Influence of Damping on Understeer Gradient Using Simulation and Measurement

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ABSTRACT

Safety one of the important criteria for assessment of vehicle suspension when cornering. For this reason, we studied the relationship of handling behaviour in term of tire deflection, which is one index for evaluating how good of the road holding .The quarter car suspension model is derived and used together with the kinematics measurement data. The testing results in form of suspension deflection are calculated from the sprung mass displacement, body roll and pitch, then those parameters are adjusted to be fit. After that, the lateral kinematics model is applied for an objective assessment of cornering manner. The experiment conducted by varying the damping constant. The excited frequency as well as vehicle speed is in the range of sprung mass mode natural frequency. The results are in accordance with the theory that the vehicle has the understeer behaviour.

Keywords: vehicle safety, tire deflection, kinematics measurement, understeer gradient

1. Introduction

Good vehicle suspension has to concern about the vehicle safety like yaw stability, which is the topic in our study. In order to keep good road holding performance, we will identify the dynamics interaction between sprung and unsprung mass, tire and ground as handling performance. Forces reacted to vehicle are commonly generated by a tire contact with the road surface. It is proportional to vertical tire load during driving considering spring element.

There are some former scientific works dealt with tire deflection analysis and related topic in (Spelta et al., 2009) proposed an interaction model between the suspension dynamics and the tire-road forces, on the basis an analysis focused on the evaluation metrics for handling performances based on optimal predictive control with road preview. The algorithm to estimate tire vertical forces presented by (Doumiati et al., 2008) applied roll and combined longitudinal lateral dynamics. The proposed method used measurements from currently available sensors like accelerometers and relative suspension sensors. There are also various work in related topic in suspension as well as damper; (Popescu, 2009) presented the performance test of suspension damper to assess the damp behaviour in simulation and experiment. Damping parameters were identified by 3 DOF model based on curve fitting method was proposed (Zhao et al., 2016). The study form (Zehsaz et al.,

2012) about suspension sensitivity analysis under off road excitation was demonstrated by comparing the results when the spring coefficient and damping constant changed.

. In the case of vehicle dynamics, the subject deals with the study of vehicle response related to driver input, ability of the vehicle to stabilize its motion against external disturbances for purpose of handling characteristics, especially cornering dynamic characteristics (e.g., lateral acceleration, side slip angle, Yaw rate and understeer gradient) (Kim and Ryu, 2011). The study aimed at evaluating the bus performance by considering understeer gradient are stated by (Prompakdee et al., 2016)

Our study aims at evaluating a handling performance when varying the damping constant by means of tire deflection approximation. We focus on road vehicles with quarter car suspension model. The kinematics measurement data via inertial measurement unit integrated with GNSS is used for our analysis together with the Matlab/Simulink/System identification/ Toolbox... The deflection will be translating at the additional weight distribution in front and rear mass. Finally understeer gradient is calculated and interpreted.

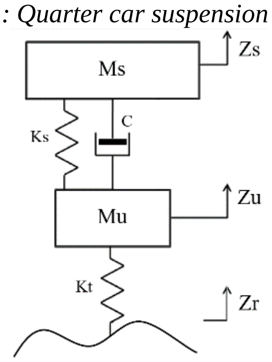
2. Vehicle and Modelling

There are two types of model concerning in the study as vertical dynamics model to determine the suspension deflection and the lateral dynamics model for investigating the cornering performance.

2.1 Vertical Dynamics Model

A simple quarter-car system is applied to this model. It will be described in figure 1 with the following state equations as in equation (1). z_s, z_u and z_r are the vertical positions of the sprung mass, the unsprung mass, and the road profile, respectively; m_s, m_u are the quarter-car body mass and unsprung mass including tire, wheel, brake calliper, suspension linkage.

Figure 1: Quarter car suspension model



$$\dot{x} = Ax + Bz_r \tag{1}$$

Where

$$x=\begin{bmatrix} z_s \\ \dot{z}_s \\ z_u \\ \dot{z}_u \end{bmatrix} \quad A=\begin{bmatrix} 0 & 1 & 0 & 0 \\ -k_s/m_s & -c/m_s & k_s/m_s & c/m_s \\ 0 & 0 & 0 & 1 \\ k_s/m_u & c/m_u & \dfrac{-k_s-k_t}{m_u} & -c/m_u \end{bmatrix} \quad B=\begin{bmatrix} 0 \\ 0 \\ 0 \\ k_t/m_u \end{bmatrix}$$

Normally we are interested in the criteria from this model as ride comfort in term of vertical acceleration, suspension stroke and tire deflection, then the three transfer function will be derived in equation (2) -(4) as ride comfort to the road input, suspension stroke to the road input and tire deflection to the road input.

$$H_{\ddot{z}_s/z_r}=s^2H_{z_s/z_r}=\frac{k_t(cs+k_s)s^2}{(m_ss^2+cs+k_s)(m_us^2+k_t)+m_ss^2(cs+k_s)} \tag{2}$$

$$H_{(z_s-z_u)/z_r}=\frac{k_tm_ss^2}{(m_ss^2+cs+k_s)(m_us^2+k_t)+m_ss^2(cs+k_s)} \tag{3}$$

$$H_{(z_u-z_r)/z_r}=\frac{m_us^2(m_ss^2+cs+k_s)+m_ss^2(cs+k_s)}{(m_ss^2+cs+k_s)(m_us^2+k_t)+m_ss^2(cs+k_s)} \tag{4}$$

The suspension parameters as spring stiffness, damping coefficient and mass in each wheel are roughly known from their specifications. We investigated spectrum analysis especially in vertical acceleration, vertical velocity in various cases from the experiment as shown in amplitude spectrum in figure. 2 compare to frequency response analysis from the equations above in order to evaluate the natural frequency. We compare both in sprung and unsprung mass mode and find that sprung mass mode natural frequencies varies between 1.7-2.1 Hz regarding to each mass and stiffness, while the unsprung mass mode is approximated between 11-12 Hz.

2.2 Lateral Dynamics Model

Aimed at cornering performance, the simplified bicycle model is considered to identify the relationship among the various parameters like lateral, longitudinal as well as yaw dynamics. In figure 3, X-Y fixed frame and x-y rotating frame at the vehicle's centre of gravity are defined. Assuming plane kinematics, the rotation angle about vertical axis, called yaw angle (ψ), is taken into account.

We conduct the test in steady state cornering which can verify the handling characteristics. The aim of this test is to determine the understeer gradient of difference configuration of weigh transfer. The front and rear wheel slip angle denote by α_f and α_r respectively as shown in figure 3. The instantaneous turn centre O of the vehicle is the point at which the two lines perpendicular to the velocities of the two wheels meet. Under the assumption that the road radius is much larger than the wheel base of the vehicle. From the geometry, we have the relationship of the steering angle δ that can be approximated as

$$\delta-\alpha_f-\alpha_r\approx\frac{L}{R} \tag{5}$$

Figure 2. Amplitude spectrum compare to frequency response.

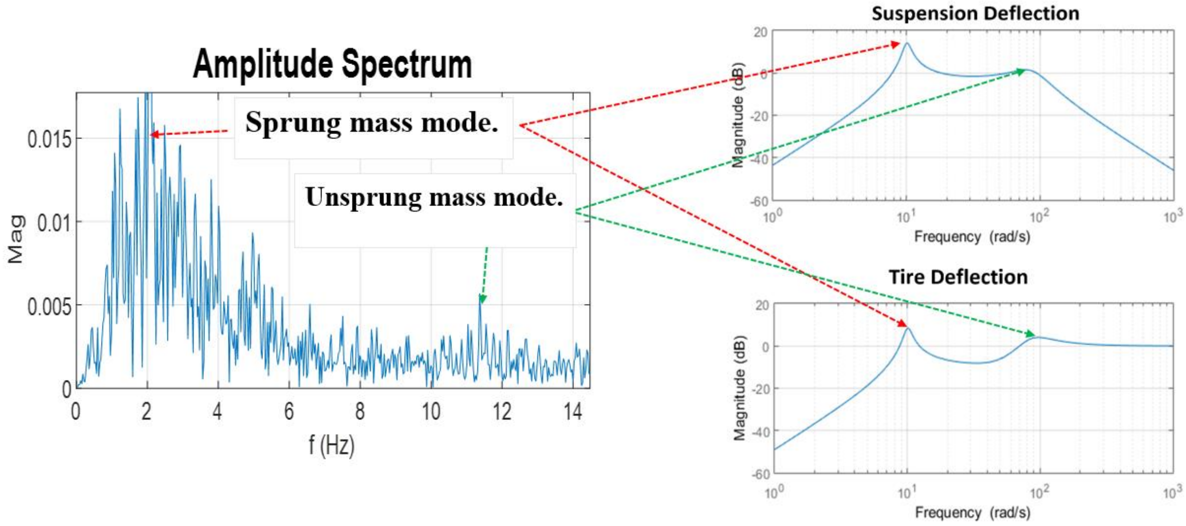
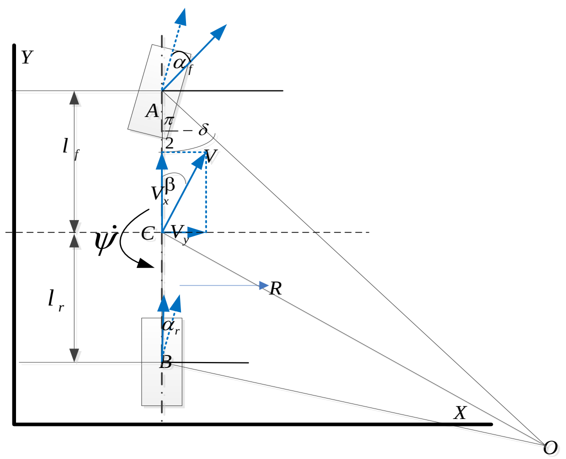


Figure 3 Kinematics model when cornering



Consider the dynamics equations, α_f and α_r are related to the forces acted on tired and the small slip is assumed so that the lateral tired force is proportional to its slip angle with the constant called cornering stiffness for front C_f and rear C_r

The steady state steering angle in equation 3 can be rewritten to

$$\delta = \frac{L}{R} + \alpha_f - \alpha_r = \frac{L}{R} + \left(\frac{m_f}{2C_f} - \frac{m_r}{2C_r} \right) \frac{V_x^2}{R} \tag{6}$$

or

$$\delta = \frac{L}{R} + K_v a_y \tag{7}$$

In this case, of understeer, the understeer gradient K_v should greater than zero due to a larger slip angle at the front tires compared to the rear tires.

2.3 Tested Vehicle

Passenger car equipped with IMU is used in this experiment. The dimension's width and track of vehicle described graphically in figure 4. The static weights in front left, front right, rear left and rear right are demonstrated in table 1. There are 4 shock absorber at each wheel and they can adjustable from 20% to 80 % of orifice.

Figure 4 Tested vehicle and its dimension.

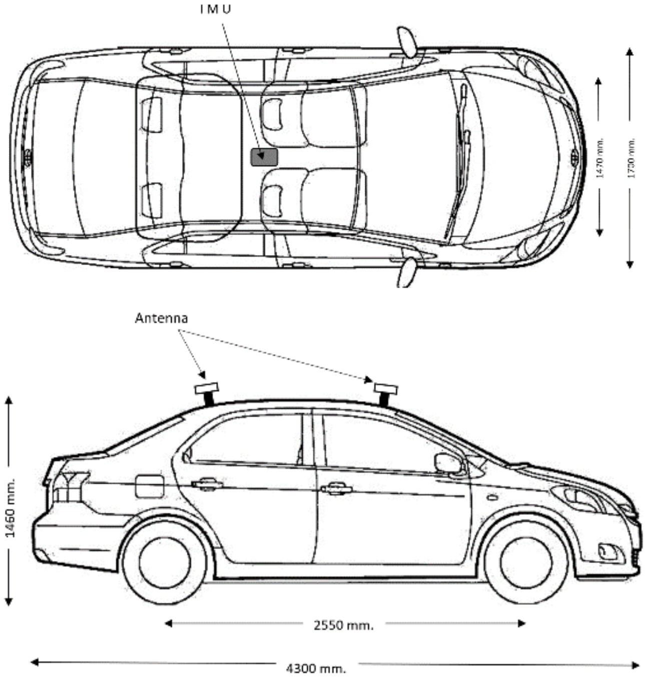


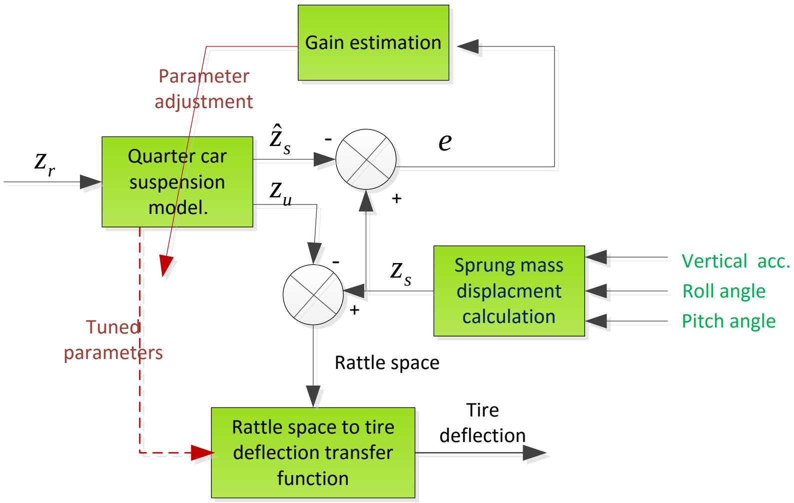
Table 1 Static Weight distribution.

	Front	Rear
Left (Kg.)	330	205
Right (Kg.)	325	210

3. Experiment Method

Since we can only measure the kinematics relationship at center of mass in term of the accelerations and angular rates which are received from IMU. We cannot directly calculate the tire deflection so that we need to approximate via the experiment data by the method stated in figure 5 . Road profile input are generated refer to the real one the response of sprung and unsprung mass displacements as well as velocity is is derived following equation (1) .At the same time the sensor reading data as vertical accelerations are integrated to get the displacement and the offset is compensated via roll and pitch angles .in equation)6)- (9 (This process, we got the sprung mass travelling for each wheel .The model and the data are compared as the sprung mass displacement error .Then we try to control the error via the gain estimation .The common PD rule for gain determination is described as three step error and three step change of error .The parameter tuning step is assisted by Matlab/ System identification toolbox .Once the parameters are adjusted, they are sent to block transfer function between rattle space together with the estimated unsprung mass displacement via the model response to tire deflection .

Figure 5 Tired deflection approximation method.



$$z = \iint a_z dt \tag{8}$$ (

$$z_{FL} = z + l_f\theta - b_L\varphi \tag{9}$$

$$z_{FR} = z + l_f\theta + b_R\varphi \tag{10}$$

$$z_{RL} = z - l_f\theta - b_L\varphi \tag{11}$$

$$z_{RR} = z - l_f\theta + b_L\varphi \tag{12}$$ (

After that the data from the stroke displacement and tire deflection are calculated in term of additional weights applied to equation (6) and the understeer gradient is calculated in each situation.

4. Results and Discussion

We conduct the experiments by varying the damping constant. Theoretically, it affects the response only when the excited frequency of riding is near to both mode of vehicle natural frequencies. In this test, we apply the steady state cornering test by keeping constant velocity as 25 km/h in a certain road surfaces, which leads to the input frequency is almost equal to sprung mass mode one.

The clockwise cornering test path in figure 6 is allowed in our test with the sensor data such as longitudinal and lateral acceleration. Roll and pitch angle are varied a little bit due to the disturbances like tilt, bank, hole and etc. The suspension deflection of each wheel is shown in figure 8 as well as the tire deflection in figure 9. Semi-active suspension is used in our experiments. The damping coefficient can be adjusted in our case 20% open.

From figure 8, the front is rebound. While cornering, lateral acceleration induces a roll angle, the right side of vehicle is deflected more than the left one because of weight transfer from counter-clockwise cornering. The effect from 20% damp open, bump and rebound rate are slower than 80% damp open because it has more compression than the other does.

The tire deflection of both case is not significant different from both cases of shock absorber but 80% damp open has more displacement on tire than 20% damp open .Objective validation is difficult to conclude, since it depends upon many factors like a tire configuration, suspension arrangement and etc., although there are many studies available.

For this reason, tire deflections are calculated from the rattle space data information and the results is in an agreement with our subjective evaluation via the questionnaire from experienced driver .

Figure 6 Cornering path

Figure 7 Measurement data from sensor.

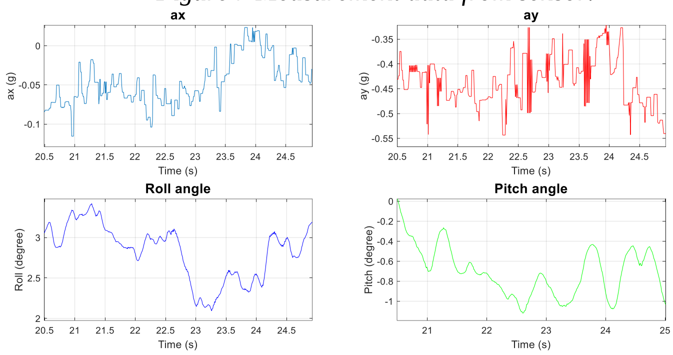


Figure 8 Suspension deflection approximation.

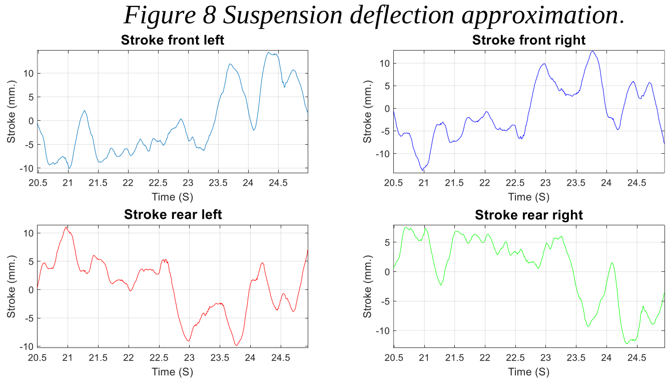


Figure 9 Tire deflection approximation.

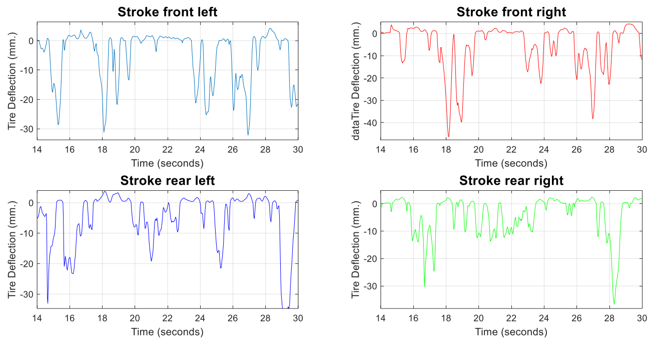


Table 2. Understeer gradient by varying damping ratio

Damp (%Orifice open) Front/Rear	Understeer Gradient (deg/g)	Damp (%Orifice open) Front/Rear	Understeer Gradient (deg/g)
20/60	2.35	60/20	1.70
40/60	2.21	60/40	1.75
60/60	1.89	60/60	1.89
80/60	1.48	60/80	1.90

Table 2 displayed the comparison of percent of damper orifice open. At the left side, we keep rear damper as constant and vary the front one. We found that at the less damping constant, the more deflection will be occurred. Then the weight is transferred to the front (Prompakdee et al.,2017) and it leads to understeer behaviour. In the configuration 80/60, the gradient is decreased but it is still understeer because of the nature of this car which has more weight in the front than rear. On the right side, we tried to fix the front damping constant and vary the rear damping. The gradient does not change significantly since its static weight.

5. Conclusion

This study focused on the influence of damping constant on the cornering understeer gradient. The vehicle math model as quarter car and the kinematics measurement data are applied together to approximate the tire deflection. Then the lateral kinematics model is considering for objective assessment of the cornering behaviour. The gradient change can explain in accordance with the theorem.

6. References

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