2011 11th International Conference on Control, Automation and Systems
Oct. 26-29, 2011 in KINTEX, Gyeonggi-do, Korea

Skyhook Damper Modification Using Fuzzy Logic Control for Control Effort and NVH Reduction

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Abstract: This paper suggests a modified a semi-active vehicle suspension design based on the conventional skyhook damper mechanism. The skyhook algorithm is modified using the fuzzy logic control and a 6D IMU which measures the accelerations and angular rates of the vehicle about its C.G. to observe the vertical velocities of the sprung mass of each corner. Also, the proposed scheme utilizes the spring compression measurements of each tire to estimate the unsprung mass movement. These are combined to maximize the separation performance between the road and the sprung mass, while reducing the overall control effort and chattering that directly relates to vehicle NVH. A confirmatory assessment of the entire system is arranged via simulation scenarios involving an asymmetric vehicle chassis excitation and a sinusoidal excitation of varying frequency with the aid of well-known vehicle simulation tools, CarSim, and Matlab/Simulink.

Keywords: Semi-active suspension, fuzzy logic control, vibration

1. INTRODUCTION

A semi-active suspension system is known to be an efficient way to raise the suspension performance through effectively changing the damping coefficients either by controlling the variable orifice or applying electric current to the MR (magnetorheological) fluid in the damper.

A semi-active suspension comes with the advantage that it does not need an energy source like an active suspension system, at the cost that it can only dissipate energy and the damping coefficient must saturate to a near-zero limit when the desired damping force is in the same direction as the damper movement. This indicates that the damper can only operate half of the time.

Here, maximizing the operation range of the semi-active skyhook damper used in the vehicle suspension system may be an effective method to reduce the vertical acceleration experienced by the passengers and to separate the road disturbance from them. However, unrestricted control effort within the physical boundary to maximize the damper operation can bring excessive damper response or response rate, which directly relates to generation of an unpleasant noise from the variable orifice actuator or harshness from vehicle jerking.

Aware that the comfort factor inside the vehicle cabin space is emphasized to satisfy the customers' needs, the noise, vibration, and harshness generated from the semi-active suspension actuators are extremely undesirable. Such drawbacks have served obstacles in the previous efforts to develop a wholly satisfactory semi-active suspension system [1] [2] [3] [4] [5] [6] [7].

This paper introduces a semi-active suspension system control scheme based on the skyhook control, that minimizes the control effort and the actuator input rate, without deteriorating the conventional skyhook damper performance. This is done using a fuzzy logic control, and its performance is verified through simulation using CarSim, and Matlab/Simulink.

2. QUARTER CAR MODEL OF THE VEHICLE SUSPENSION SYSTEM

A quarter car model is considered for the modeling of the vehicle suspension for the sake of simplicity, as shown in Fig. 1. The damping effect of the tire is assumed to be negligible, and the tire excitation is modeled through the input from the road terrain z_r. In addition to the spring with the spring constant k_s and the passive damper with the damping coefficient b_s, the controllable damper is added to each corner with the damping coefficient b_{semi}.

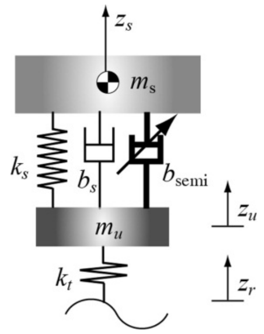

Fig. 1 Quarter car model

From the above model, the equations of motion are derived.

$$m_s\ddot{z}_s = -b_s\left(\dot{z}_s - \dot{z}_u\right) - k_s\left(z_s - z_u\right) - b_{semi}\left(\dot{z}_s - \dot{z}_u\right) \tag{1a}$$

$$m_u \ddot{z}_u = -b_s(\dot{z}_u - \dot{z}_s) - k_s(z_u - z_s) - k_t(z_u - z_r) - b_{semi}(\dot{z}_u - \dot{z}_s) \quad (1b)$$

3. CONVENTIONAL SKYHOOK DAMPER

The fully active skyhook damping force is obtained as the following.

$$F_a = -b_a \dot{z}_s \quad (2)$$

In case of the semi-active suspension, however, b is constrained to only dissipate energy from the suspension system. Within this physical boundary, the skyhook damper must operate as much as possible to reduce the sprung mass velocity. To resist the sprung mass movement, it is required that F_a is applied to the suspension system.

It must be noted here that for the ideal operation of the skyhook damper system, it requires an assumption that the unsprung mass to which the controllable damper is connected only undergoes a negligible amount of movement as if the sprung mass is connected to a completely stationary reference like the sky. This assumption is fulfilled only in the condition in which the road terrain serves no external excitation to the unsprung mass. It may sound practical in the situation where the suspension system is compressed or expanded solely due to the control effort of the driver to maneuver the vehicle on a completely flat ground. However, the validity of this assumption is the most deteriorated when the road disturbance is the main source of the spring damper system excitation, like when the vehicle is going over a speed bump.

Considering that the damping force actually is generated by the following equation,

$$F_d = -b_{semi}(\dot{z}_s - \dot{z}_u) \quad (3)$$

the influence of $\dot{z}_u$ to the skyhook damper performance is not desirable, as mentioned.

In order to take $\dot{z}_u$ into account effectively, b_{semi} is designed so that F_d tracks F_a.

Considering the aforementioned points, the semi-active damping coefficient b_{semi} can be defined as follows.

$$b_{semi} = \begin{cases} b_{\max} & , \ \gamma b_a > b_{\max} \\ \gamma b_a & , \ 0 < \gamma b_a \le b_{\max} \\ 0 & , \ \gamma b_a \le 0 \end{cases} \quad (4)$$

where $\gamma = \dfrac{v_s}{v_s - v_u}$

With the above b_{semi}, F_d is indeed made to track F_a within the physical and structural limitations of the semi-active suspension system, since

$$\begin{aligned} F_d &= -b_{semi}(\dot{z}_s - \dot{z}_u) = -\gamma b_a(\dot{z}_s - \dot{z}_u) \\ &= -b_a \frac{\dot{z}_s}{\cancel{\dot{z}_s - \dot{z}_u}} \cancel{(\dot{z}_s - \dot{z}_u)} \\ &= -b_a \dot{z}_s = F_a. \end{aligned} \quad (5)$$

Since Eq. (4) involves abrupt discontinuity, it increases the risk of excessive actuator input rate. A simple application of the low pass filter does not resolve the issue, because the phase lag problem may lead to system instability (to an extent that the road disturbance separation performance is majorly deteriorated). This calls for the need of a modified skyhook control, which is dealt in the following section.

4. SKYHOOK DAMPER MODIFICATION

This section deals with the modification of the conventional skyhook damper system. The following figure describes the schematics of the fuzzy logic based skyhook damper control.

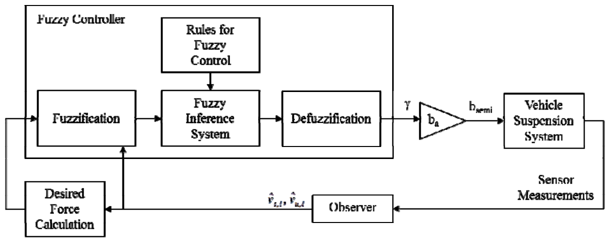

Fig. 2. Schematics of the fuzzy logic based control

4.1. Sprung Mass and Unsprung Mass Velocity Estimation

Critical information required by the skyhook control is the vertical velocity of each corner. Given the vehicle angular rates – roll rate p and pitch rate q – the vertical sprung mass velocity of each corner can be obtained based on the vertical velocity v_z estimated at the vehicle C.G. This indicates the need for the observer that estimates v_z, and it is done through the vehicle velocity and attitude observer [8]. In this work, an observer based on the bicycle model and the vehicle kinematic laws is designed.

With v_z available, individual sprung mass velocity of each corner can be obtained as follows.

$$\hat{v}_{s,fL} = \hat{v}_z - l_f q + wp \quad (6a)$$

$$\hat{v}_{s,fR} = \hat{v}_z - l_f q - wp \quad (6b)$$

$$\hat{v}_{s,rL} = \hat{v}_z + l_r q + wp \quad (6c)$$

$$\hat{v}_{s,rR} = \hat{v}_z + l_r q - wp \quad (6d)$$

Since the sensor measurements for the suspension jounce z_j of each corner are available, it is possible to express the vehicle unsprung mass as the following.

$$z_j = z_u - z_s \;\Rightarrow\; \dot{z}_j = \dot{z}_u - \dot{z}_s \quad (7)$$

$$\Rightarrow v_u = \dot{z}_u = \dot{z}_j + \dot{z}_s \approx \dot{z}_j + \hat{v}_s \quad (8)$$

Thus, the estimated unsprung mass velocity of each corner expressed in terms of v_z is as follows.

$$\hat{v}_{u,fL} = \dot{z}_{j,fL} + \hat{v}_{s,fL} = \dot{z}_{j,fL} + \hat{v}_z - l_f q + wp \quad (9a)$$

$$\hat{v}_{u,fR} = \dot{z}_{j,fR} + \hat{v}_{s,fR} = \dot{z}_{j,fR} + \hat{v}_z - l_f q + wp \quad (9b)$$

$$\hat{v}_{u,rL} = \dot{z}_{j,rL} + \hat{v}_{s,rL} = \dot{z}_{j,rL} + \hat{v}_z - l_f q + wp \qquad (9c)$$

$$\hat{v}_{u,rR} = \dot{z}_{j,rR} + \hat{v}_{s,rR} = \dot{z}_{j,rR} + \hat{v}_z - l_f q + wp \qquad (9d)$$

4.2 Damping Coefficient Attenuation through Fuzzy Logic Control

The fuzzy logic control of the semi-active suspension damping coefficient is designed so that it takes two input values – sprung mass velocity $\hat{v}_{s,i}$ and unsprung mass velocity $\hat{v}_{u,i}$ – to determine γ_f. Once γ_f is obtained, γ can be defined as follows.

$$\gamma = \gamma_f{}^n \qquad (10)$$

where n is a positive constant.

Here, γ holds the same meaning as that dealt in the previous section. This time, in order to eliminate discontinuity involved in the value of γ when obtained analytically as in the case of conventional skyhook control, it is obtained through the fuzzy logic method.

Table 1 shows the linguistic variables for fuzzy control and the center value of each membership function.

Table 1. Linguistic variables for fuzzy control

$\hat{v}_{s,i}$	AN4	AN3	AN2	AN1	A0	AP1	AP2	AP3	AP4
[m/s]	-1	-0.75	-0.5	-0.25	0	0.25	0.5	0.75	1
$\hat{v}_{u,i}$	BN4	BN3	BN2	BN1	B0	BP1	BP2	BP3	BP4
[m/s]	-1	-0.75	-0.5	-0.25	0	0.25	0.5	0.75	1
γ_f	Y0	Y0.25	Y0.5	Y0.75	Y1	Y1.33	Y2	Y3	Y4
-	0	0.25	0.5	0.75	1	1.33	2	3	4

Table 2. Rules for fuzzy control

$\hat{v}_{s,i}$ \ $\hat{v}_{u,i}$	BN4	BN3	BN2	BN1	B0	BP1	BP2	BP3	BP4
AN4	Y1	Y4	Y2	Y1.33	Y1	Y0.75	Y0.75	Y0.5	Y0.5
AN3	Y0	Y1	Y3	Y1.33	Y1	Y0.75	Y0.5	Y0.5	Y0.5
AN2	Y0	Y0	Y1	Y2	Y1	Y0.75	Y0.5	Y0.5	Y0.25
AN1	Y0	Y0	Y0	Y1	Y1	Y0.5	Y0.25	Y0.25	Y0.25
A0	Y0	Y0	Y0	Y0	Y1	Y0	Y0	Y0	Y0
AP1	Y0.25	Y0.25	Y0.25	Y0.5	Y1	Y1	Y0	Y0	Y0
AP2	Y0.25	Y0.5	Y0.5	Y0.75	Y1	Y2	Y1	Y0	Y0
AP3	Y0.5	Y0.5	Y0.5	Y0.75	Y1	Y1.33	Y3	Y1	Y0
AP4	Y0.5	Y0.5	Y0.75	Y0.75	Y1	Y1.33	Y2	Y4	Y1

Table 2 shows the rules used in the fuzzy control. The variables on the left column and the top row are the IF elements, whereas the variables elsewhere are the THEN elements. The logical relationship between the two are always AND.

Figure 3 shows the shapes and distribution of the membership functions that correspond to the two inputs, $\hat{v}_{s,i}$ and $\hat{v}_{u,i}$, and an output γ_f graphically in order, and figure 4 shows the overall fuzzy control surface between the variables.

Once the output γ is obtained, its implementation into the semi-active suspension is analogous to that shown in Eq. (4). A difference from Eq. (4) is that, here, simply

$$b_{semi} = \gamma b_a \qquad (11)$$

will suffice, since the physical boundaries are already accounted when specifying the variables for the fuzzy control.

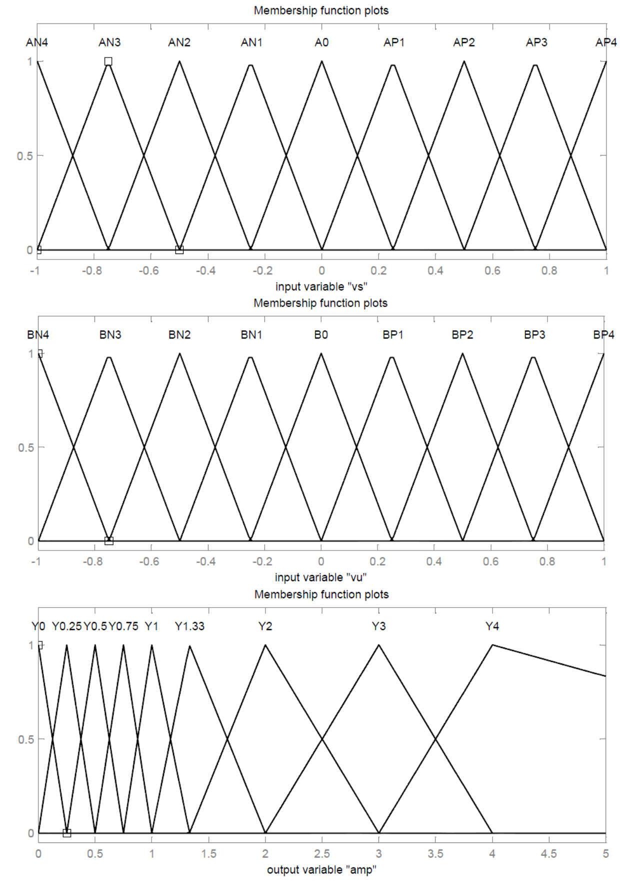

Fig. 3. Fuzzy membership function partition

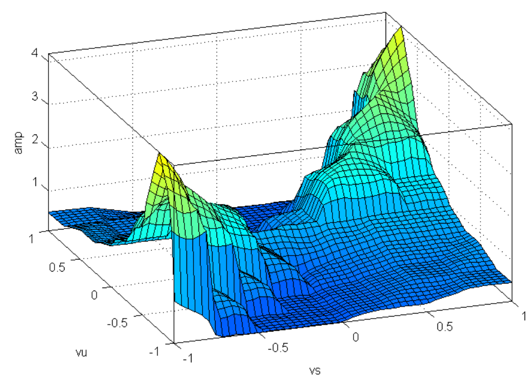

Fig. 4. Fuzzy control surface between variables

Regarding Eq. (10), it must be noted that an increase in n generally brings more separation between the road disturbance and the sprung mass, but at the same time increases the amount of control effort.

5. SIMULATION RESULTS

Simulation is conducted using the vehicle simulation program Carsim and Simulink. In the first scenario, the vehicle goes over a speed bump with the height of 20 cm, and the longitudinal vehicle speed is maintained at 40 km/h. This is immediately followed by a double lane change at the same speed in order to see the suspension response caused by the driver steering maneuver.

The second scenario focuses more in terms of the frequency variation than in terms of time domain chassis movement. The sinusoidal road disturbance that has an 180° phase difference between the left and the right wheels is applied to the vehicle, and its frequency is increased from 0.1 to 5 Hz.

For the simulation, the following constants are used.

$m_s = 342.5$ kg; $m_u = 40$ kg; $k_s = 15300$ N/m;

k_t=230000 N/m; $b_s = 100$ N·s/m;

$b_a = 5000$ N·s/m; $n = 1$

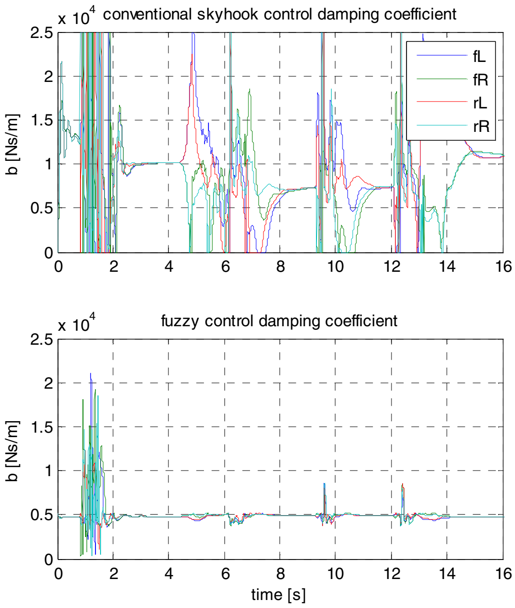

Fig. 5. Damping coefficient of each corner on a speed bump and during DLC

Figure 5 shows that the operation range of the damping coefficient for the conventional skyhook control is more than ten times that for the fuzzy control design. This certainly indicates that much harshness from the passenger perspective can be reduced by the proposed scheme.

This is proven in Figure 7 and 9, where much of the jerk involved in the conventional skyhook control is reduced. Additionally, Figure 6 and 8 show that the road disturbance separation performance has been well-maintained.

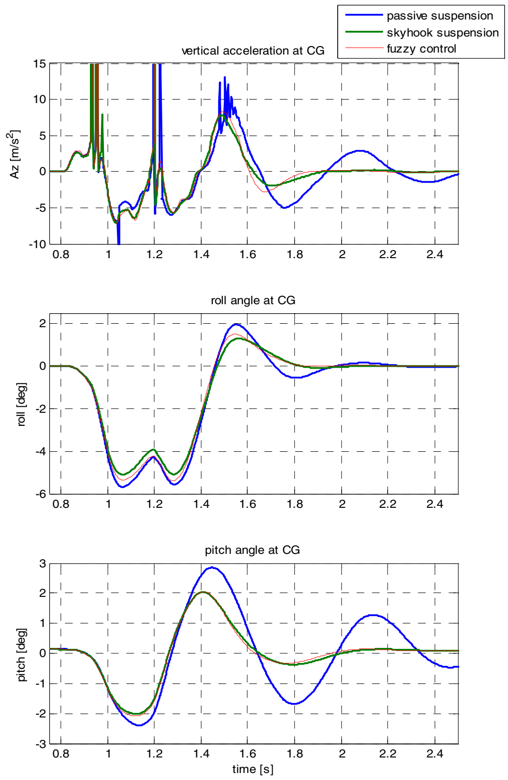

Fig. 6. a_z , roll and pitch while going over a speed bump

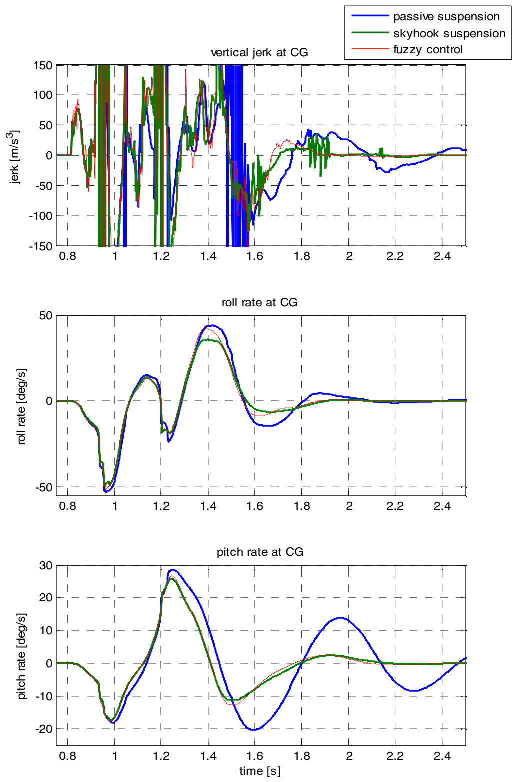

Fig. 7. Jerk, p and q while going over a speed bump

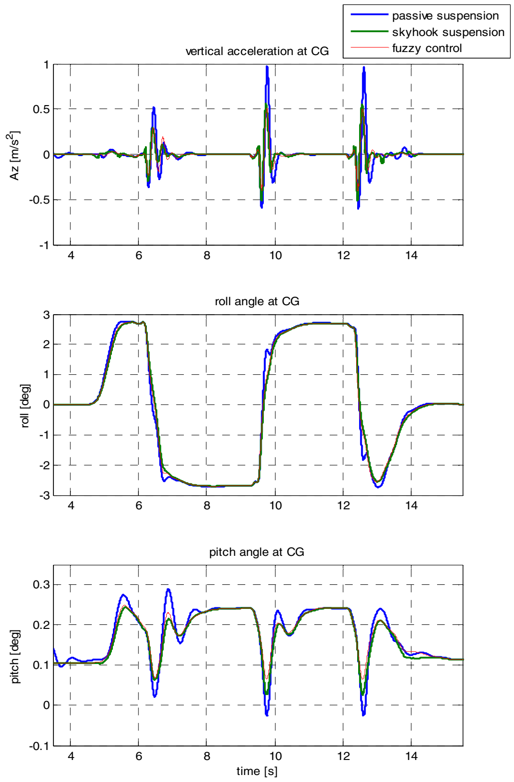

Fig. 8. a_z , roll and pitch during DLC

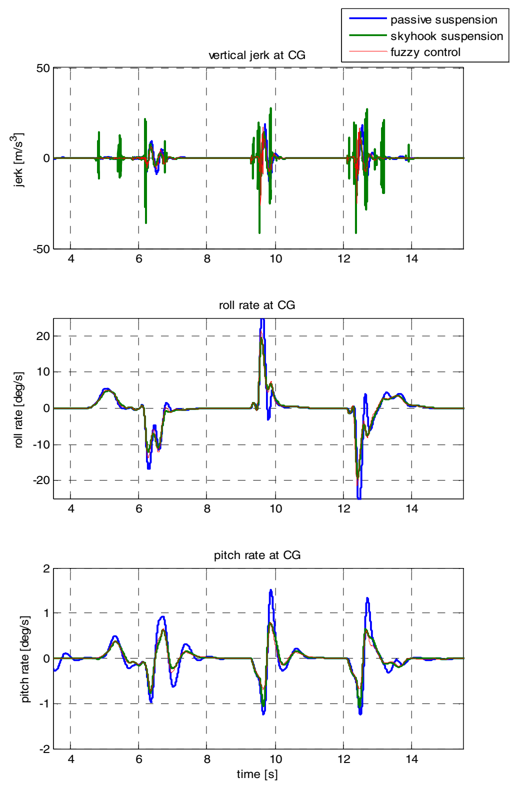

Fig. 9. Jerk, p and q during DLC

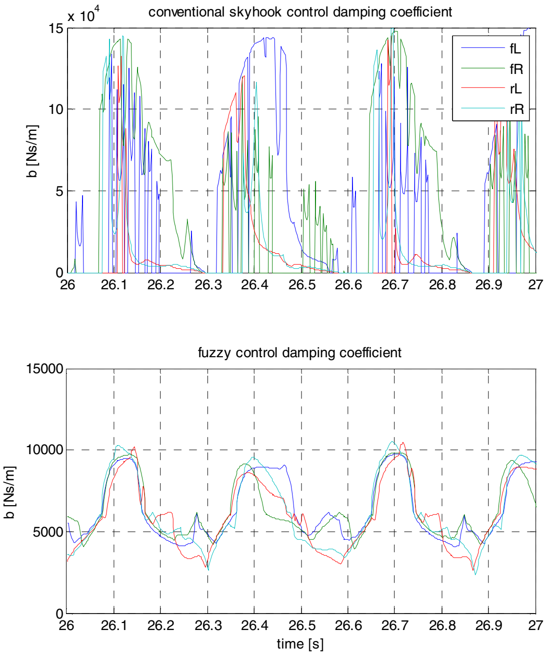

Fig. 10. Damping coefficient of each corner on the cross-sine wave road

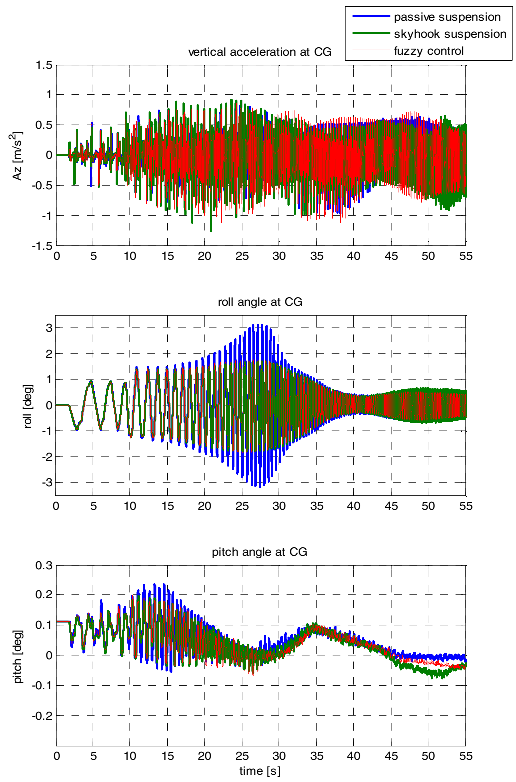

Fig. 11. a_z , roll and pitch on the cross-sine wave road

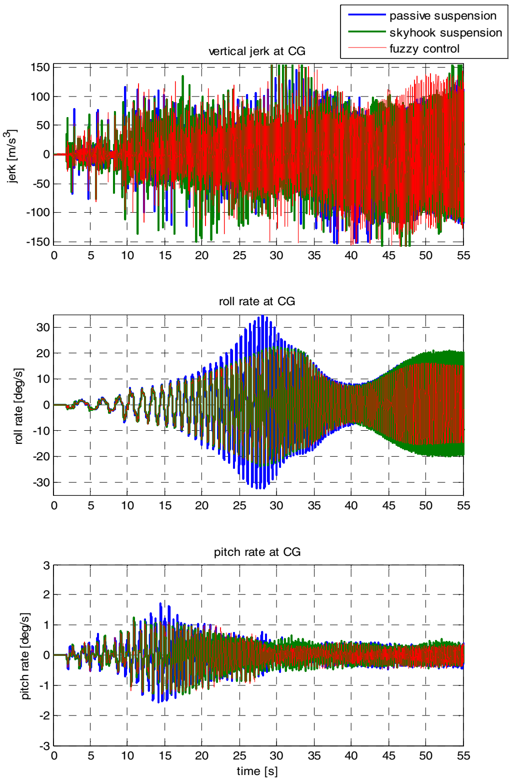

Fig. 12. Jerk, p and q during DLC on the cross-sine wave road

Figure 10 displays the damping coefficient operation range at a magnified portion of the second simulation scenario. Not only the peak-to-peak value of the fuzzy control damping coefficient is less than that of the conventional skyhook control damping coefficient, but also significant amount of chattering involved in the conventional scheme is removed in the modified scheme. At the same time, Figure 11 and 12 both show that the ride qualities are made more pleasant for the passenger.

By setting a condition dependent on $\hat{v}_s - \hat{v}_u$, it is actually possible to reduce the control input variation for the case of the conventional design as well. However, it was verified that the performance of the conventional skyhook suspension system with the implementation of such condition is far worse than that of the proposed scheme while the control input is still more vigorously varying.

6. CONCLUSION

This paper presents a newly designed skyhook damper control scheme aided by the sensor measurements of the suspension jounce, vehicle acceleration, and angular rates at C.G. As planned, the designed fuzzy logic skyhook damper controller effectively reduces the jerk and the excessive control effort, and at the same time exhibits a similar or enhanced level of performance compared to the conventional skyhook control scheme.

A noteworthy reduction of the amount of control effort required to guarantee the similar or better level of semi-active suspension performance leaves an opportunity for the further enhancement of the vehicle noise, vibration and harshness.

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