Influence of Spring Ratio on Variable Stiffness and Damping Suspension System Performance

Unggul Wasiwitono[1,a], Agus Sigit Pramono[2,b], I Nyoman Sutantra[1,c], and Yunarko Triwinarno[3,d]

[1]Automotive Laboratory, Mechanical Engineering Department, Faculty of Industrial Technology, Institut Teknologi Sepuluh Nopember (ITS), Surabaya, Indonesia

[2]Solid Mechanics Laboratory, Mechanical Engineering Department, Faculty of Industrial Technology, Institut Teknologi Sepuluh Nopember (ITS), Surabaya, Indonesia

[3]Vibration and Dynamics Laboratory, Mechanical Engineering Department, Faculty of Industrial Technology, Institut Teknologi Sepuluh Nopember (ITS), Surabaya, Indonesia

[a]unggul@me.its.ac.id, [b]pramono@me.its.ac.id, [c]tantra@me.its.ac.id, [d]yunarko@me.its.ac.id

Keywords: Variable stiffness and damping, Magnetorheological damper, Bouc-Wen model, Vehicle suspension performance

Abstract. The variable stiffness and damping (VSVD) suspension system offers an interesting option to improve driver comfort in an energy efficient way. The aim of this study is to analyze the influence of the spring ratio on the VSVD. The realization of the VSVD is obtained by the application of variable damping with magnetorheological (MR) damper. In this study, the nonlinear damping force characteristic of the MR damper is modeled with the Bouc-Wen model and the road disturbance is modeled by a stationary random process with road displacement power spectral density. It is shown from simulation that VSVD has a potential benefit in improving performance of vehicle suspension.

Introduction

The requirements regarding vehicle suspension systems are constantly increasing. The main function of the vehicle suspension is to provide drive comfort and handling performance. However, this performance requirements are conflicting. Various approaches have been proposed to improve the performance and to manage the trade-off between conflicting performance requirements of the vehicle suspension system such as adaptive control [1, 2], multi-objective control [3, 4], and model predictive control [5]. Among the many different types of electronically-controlled vehicle suspension systems, semi-active suspensions provide a very good compromise between cost (energy-consumption and actuators/sensors hardware) and performance.

A common semi-active suspension is characterized by the closed-loop regulation of damping. While the modulation of the damping coefficient is commonly used and can be easily obtained, the modulation of spring stiffness is a elusive problem. Recently, new method for semi-active actuation of suspension system is proposed in [6, 7, 8, 9] where both damping and stiffness coefficient are regulated. This method is commonly known as variable stiffness and damping (VSVD) suspension system. The performance achieved by VSVD suspension system using hydro-pneumatic technology is evaluated in [6]. The potential benefit in using VSVD suspension system in improving tire normal force and vehicle stability is presented in [7, 8].

In this study, it is analyzed to what extent the performance of VSVD vehicle suspension system can be attained by using MR damper and the spring ratio should be given to ensure both ride comfort and ride safety. The focus of the current study is the performance of vehicle suspension considering the ride quality, road holding and suspension deflection limit. Although the analysis in this study is based on simulation, a very realistic framework including the nonlinear behavior of MR dampers by using the well-known Bouc-Wen model and the use of disturbance model for the road excitation presented in [10], which is already validated by comparing it to a measured road's profile power spectrum.

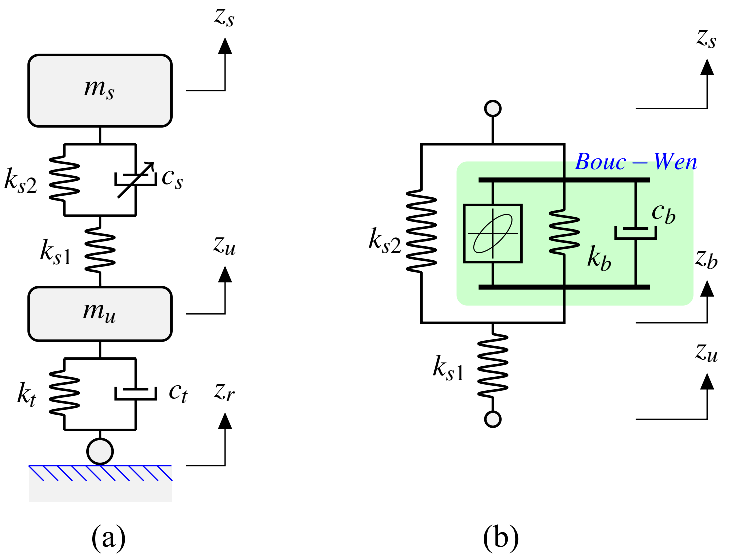


Fig. 1: (a) Quarter car model with VSVD; (b) VSVD with MR damper model

Quarter Car Model with Variable Stiffness and Damping System

The quarter-car model shown in Fig. 1a is considered in this study. In Fig. 1a, m_s is the sprung mass, which represents the car chassis; m_u is the unsprung mass, which represents the mass of the wheel assembly; k_{s1} and k_{s2} are the stiffness of the lower and the upper springs of the suspension system, respectively; c_s is the variable damping of the suspension system; k_t and c_t stand for the compressibility and damping of the pneumatic tire, respectively; z_s and z_u are the displacements of the sprung and unsprung masses, respectively; z_r is the road displacement input. The equivalent stiffness and damping coefficient can be varied by varying the value of c_s [7], as

$$k_{s_{eq}} = k_{s1} - \frac{k_{s1}^2 (k_{s1} + k_{s2})}{(k_{s1} + k_{s2})^2 + c_s^2 \omega^2} \tag{1}$$

$$c_{s_{eq}} = \frac{k_{s1}^2 c_s}{(k_{s1} + k_{s2})^2 + c_s^2 \omega^2} \tag{2}$$

where ω is the excitation frequency. Eqs (1) and (2) shows that the equivalent stiffness and damping coefficient are variable depending on the damping coefficient c_s and excitation frequency. The equivalent stiffness $k_{s_{eq}}$ will tend to be k_{s1}, and equivalent damping coefficient $c_{s_{eq}}$ will tend to be zero if damping coefficient c_s is very large. When damping coefficient c_s is closed to zero, the equivalent stiffness $k_{s_{eq}}$ will tend to be the value of k_{s1} and k_{s2} in series. Since the characteristics of the suspension system can be varied, the performance of the suspension system can be improved by the change of these characteristics.

The most common adjustable damper in use for today suspension system is MR damper. The nonlinear behavior of MR dampers can be described by several hysteresis models. In this study, the Bouc Wen's model is retained because it represents a good accuracy/complexity ratio. Fig. 1b shows the VSVD with MR damper modeled as Bouc-wen model. The model contains spring k_b, viscous damper c_b and a hysteretic component. The Bouc-Wen model can be described as the following couple equations [2]

$$F_b = (c_{bo} + c_{bv} v)(\dot{z}_s - \dot{z}_b) + k_b (z_s - z_b) + (\alpha_o + \alpha_v v)\,\omega \tag{3}$$

$$\dot{\omega} = -\rho\,|\dot{z}_s - \dot{z}_b|\,\omega - \beta\,(\dot{z}_s - \dot{z}_b)\,|\omega| + \lambda\,(\dot{z}_s - \dot{z}_b) \tag{4}$$

where ω is the internal state variable, v is the input voltage, k_b is the linear spring stiffness of Bouc Wen's model, α_o is the stiffness of ω, α_v is the stiffness of ω influenced by v, c_{bo} is the viscous damping coefficient of Bouc Wen model, c_{bv} is the viscous damping coefficient of Bouc Wen model influenced by v, and ρ, β, λ are the positive parameters characterizing the shape and size of the hysteresis loop.

Considering that the joint that connect k_{s1} and k_{s2} has a mass m_b, the equations of motion for the system shown in Fig. 1 are as follows:

$$m_s\ddot{z}_s = -k_{s2}\left(z_s - z_b\right) - F_b \tag{5}$$

$$m_b\ddot{z}_b = -k_{s1}\left(z_b - z_u\right) + k_{s2}\left(z_s - z_b\right) + F_b \tag{6}$$

$$m_u\ddot{z}_u = k_{s1}\left(z_b - z_u\right) - k_t\left(z_u - z_r\right) - c_t\left(\dot{z}_u - \dot{z}_r\right) \tag{7}$$

where F_b is the damping force. Define the following state variable $x_1 = z_s - z_b,\ x_2 = z_b - z_u,\ x_3 = z_b - z_u,\ x_4 = \dot{z}_s,\ x_5 = \dot{z}_b,\ x_6 = \dot{z}_u$. Further, define $w = \dot{z}_r$ and $x = \left[\begin{array}{cccccc} x_1 & x_2 & x_3 & x_4 & x_5 & x_6 \end{array}\right]^T$ the state-space form of the vehicle suspension system can be written as

$$\dot{x} = Ax + B_w w + B_f F_b \tag{8}$$

$$y = Cx + D_f F_b \tag{9}$$

Matrices $A,\ B_w,\ B_f,\ C,\ D_f$ are real constant matrices of appropriate dimension and defined based on Eqs. (5 - 7).

The road disturbances can be generally classified as shock and vibration [11]. An isolated bump or pothole in a road surface will caused a relatively short duration and high intensity disturbance that can be classified as shock. The latter road disturbance are consistent and typically specified as stationary random process with a road displacement power spectral density (PSD). A frequently used approximation of road displacement PSD is given in [11] as

$$S_{z_r}\left(f\right) = \frac{R_f}{v}\left(\frac{2\pi f}{v}\right)^n \tag{10}$$

where R_f is a constant roughness factor, f is the excitation frequency and v is the vehicle velocity. In [10] the PSD of approximation road profiles (10) is compared with the measured highway road profile. It is reported that the approximation by the disturbance model reflects the characteristics of the real road over a wide frequency range. When the parameter $n = -2$, PSD ground velocity is given by [10]

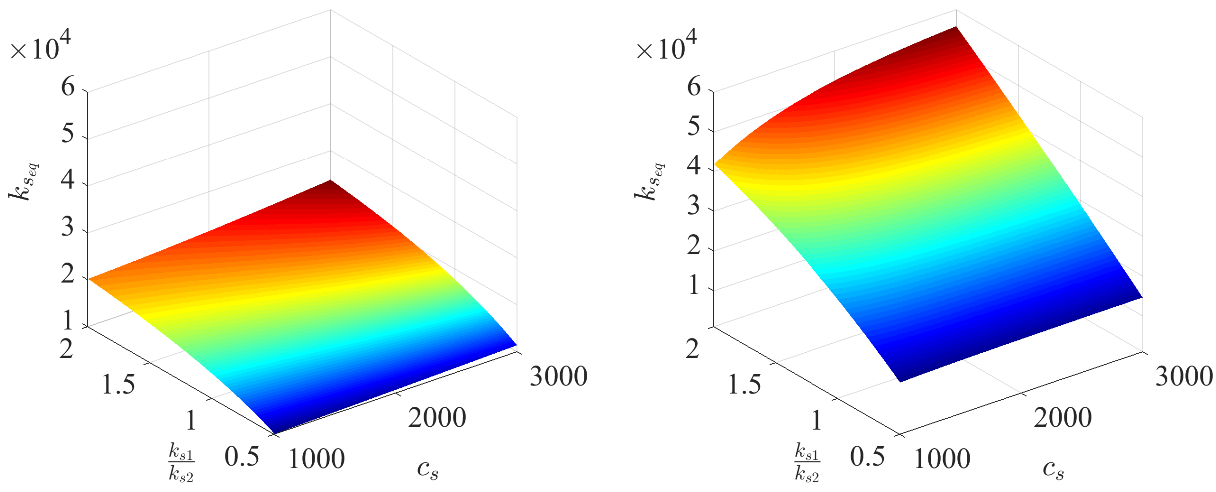
$$S_{\dot{z}_r}\left(f\right) = \left(2\pi f\right)^2 \cdot S_{z_r}\left(f\right) \tag{11}$$

depending on road conditions and driving speed.

Evaluation of Achievable Performance

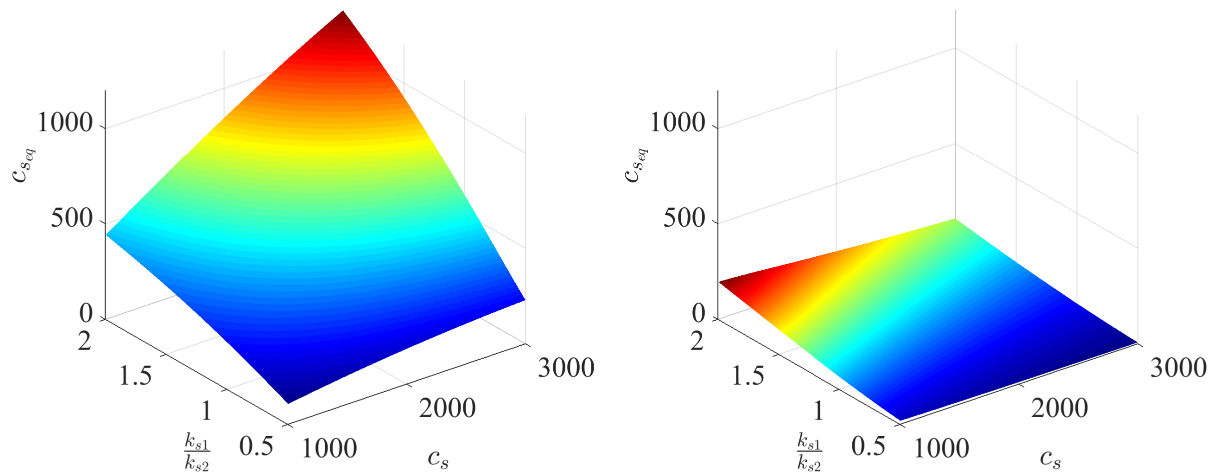
When designing the vehicle suspension, the following aspects should be considered: *Ride quality*: A vehicle suspension should provides isolation by reducing forces transmitted from the road unevenness to the vehicle body. The rms acceleration of the sprung mass is a widely used measure to quantify the ride comfort. Therefore, in control design, one of the objectives is to minimize the sprung mass acceleration; *Road holding*: In order to ensure a firm uninterrupted contact between the wheel and the road under disturbance, the dynamic tyre load should not exceed the static one; *Suspension deflection limit*: Suspension deflection has to be considered to avoid excessive suspension bottoming, which may lead to structural damage. The suspension stroke should be confined to a prescribed range.

Firstly, let analyze the influence of the spring coefficient ratio and the value of c_s on the equivalent stiffness and damping described in equations (1) and (2). The value of c_s is taken from 1000 to 3000 Ns/m, while the ratio is from 0.5 to 2. It can be seen from Figures 2 and 3 that in general increasing the ratio of spring coefficient k_{s1} to k_{s2} will increase both the equivalent stiffness and damping. In addition, setting the value of k_{s1} higher k_{s2} will result in longer range of $k_{s_{eq}}$ and $c_{s_{eq}}$ for a certain range of damping coefficient c_s. Furthermore, the equivalent damping is significantly influenced by



(a) (b)

Fig. 2: The influence of the spring coefficient ratio and c_s on $k_{s_{eq}}$ for frequency (a) 1.59 Hz and (b) 15.92 Hz



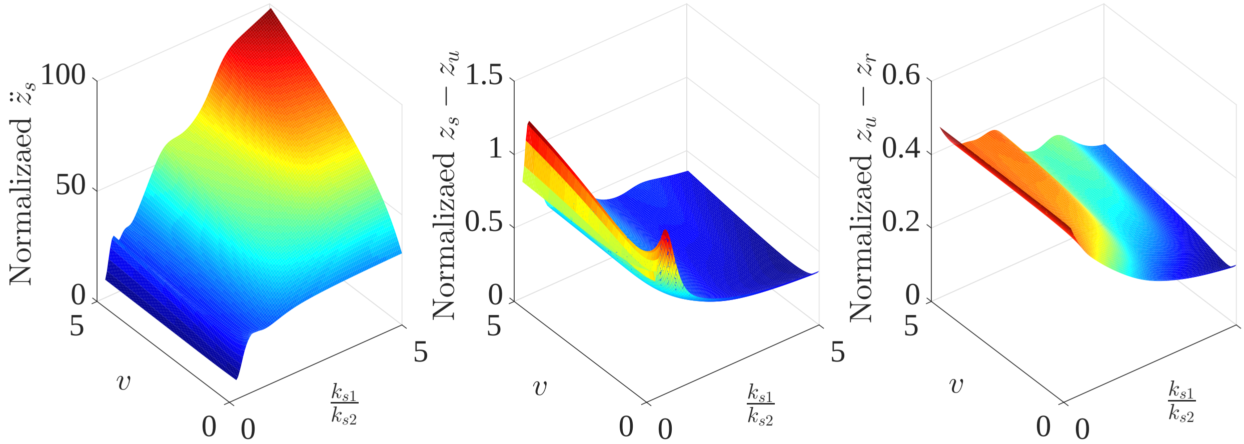
(a) (b)

Fig. 3: The influence of the spring coefficient ratio and c_s on $c_{s_{eq}}$ for frequency (a) 1.59 Hz and (b) 15.92 Hz

the excitation frequency. At excitation frequency, increasing damping coefficient c_s will increase $c_{s_{eq}}$. On the other hand, increasing damping coefficient c_s will decrease $c_{s_{eq}}$ at high excitation frequency.

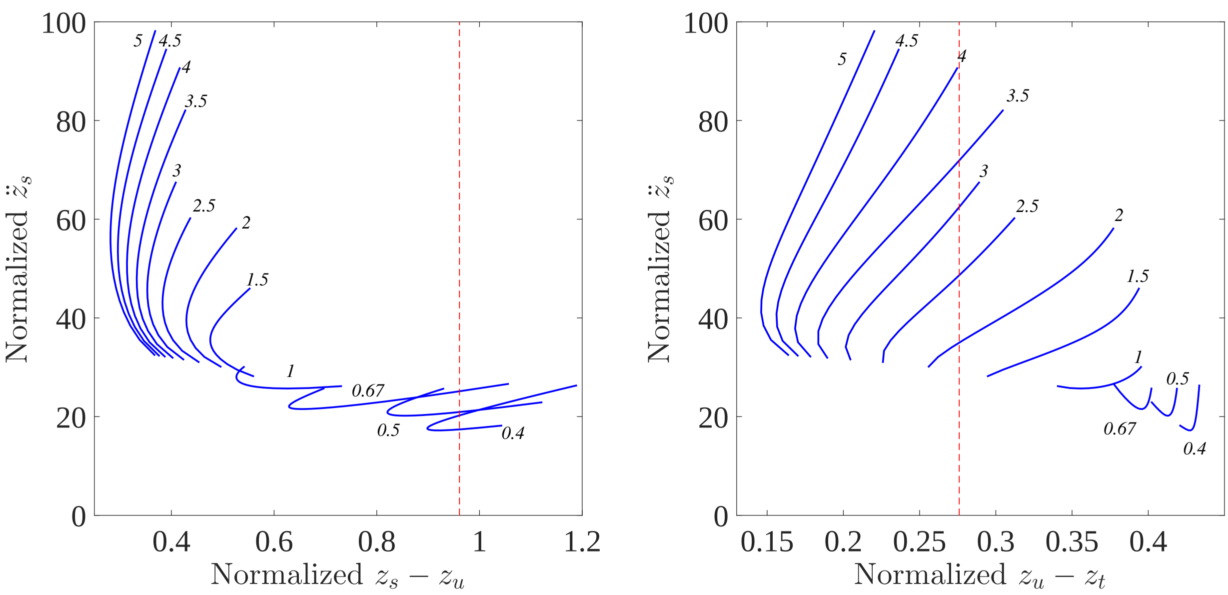
Based on the fact that the range of damping coefficient of MR damper is limited on certain value, in order to point out the potential of VSVD system, in this study, the vehicle and MR damper parameters used in [2] is used for simulation. Then, the influence of voltage input that correspond to the damper force of the MR damper and the ratio r_k of spring coefficient k_{s1} to k_{s2} is analyzed in similar way as presented in [10] where the normalized rms-value of performance output are used to ensure comparability of the systems performance independent of the road excitation. The road roughness coefficient is $A = 4.9 \times 10^{-6}$ m, which corresponds to a medium quality road [10, 11] is used for the simulation. Fig. 4 shows the influence of MR damper voltage input and and the spring ratio on normalized sprung mass acceleration, normalized suspension deflection and normalized tire deflection. It can be seen that increasing voltage input that correspond to increasing the suspension damping ratio will increase the acceleration of the sprung mass. Increasing voltage input in general also increase the suspension deflection and tire deflection. Whilst, increasing this ratio make the acceleration of the sprung mass increase, the suspension deflection and tire deflection will decrease.

Carpet plots of the normalized rms body acceleration versus the normalized rms suspension deflection and the tire deflection, respectively, are shown in Figure 5. In this Figure, the line represent the spring ratio from 0.4 to 5. Considering, the maximum acceptable suspension travel is ± 0.08 m and the tire deflection is limited to ± 0.023 m to ensure ride safety and using operating condition dependent constraints described in [10], the normalized constraint then equal to 0.961 and 0.276 for suspension travel and tire deflection, respectively. That constraint is showed by dashed line in Figure 5. Therefore, to ensure ride safety the spring ratio should be higher than 3.



(a) (b) (c)

Fig. 4: VSVD performance depending on the ratio of spring coefficient and MR damper voltage input



(a) (b)

Fig. 5: Carpet plots of VSVD for different ratio of spring coefficient

Summary

The performance of VSVD suspension system is analyzed considering the nonlinear behavior of MR damper. It is shown the spring ratio has significant effects on the VSVD performance. Therefore The challenge is to find a setting for spring ratio. VSVD has a potential in improving performance of the vehicle suspension by controlling the MR damper damping force. Further, research to find suitable control law for VSVD is another future work need to be considered.

References

[1] W. Sun, H. Gao, and O. Kaynak, Adaptive backstepping control for active suspension systems with hard constraints, IEEE/ASME Transaction on Mechatronics, Vol 18, No. 3, pp. 1072-1079, 2013.

[2] K. El Majdoub, D. Ghani, F. Giri, F. Z. Chaoui, Adaptive semi-active suspension of quarter-vehicle with magnetorheological damper, ASME journal of Dynamic Systems, Measurement, and Control, Vol. 137, No. 2, 2014.

[3] P. Li, J. Lam, and K. C. Cheung, Multi-objective control for active vehicle suspension with wheel base preview, Journal of Sound and Vibration, Vol 333, No. 21, pp. 5269-5282, 2014.

[4] J. H. Crews, M. G. Mattson and G. D. Buckner, Multi-objective control optimization for semi-active vehicle suspensions, Journal of Sound and Vibration, Vol. 330, No. 23, pp. 5502-5516, 2011.

[5] Y. Hu, M. Z. Q. Chen, and Z. Hou, Multiplexed model predictive control for active suspensions, International Journal of Control, Vol. 88, No. 2, pp. 347-363, 2015.

[6] C. Spelta, F. Previdi, S. M. Savaresi, P. Bolzern, M. Cutini, C. Bisaglia, and S. A. Bertinotti, Performance analysis of semi-active suspension with control of variable damping and stiffness, Vehicle System Dynamics, Vol. 49, No. 1-2, pp. 237-256, 2011.

[7] Y. Xu, and M. Ahmadian, Improving the capacity of tire normal force via variable stiffness and damping suspension system, Journal of Terramechanics, Vol. 50, pp. 121-132, 2013.

[8] Y. Xu, M. Ahmadian, and R. Sun, Improving vehicle lateral stability based on variable stiffness and damping suspension system via MR damper, IEEE Transaction on Vehicular Technology, Vol. 63, No. 3, pp. 1071-1078, 2014.

[9] Y. Liu, H. Matsuhisa, and H. Utsuno, Semi-active vibration isolation system with variable stiffness and damping control, Journal of Sound and Vibration, Vol. 313, pp. 16-28, 2008.

[10] G. Koch, O. Fritsch, and B. Lohmann, Potential of low bandwidth active suspension with continuously variable damper, Control Engineering Practice, Vol. 18, pp. 1251-1262, 2010.

[11] D. Hrovat, Survey of advanced suspension developments and related optimal control application, Automatica, Vol 33, No. 10, pp. 1781-1817, 1997.