Effects of Spring Stiffness on Suspension Performances Using Full Vehicle Models

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ABSTRACT

Suspension system has significant influence on the passenger safety, providing comfortable ride, stability, and handling of the vehicle. The aims of the present research are to investigate and quantify the effect of spring weakness on suspension performance. This is based on a MATLAB simulation analysis of a seven degree-of-freedom (7-DOF) model for a full vehicle. In the simulation, the suspension faults were seeded by reducing the spring stiffness by 25%, 50% and 80%. The model was validated using experimental data, collected by driving the vehicle across bumps.

The simulation results for varying degrees of spring stiffness indicated that the ride comfort was decreased as the spring stiffness was increased for excitation frequencies close to resonant frequencies of the vehicle body (approximately 1 Hz). As spring stiffness was increased at excitation frequencies below 1 Hz, the suspension travel was reduced. Within the zone of resonant frequency of sprung mass, the deformation amplitudes were increased as the spring stiffness increased. Moreover, Frequency Response Functions analysis has been used for fault detection of reduction of spring's stiffness by 25%, 50% and 80%.

Keywords: Ride comfort; Road handling; vehicle stability; Vibration measurement; spring stiffness.

1 Introduction

The main function of suspension system is to support and carry the vehicle weight, to protect drivers and passengers from vibrations, and to maintain significant contacts between the tyre and the road surface [1]. For vehicles, it is a difficult challenge to consistently maintain a high standard of ride comfort and vehicle handling under a range of driving conditions. Between October 2010 and September 2011, the Ministry of Transport (M.O.T) collected data [2] in the UK in respect of MOT tests for approximately 24.2 million vehicles. These data was illustrates that lighting and signalling problems accounted for the highest number of re-tests (19.79%), followed by suspension faults (13.18%) and 8.75% (the fourth most common fault) were tyre faults. Early detections of abnormal events in automotive suspension systems can reduce the damage caused to the vehicle in driving situations, in addition to improving passenger comfort and security. The performance of a vehicle is often downgraded due to the appearance of faults with the suspension [3]. The common faults associated with suspension components are damaged or leaking shock absorbers, spring weakness, wearing down of the pivot and bushing and damage to the main support member assembly, as shown in Figure 1.

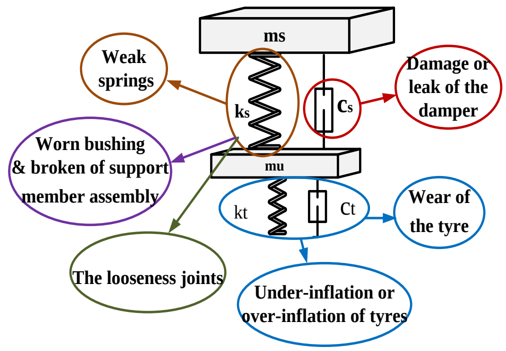

Figure 1: *common faults in suspension systems*

Faults occurring in the damping system can be as a result of one or more of the following factors: worn seals, a reduction in the oil volume due to leakages, broken mounts and extruded or worn bushings. All of the aforementioned causes can lead to a decline in the performance of the shock absorber, resulting in longer braking distances. This then causes the tyres to wear away reducing the car handling during cornering [4].

In order to study the performance of the suspension in terms of ride quality, handling and stability of the vehicle, some important parameters must be considered. These parameters are: wheel deflection, suspension travel and the vehicle body acceleration, with the aim of achieving a small amplitude value for each of the same [3]. Road handling is associated with the relative displacement between the suspension and the road input (Zu - Zr). This is represented as wheel deflection. Suspension travel is defined as the relative displacement between the vehicle body and the wheel (Zs – Zu). This can be used for assessing the space required to accommodate the suspension spring. Ride comfort is related to the vehicle body motion sensed by the passenger. This requires the acceleration of the vehicle body (sprung mass) to be relatively small. According to ISO: 2631-1-1997[5] proper road handling must be in the region of 0.0508 m, whilst the standard value for suspension travel must be in the region of 0.127m (as a minimum value). The passenger is thought to feel highly comfortable if the RMS acceleration is below 0.315 m/s2.

A number of researchers have investigated suspension performance using modelling/simulation and experimental investigation. Faheem [6] investigated a mathematical model for a quarter car with 2-DOF and a half car with 4-DOF. Rao[7] developed a mathematical model of a 3-DOF quarter car with a semi-active suspension system. This model was used for the testing of skyhook and other strategies involving semi active suspension systems. Esslaminasa et al [8] developed a semi-active twin-tube shock via the modelling of one and two DOF, for a quarter car design. Darus [9] adopted a state space approach in

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developing a mathematical model for both a quarter car and a full car using MATLAB packages. Metallidis [10], applied a statistical system identification technique to perform parametric identification and fault detection of nonlinear vehicle suspension systems. Kashi [11] applied model-based fault detection on a vehicle control system, which relied on mathematical descriptions of the system, yielding robust fault detection and an isolation of faults affecting the system. Agharkakli et al [12] presented a mathematical model for passive and active quarter car suspension systems. Ikenaga et al [13]conducted a research study to improve road handling and ride comfort. An active suspension control system was presented based on a full vehicle model which included the performance of the suspension system. Lu et al [14]discussed the effect of truck speed on shock and vibration levels indicating that the effect of truck speed on the root mean square acceleration of the vibrations, were strong at a lower speed but weak at a higher speed.

Sekulic et al[15] present a research to study the effects of spring stiffness and shock absorber damping on the vertical acceleration of the driver's body, suspension deformation and dynamic wheel load, with the purpose to define recommendations for selecting oscillation parameters while designing the suspension system of a (intercity) bus. Results of this research indicated that, the parameters which ensured good rid comfort of the driver were conflicting with the parameters which ensured the greatest stability of the bus and the corresponding wheel travel. Breytenbach [16] discussed the ride comfort versus handling argument for off-road vehicles. This research investigated a new approach of a semi-active suspension mode called "4 State Semi-active Suspensions", allowing a switch between low and high damping.

The objective of the present research is to investigate the effect of spring weakness on suspension performance, in addition to developing suspension condition monitoring based on a full vehicle mathematical model

2 Suspension System Model and Dynamics

Development of the vehicle model operates under the assumptions that the vehicle is a rigid body, represented as sprung mass (ms), and the suspension axles are represented as unsprung mass (mu) as shown in Figure 2. The suspension between the vehicle body and wheels are modelled by linear spring and damper elements and each tyre is modelled by a single linear spring and damper.

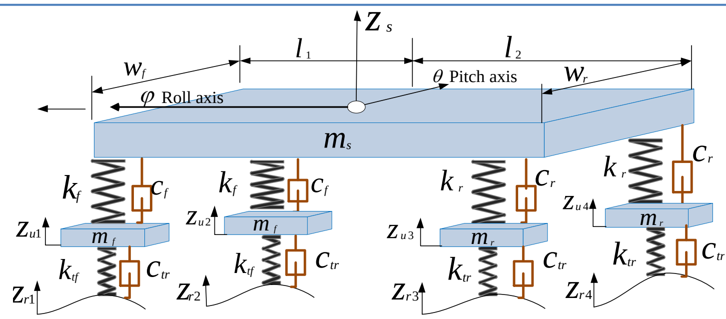

Figure 2: *Full vehicle models*

The equations of all motions are derived separately resulting in the equations of the body motions [8].

Equation of motion for bouncing of sprung mass:

$$m_s \ddot{z_s} = k_f (z_{u1} - z_{s1}) + k_f (z_{u2} - z_{s2}) + k_r (z_{u3} - z_{s3}) + k_r (z_{u4} - z_{s4}) +$$
$$c_f (\dot{z}_{u1} - \dot{z}_{s1}) + c_f (\dot{z}_{u2} - \dot{z}_{s2}) + c_r (\dot{z}_{u3} - \dot{z}_{s3}) + c_r (\dot{z}_{u4} - \dot{z}_{s4}) \quad (1)$$

For pitching of sprung mass

$$I_P \ddot{\theta} = k_f l_1 (z_{u1} - z_{s1}) + k_f l_1 (z_{u2} - z_{s2}) - k_r l_2 (z_{u3} - z_{s3}) - k_r l_2 (z_{u4} - z_{s4}) + c_f l_1 (\dot{z}_{u1} - \dot{z}_{s1}) +$$
$$c_f l_1 (\dot{z}_{u2} - \dot{z}_{s2}) - c_r l_2 (\dot{z}_{u3} - \dot{z}_{s3}) - c_r l_2 (\dot{z}_{u4} - \dot{z}_{s4}) \quad (2)$$

For rolling motion of sprung mass

$$I_R \ddot{\varphi} = k_f \frac{w_f}{2} (z_{u1} - z_{s1}) - k_f \frac{w_f}{2} (z_{u2} - z_{s2}) + k_l \frac{w_r}{2} (z_{u3} - z_{s3}) - k_r \frac{w_r}{2} (z_{u4} - z_{s4}) + c_f \frac{w_f}{2} (\dot{z}_{u1} - \dot{z}_{s1}) -$$
$$c_r \frac{w_f}{2} (\dot{z}_{u2} - \dot{z}_{s2}) + c_r \frac{w_r}{2} (\dot{z}_{u3} - \dot{z}_{s3}) - c_r \frac{w_r}{2} (\dot{z}_{u4} - \dot{z}_{s4}) \quad (3)$$

For each wheel motion in vertical direction

$$m_f \ddot{z}_{u1} = -k_f (z_{u1} - z_{s1}) - c_f (\dot{z}_{u1} - \dot{z}_{s1}) + k_{tf} (z_{r1} - z_{u1}) + c_{tf} (\dot{z}_{r1} - \dot{z}_{u1}) \quad (4)$$

$$m_f \ddot{z}_{u2} = -k_f (z_{u2} - z_{s2}) - c_f (\dot{z}_{u2} - \dot{z}_{s2}) + k_{tf} (z_{r2} - z_{u2}) + c_{tf} (\dot{z}_{r2} - \dot{z}_{u2}) \quad (5)$$

$$m_r \ddot{z}_{u3} = -k_r (z_{u3} - z_{s3}) - c_r (\dot{z}_{u3} - \dot{z}_{s3}) + k_{tr} (z_{r3} - z_{u3}) + c_{tr} (\dot{z}_{r3} - \dot{z}_{u3}) \quad (6)$$

$$m_r \ddot{z}_{u4} = -k_r (z_{u4} - z_{s4}) - c_r (\dot{z}_{u4} - \dot{z}_{u4}) + k_{tr} (z_{r4} - z_{u4}) + c_{tr} (\dot{z}_{r4} - \dot{z}_{u4}) \quad (7)$$

The equation variables are defined and summarized in Table 1 (adopted from [9]), along with the parameters of the suspension system. This is with the exception of the damping coefficient of the tyres for different pressures, which were adopted from [17]. Amendments were also made to some of the variables in order to meet the specifications of the vehicle used in the experiment. The road profile was calculated and created according to vehicle speeds and the height and width of the bumps by the following equation:

$$u(p) = 1/2a\sin(2\pi f_p t) \qquad (8)$$

The road profile was also assumed to be a single bump with a sin wave shape. Where a is the bump height (50)

Table 1: *Defines the equation variables and parameters of suspension*

Variables	Definition	Units	variables	Definition	Units
ms= 1200	Sprung mass	Kg	wf=90	Front vehicle width	m
mf=90	Unsprang mass	kg	wr=1.70	Rear vehicle width	m
kf=36279	Front spring stiffness	N/m	Zs≤0.06	Displacement of the vehicle body	m
kr=19620	Rear spring stiffness	N/m	zu1-zu4≈0.0508	Displacement of each wheel	m
cf=3924	Front damper coefficient	Nm/sec	Ir=5340	Roll and pitch of moment of inertia	$Kg.m^2$
cr =2943	Rear damper coefficient	Nm/sec	Ip =6430	Pitch of moment	$Kg.m^2$

3 Experimental Set up and Test Procedure

To validate the theoretical model, a front wheel drive Vauxhall ZAFIRA (2001) car, equipped with two different sensors was used. The sensors mounted on the car include: (1) a vibration sensor with a sensitivity of (3.770 pc/ms-2) mounted on the upper mounting point of the front left shock absorber, and (2) a dynamic tyre pressure sensor (DTPS) with a sensitivity of (11.43 Pc/0.1Mpa) connected to the valve stem of the front left wheel. The pressure sensor was situated in the centre rim of the front left wheel and the vibration sensor on the inside of the car. They were placed in these positions after being assembled and connected to the wireless sensor nodes (transmitters).The gateway (receiver) was equipped with a laptop inside the car. In order to ensure a sound installation of the sensors, two different adapters were designed and manufactured at the University of Huddersfield. In addition to this, a wireless measurement system was also designed and installed on the car, to offer a complete remote measurement for the vibration and pressure data being extracted. The most fundamental aim of the test was to obtain the acceleration (vibration) response of the suspension system to validate the model and also to enable a thorough analysis of the effects that different spring rates have on the performance of the suspension system. The test was conducted with the standard tyre pressure (2.3bar) and a vehicle speed of 8km/h.

4 Results and Discussion

The model was validated using experimental data collected when the vehicle was being driven, at a speed of 8km/h, over Bump 1 (located within the premises of The University of Huddersfield. The bump profile was 5.80 m width, 0.50 m length and 0.050 m height and this was assumed to be the input for the system. MATLAB software was used to analyze the vehicle response. Figure 3 depicts the acceleration of the vehicle body in the time domain based on the model simulation and experiments. Upon a comparison of the experimental results, it can be noted that the model fairly predicts the suspension performance.

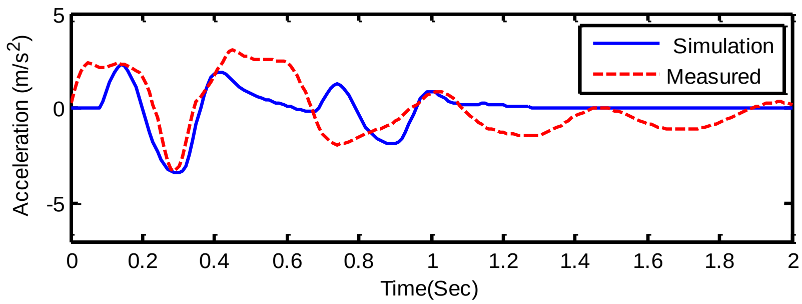

Figure 3 *Vibration (acceleration) of suspension simulation and experimental*

Figure 4(a) shows the plots of the road profile in the time domain for both the front and rear wheels of the vehicle. For the simulation study, road disturbance is assumed to be the input for the system. Figure 4(b) shows the effect of varying the spring rates on the vehicle body response. From these results, it was observed that decreases in spring stiffness causes a resultant decrease in the amplitude of the relative displacement of the car body. Figure 5 depicts the displacement of four wheels (unsprung mass) with different spring stiffness in the time domain. The results show that the amplitude/peak value of the wheels decreases with a corresponding decrease in the stiffness value. This indicates that the performance of the suspension may be affected by the changes made to the spring stiffness.

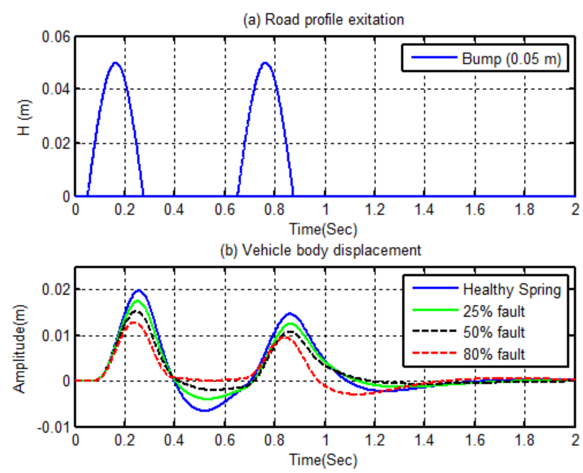

Figure 4 (a): *Road profile excitation and* **(b):** *Displacement of vehicle body for different spring stiffness.*

An analysis of different parameters such as: wheel deflection, suspension travel and acceleration of the vehicle body was carried out in order to consider the different effects the spring stiffness level has on the performance of the suspension, which includes, the ride quality and handling and stability of the vehicle. The road handling profile (wheel deflection) for a vehicle is associated with the contact forces between the road surface and the vehicle tyre (zu – zr).

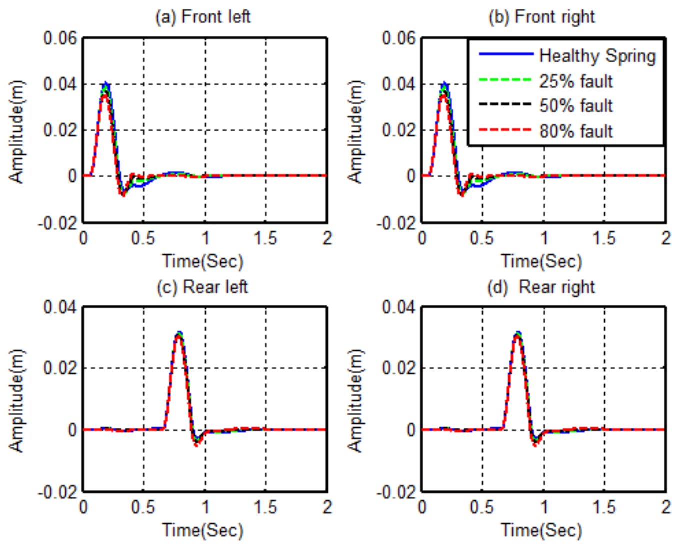

Figure 5 *Vehicle wheel's displacement with different spring stiffness*

For this simulation, the wheel deflections were approximately 0.015 m, 0.012 m, 0.008 m and 0.006 m for a healthy, 25%, 50%, 80% faulty spring respectively, as presented in Figure 6. From this figure, a noticeable change in the peak value of the wheel deflection can be observed. However, it can be noted that the vertical deflection does not decay quickly with the healthy spring, in particular, those with healthy and 25% faults. When compared with proper road handling as per ISO: 2631-1-1997 [4] (which must be in the range of m) this range is acceptable.

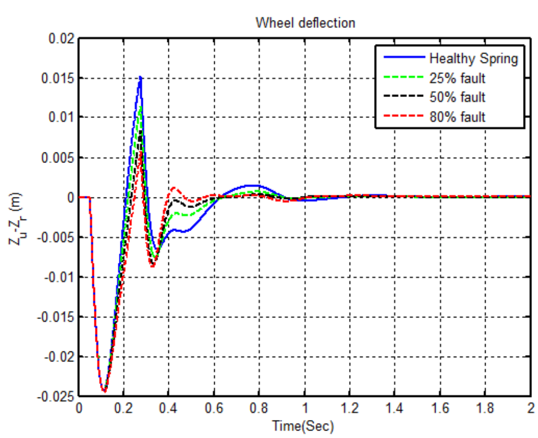

Figure 6 *Wheel deflections for different spring stiffness*

The suspension travel can be defined as a relative displacement between the vehicle body and the wheel (zs – zu) as shown in Figure 7. From this figure, it can be observed that lower spring stiffness provides for a lower suspension travel therefore to reduce the suspension travel a soft spring is required. In accordance with ISO: 2631-1-1997 [4] the passenger is thought to feel highly comfortable if the RMS acceleration is below 0.315 m/s^2.

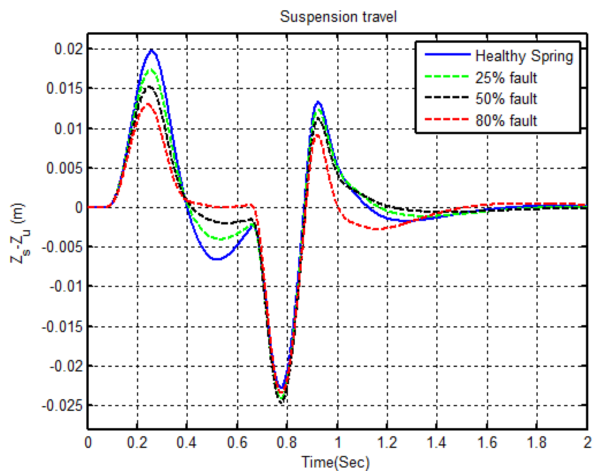

Figure 7 *Suspension travel for different spring stiffness*

In Figure 8 the amplitudes of the vertical acceleration were increased within the domain of the vehicle body (sprung mass) as the spring stiffness was increased. Lower values for the spring stiffness provided better oscillatory comfort for the passenger at excitation frequencies approximating the resonant frequency of the vehicle body. However, the vertical acceleration decays quickly with the reduction of the stiffness.

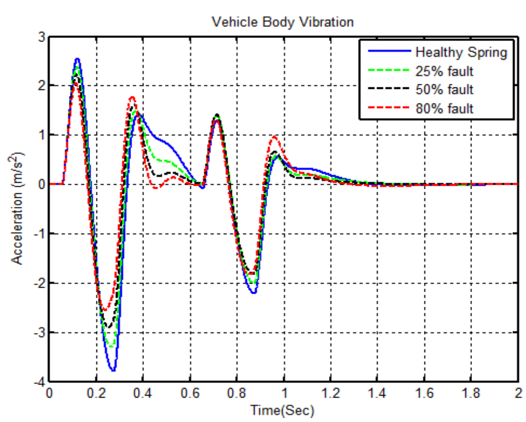

Figure 8 *Acceleration of the vehicle body for different spring stiffness*

To develop conditioned monitoring tools for suspension faults, detect the level of spring stiffness and also to predict potential suspension faults which may arise in the future, the Frequency Response Function (FRF) technique was used. The FRF is a fundamental measurement that isolates the inherent dynamic properties of a mechanical structure and also describes the input-output relationship between two points on a structure as a function of

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frequency. Figure 9 shows the amplitude-frequency characteristic curves for the changes to the spring stiffness of the suspension in four different output cases (vehicle body vertical displacement, vehicle body velocity, displacement of front and rear wheel) and the front left road input. The results show that decreasing the stiffness affects and reduces the value of the suspension displacement at the sprung mass natural frequency. A change in the amplitudes of displacement was more significant within the domain of resonant frequency of sprung mass (around 1Hz). A change of spring stiffness did not produce any effect on the change of the displacement and velocity of the vehicle body within the domain of resonant frequency of unsprang mass of the vehicle (around 10 Hz). However, the area under the curve does not necessarily decrease with a reduced peak value of the suspension displacement. From this, it can be concluded that in the frequency range close to the natural frequency of the vehicle body, a soft stiffness is required. However, lower stiffness also affects and produce vibrations in the mid- to high frequency range as shown in the front wheel responses.

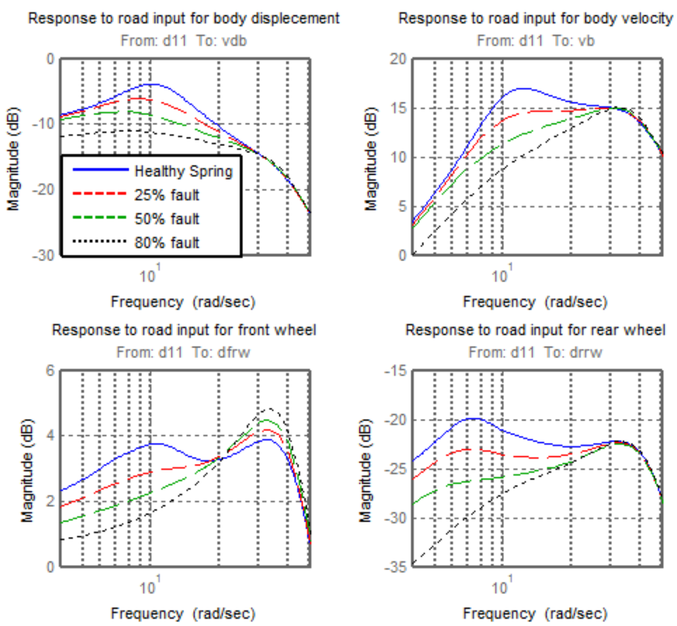

Figure 9 *Transfer function response for the vehicle body and vehicle wheels*

5 Conclusions

After collecting the relevant data using the full vehicle model, an analysis of the findings illustrates the effects on suspension performance when applying a range of spring stiffness. MATLAB was used to develop the full vehicle model with 7-DOF. Following this, analyses were carried out on the time and the frequency response of the vehicle. The analyses focus on the performances of the suspension in terms of ride comfort, road handling and stability of the vehicle. This study considered the faults of spring stiffness which were simulated by reducing the spring stiffness by 25%, 50% and 80%.

The simulation results indicated that the various parameters of suspension performance such as, ride comfort, road handling and vehicle stability need a design optimization due to the need to balance their conflicting requirements. For instance, the simulation results for varying of spring stiffness indicated that the ride comfort was decreased as the spring stiffness was increased for excitation frequencies close to resonant frequencies of the vehicle body (approximately 1 Hz).

FRF results show that, decreasing the stiffness affects and reduces the value of the suspension displacement at the sprung mass natural frequency. A change in the amplitudes of displacement was more significant within the domain of resonant frequency of sprung mass (around 1Hz). A change of spring stiffness did not produce any effect on the change of the displacement and velocity of the vehicle body within the domain of resonant frequency of unsprang mass of the vehicle (around 10 Hz). It can be concluded that, FRF methods can be effectively used for fault detection of suspension system.

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