot{z}_p - \dot{z}_f + (L_f - L_{psf})\theta) + (L_{psf} - L_i)\theta) + k_p (z_p - z_f) + (L_f - L_{psf})\theta + (L_{psf} - L_i)\phi = 0 \\
\mathbf{m}_{z}\ddot{x}_p - c_p (\dot{z}_p - \dot{z}_f + (L_f - L_{psf})\theta + (L_{psf} - L_i)\theta) + k_p (z_p - z_f) + (L_f - L_{psf})\theta) + (L_{psf} - L_i)\phi = 0
\end{align*}
\]

\[ l_0^2 \ddot{z}_p [c_p (L_{psf} - L_f)] - 2 \theta [(c_f + c_r) l_f - (c_f + c_r) l_R + c_p (L_f - L_{psf})] - \theta [(c_f + c_r) l_f - (c_f + c_r) l_R + c_p (L_f - L_{psf})] - \theta l_f [k_f + k_r] l_R - k_p (L_f - L_{psf}) = 0 \]

\[ l_0 \ddot{\phi} - \dot{z}_p [c_p (L_{psf} - L_f)] = 2 \theta [(c_f + c_r) l_f - (c_f + c_r) l_R + c_p (L_f - L_{psf})] - \theta [(c_f + c_r) l_f - (c_f + c_r) l_R + c_p (L_f - L_{psf})] - \theta l_f [k_f + k_r] l_R - k_p (L_f - L_{psf}) = 0\]

III. MODELING OF ROAD ROUGHNESS

The analysis of the effect of the bump height on the vehicle dynamics and of the forces on vehicle components for two different road profiles is shown in Figures 2-3. In the model illustrated in Figure 2, the amplitude of the first bump is higher than the amplitude of the second bump. The mathematical expression is given in (9), where \( E1, E2 \) represent the amplitudes and \( G1, G2 \) represent the width of the road defects. In (9), \( T1 = B + G1 \) and \( T2 = B + G1 + A + G2 \). In the second model (Figure 3), the amplitudes of the irregularities in the road are the same. The mathematical expression is given in (12) [31].
The effects of speed bumps, on the dynamics of the vehicle were examined for five different vehicle speeds and two road profiles. These speeds were taken as 60km/h, 90km/h, 120km/h, 150km/h, and 180km/h respectively.

A. Results of the First Road Irregularity

The first road irregularity profile is given in Figure 2. Under this road irregularity, passenger seat vertical displacement and car body vertical, pitch, and roll displacements are shown in Figure 4, for five different vehicle velocities. When vehicle speed increased, all displacement values decreased, and peak point locations occurred more at the right side, for all displacement figures because of the increase in the taken road during the period of oscillation. Passenger seat displacement values are higher than car body’s and this situation shows that passenger seat stiffness and damping coefficients could have more suitable values. But compared with road irregularities, more than 80% of disturbances of the road are isolated according to the slowest vehicle speed that has the highest displacement values. Pitch displacement values are rather higher than roll’s yet these values are so small, the highest value is rather smaller than 1\(^{\text{st}}\), that they could be considered negligible. For the first road irregularity profile, the speed graphs of the full vehicle model are shown in Figure 5. Like displacement graphs, it is observed that the velocity values decrease as vehicle speed increases. The changes in car body speed are greater compared to the ones at the passenger seat. Pitch and roll movement velocities and displacement values are low.

Acceleration graphs of the full vehicle model are shown in Figure 6. Like the displacement and velocity graphs, acceleration values decrease when vehicle speed increases. While passenger seat acceleration at the lowest vehicle speed is in the “extremely uncomfortable” region, as the vehicle speed increases, it moves towards the “uncomfortable” region. As for the car body acceleration, both vertical and pitch movement accelerations are in the “extremely uncomfortable region”, according to ISO 2631-1 [31, 33]. As vehicle speed increases, peak values locations exist more at the right side for all accelerations as in the displacement figures. This situation shows that the spring stiffness and the damping coefficient of the passenger and the vehicle body are not suitable and need to be reconsidered. When the passenger acceleration is taken into consideration, it is observed that the values obtained are not suitable for driving comfort. Roll movement accelerations are not considered high when compared with the pitch values. The force graphs of the full vehicle model are shown in Figure 7. Acting forces upon the passenger seat are not high, and like displacement, velocity, and acceleration graphs, when the speed of the vehicle increases, these values decrease and peak point locations move towards to the right side. When the forces acting on the vehicle body are examined, it is observed that the forces in the roll movement direction are not too high. But vertical and pitch direction forces have high-level values, and thus, they could be harmful to the car body equipment especially considering the bearing location.

| Parameter | Value | Unit |
|-----------|-------|------|
| Passenger and seat mass | 80 | kg |
| Car body mass | 1500 | kg |
| Wheel mass | 50 | kg |
| Car body’s moment of inertia on x axis | 1680 | kgm\(^2\) |
| Car body’s moment of inertia on y axis | 1500 | kgm\(^2\) |
| Seat damping coefficient | 600 | Ns/m |
| Front damping coefficient | 1200 | Ns/m |
| Rear damping coefficient | 1000 | Ns/m |
| Front wheel damping coefficient | 60 | Ns/m |
| Rear left wheel damping coefficient | 90 | Ns/m |
| Seat suspension stiffness | 8000 | N/m |
| Front suspension stiffness | 30000 | N/m |
| Rear suspension stiffness | 20000 | N/m |
| Front wheel suspension stiffness | 150000 | N/m |
| Rear wheel suspension stiffness | 120000 | N/m |
| Center of gravity - right wheel distance | 0.75 | m |
| Center of gravity - right wheel distance | 0.75 | m |
| Center of gravity - front wheel distance | 1.4 | m |
| Center of gravity - rear wheel distance | 1.1 | m |
| Passenger - left wheel distance | 0.4 | m |
| Passenger - front wheel distance | 1 | m |

TABLE II. ROAD PARAMETERS

| Parameters | Values | Parameters | Values |
|------------|--------|------------|--------|
| B          | 5m     | B          | 10m    |
| A          | 10m    | A          | 3m     |
| G1         | 1m     | D          | 0.3m   |
| G2         | 1m     | E          | 0.05m  |
| E1         | 0.1m   |            |        |
| E2         | 0.05m  |            |        |
Fig. 4. For the first road irregularity profile: passenger seat vertical displacement and car body vertical, pitch, and roll movement displacements.

Fig. 5. For the first road irregularity profile: passenger seat vertical velocity, car body vertical, pitch, and roll movement velocities.
Fig. 6. For the first road irregularity profile: passenger seat vertical acceleration, car body vertical, pitch, and roll movement accelerations

Fig. 7. For the first road irregularity profile: passenger seat vertical force, car body vertical, pitch, and roll movement forces
B. Results of the Second Road Irregularity

Fig. 8. For the second road irregularity profile: passenger seat vertical displacement, car body vertical, pitch, and roll movement displacements

Fig. 9. For the second road irregularity profile: passenger seat vertical velocity, car body vertical, pitch, and roll movement velocities
Fig. 10. For the second road irregularity profile: passenger seat vertical acceleration, car body vertical, pitch, and roll movement accelerations.

Fig. 11. For the second road irregularity profile: passenger seat vertical force, car body vertical, pitch, and roll movement forces.
The second road irregularity profile is given in Figure 3. Under this road irregularity, change of passenger seat vertical displacement and car body vertical, pitch, and roll displacements are shown in Figure 8, for five different vehicle velocities. It is observed that there is a decrease in the magnitude of displacement values that occur as vehicle speed increases. The peaks of the displacement values occur more on the right side, as in the first road graphs. In these graphs, this situation differs slightly for the first speed only and this is probably due to the fact that the bumps are closer to each other. The velocity graphs are shown in Figure 9. We can see that speed amplitude increases as vehicle speed increases from 60 to 120km/h, while speed amplitude decreases as vehicle speed increases from 120 to 180km/h. The acceleration graphs are shown in Figure 10. While passenger seat acceleration at the lowest vehicle speed is in the “little uncomfortable” region, as the vehicle speed increases, it moves towards the “not uncomfortable” region, according to ISO 2631-1. When the passenger comfort is taken into consideration, it could be said that the obtained values provide driving comfort. Also, compared with the first road profile, it is observed that while the amplitudes are reduced by half, in acceleration values significant decreases occur. According to the ISO 2631-1 standard, the acceleration values are changing from the “extremely uncomfortable” region to the “little uncomfortable” region. Both the vertical and the pitch accelerations of the car body are in the “extremely uncomfortable” region. This shows that the acceleration values on the vehicle body are at undesirable levels. Roll movement accelerations are not high when compared with pitch values. According to Figure 11, vehicle roll movement force and passenger vertical force have low amplitude levels. This shows that the equipment of the passenger seat is under the suitable condition. Similarly, bearing equipment of the car body is not under high-level force considering roll movement. As for the vertical and the pitch movement forces, they don’t have low but high force values. The car body equipment should be designed considering these dynamic acting forces.

VI. CONCLUSIONS

The main contribution of this study is the analysis of the forces applied to the vehicle body and the driver's seat. For dynamical analysis, the Newmark average acceleration method was used, which is not common in the literature. In this paper, the full vehicle model was considered and the dynamical behaviors of the vertical and rotational movements have been analyzed for the driver's seat and the car body. Two different road irregularity models were created and analysis was run with Newmark average acceleration method. For the first road irregularity, accelerations are in the “extremely uncomfortable” region except for the roll movement accelerations. When the second road profile was examined, as the amplitude of the road profile disturbance was halved, it is seen that the acceleration values pass to the “little uncomfortable” and “not uncomfortable” regions according to the ISO 2631-1 standard. For the first road profile, the vertical force and the pitch movement force of the car body have high values, and thus the equipment of the vehicle should be designed considering these forces. Similar with the acceleration values, under the second road profile irregularity, road disturbance amplitude was halved and the vertical and the pitch forces values of the car body were significantly decreased. The forces of the seat and the roll movement of the car body were not in a high-level for both road profiles. On the other hand, as the vehicle speed increased, it was observed that there was an increase in the period of movement of both the vehicle and the seat.

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