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Vehicle Chassis Control Using Adaptive Semi-Active Suspension *

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Abstract: This paper presents an adaptive semi-active control strategy to improve the stability and performance of a light commercial vehicle equipped with four continuously varying dampers. A choice between ride comfort or road holding of the vehicle is made automatically using a rule based adaptive algorithm based on various factors such as roll rate and yaw rate. The damping factor or the controller configuration of each damper is modified using a rule based adaptive algorithm and this technique is named Individual Damping Control (IDC) in this paper. The vehicle roll and yaw stability are analyzed using this technique. Simulation results on a high-fidelity realistic computer model of a light commercial vehicle are presented to validate the proposed technique.

1. INTRODUCTION

Semi-active suspension system is used in automobiles to improve ride comfort and road holding and these can be achieved using control methods like sky-hook [Karnopp et al., 1974] and ground-hook [Valasek et al., 1997] m respectively. It is a well known fact that ride comfort and road holding performance cannot be improved simultaneously. The requirement of good ride comfort and road holding varies based on several factors. For example, on snowy roads, vehicle handling is more important than comfort since poor road holding affects the stability of the vehicle. Thus, automatic switching between these two performance goals is an ideal solution to utilize the semi-active suspension effectively.

Hybrid control which is the combination of sky-hook and ground-hook methods can solve this problem. A simple hybrid control algorithm can be written as

$$F_{sa} = \beta F_{grd} + (1 - \beta) F_{sky}$$

where, F_{sa}, F_{sky}, F_{grd} are semi-active, sky-hook and ground-hook damper forces respectively and β is a factor which mixes the two control forces.

The main aim of this paper is to propose a rule based adaptive control strategy which controls each semi-active suspension damper separately to improve the stability and comfort of the vehicle using the most adequate compromise solution. This can be achieved either by changing the damping factor of each damper or by changing the controller configuration of each damper from sky-hook to ground-hook or by changing the β value of each damper in the hybrid controller or a combination of all. To proceed with this idea, different parameters and various simulations have been carried out in this paper. The sky-hook and ground-hook controllers are designed based on quarter car models. The necessity to improve vehicle handling is identified using roll rate and yaw rate.

The rest of the paper is organized as follows. Section 2 introduces the mathematical model of the full car and a quarter car model which are used to design individual suspension controllers. Section 3 presents the design of the rule based adaptive algorithm. Simulation results and conclusions are presented in the subsequent sections.

2. MATHEMATICAL MODELS

2.1 Full car suspension model

The full car suspension model as in [Sankaranarayanan et al., 2007], consists of a sprung mass (actual car body), three unsprung masses in which two are front tires and a single rear axle which connects the two rear tires. The whole system has 7 degrees of freedom and those are vertical motion of the sprung mass z, roll motion of the sprung mass ϕ, pitch motion of the sprung mass θ, vertical motion of the two front unsprung masses z_{11}, z_{12}, vertical motion of the rear unsprung mass z_{ur} and the roll motion of the rear unsprung mass ϕ_{ur}. The dynamics equation of the suspension model can be expressed as

$$\begin{aligned}
M\ddot{z} &= -f_{s11} - f_{s12} - f_{s21} - f_{s22} \\
&\quad - f_{d11} - f_{d12} - f_{d21} - f_{d22} \\
I_{yy}\ddot{\theta} &= a(f_{s11} + f_{s12} + f_{d11} + f_{d12}) \\
&\quad - b(f_{s21} + f_{s22} + f_{d21} + f_{d22}) \\
I_{xx}\ddot{\phi} &= c(f_{s11} + f_{s21} + f_{d11} + f_{d21}) \\
&\quad - d(f_{s12} + f_{s22} + f_{d12} + f_{d22}) \\
m_{11}\ddot{z}_{11} &= f_{s11} + f_{d11} - f_{t11}
\end{aligned}$$

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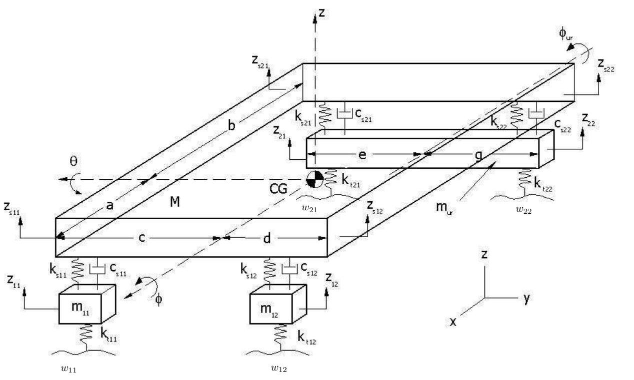

Fig. 1. Full car suspension model

$$m_{12}\ddot{z}_{12} = f_{s12} + f_{d12} - f_{t12}$$
$$M_{ur}\ddot{z}_{ur} = f_{s21} + f_{s22} + f_{d21} + f_{d22} - f_{t21} - f_{t22}$$
$$I_{ur}\ddot{\phi}_{ur} = -e(f_{s21} + f_{d21} - f_{t21}) + g(f_{s22} + f_{d22} - f_{t22})$$

The kinematic equations can be written as

$$\begin{pmatrix} z_{s11} \\ z_{s12} \\ z_{s21} \\ z_{s22} \\ z_{21} \\ z_{22} \end{pmatrix} = \begin{pmatrix} z \\ z \\ z \\ z \\ z_{ur} \\ z_{ur} \end{pmatrix} + \begin{pmatrix} -a & -c & 0 \\ -a & d & 0 \\ b & -c & 0 \\ b & d & 0 \\ 0 & 0 & -e \\ 0 & 0 & g \end{pmatrix} \begin{pmatrix} \sin\theta \\ \sin\phi \\ \sin\phi_{ur} \end{pmatrix}$$

The spring forces are

$$f_{sij} = k_{sij}(z_{sij} - z_{ij}) \quad i,j = 1,2$$

The semi-active damping forces are

$$f_{dij} = c_{sij}(t)(\dot{z}_{sij} - \dot{z}_{ij}), \quad i,j = 1,2,$$
$$c_{sij}(t) \in (c_{min}, c_{max}) \subset I\!R^+$$

The spring forces due to tires are

$$f_{tij} = k_{tij}(z_{ij} - w_{ij}), \quad i,j = 1,2$$

where,
11 - Front right (FR)
12 - Front left (FL)
21 - Rear right (RR)
22 - Rear left (RL)
z - Vertical displacement of the sprung mass
z_{s**} - Vertical displacement of the individual sprung masses
z_{**} - Vertical displacement of unsprung masses
θ - Pitch angle
ϕ - Roll angle
ϕ_{ur} - Roll angle of the unsprung mass
M - Mass of the sprung mass
I_{xx} - Inertia of the sprung mass with respect to x axis
I_{yy} - Inertia of the sprung mass with respect to y axis
m_{**} - Mass of the front unsprung masses
M_{ur} - Mass of the rear single unsprung mass
I_{ur} - Inertia of the unsprung mass
f_{s**} - Spring forces
f_{d**} - Damping forces
f_{t**} - Spring force due to tires
k_{s**} - Spring constants
$c_{s**}(t)$ - Damping factors
k_{t**} - Spring constants of the tires
w_{ij} - Road inputs
CG - Center of gravity of the sprung mass
a, b - Distance from CG to front and rear respectively
c, d - Distance from CG to right and left respectively
e, g - Distance from Center of gravity of the unsprung mass to right and left respectively

2.2 Quarter car model

The linear dynamics of a quarter car model as shown in Fig. 2 can be written as

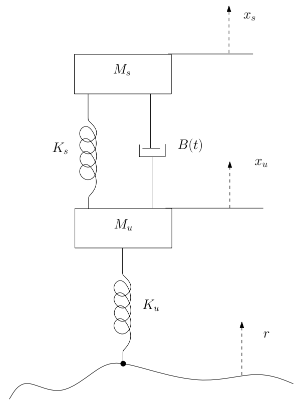

Fig. 2. Quarter car suspension model

$$M_s\ddot{x}_s + B(t)(\dot{x}_s - \dot{x}_u) + K_s(x_s - x_u) = 0$$
$$M_u\ddot{x_u} - B(t)(\dot{x}_s - \dot{x}_u) + K_s(x_u - x_s) + K_u(x_u - r) = 0$$

where,
x_s - Position of the sprung mass, x_u - Position of the unsprung mass, M_s - Mass of the sprung mass, M_u - Mass of the unsprung mass, $0 < B_{min} \leq B(t) \leq B_{max}$ - Varying damping coefficient, K_s - Spring constant of the suspension spring, K_u - Spring constant of the tire, r - Road disturbance. Further the semi-active suspension force can be defined as $F_{sem} \triangleq B(t)(\dot{x}_s - \dot{x}_u)$

3. SEMI-ACTIVE SUSPENSION CONTROL STRATEGIES

3.1 Sky-hook

The aim of the sky-hook control technique of Karnopp et al. [1974] is to minimize the vertical motion of the sprung mass by connecting a virtual damper between the body and the sky as shown in Fig. 3, hence named sky-hook. In practice, since it is not possible to connect a damper between the body and the sky, the adjustable damper is approximated to mimic the virtual damper. The necessary damping force can represented as

$$F_{sky} = \begin{cases} C_{sky}\dot{x}_s & \text{if } \dot{x}_s(\dot{x}_s - \dot{x}_u) \geq 0 \\ C_{min}(\dot{x}_s - \dot{x}_u) & \text{if } \dot{x}_s(\dot{x}_s - \dot{x}_u) < 0 \end{cases}$$

This technique certainly improves the ride comfort but may lead to poor handling.

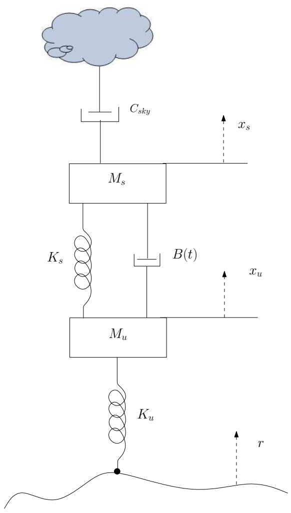

Fig. 3. Sky-hook control

3.2 Ground-hook

The ground-hook control algorithm is developed to reduce the tire motion, that is, to improve road holding (handling), by virtually connecting a damper between the ground and the tire as shown in Fig. 4. Since this is not

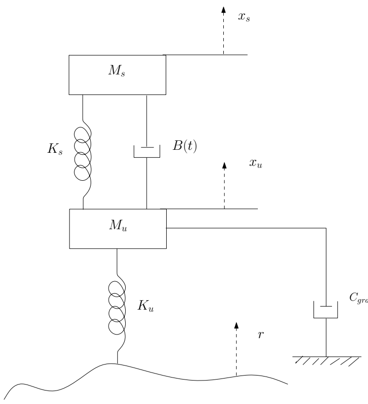

Fig. 4. Ground-hook control

possible in practice, the adjustable damping force that is necessary can be expressed as

$$F_{grd} = \begin{cases} C_{grd}\dot{x}_u & \text{if } -\dot{x}_u(\dot{x}_s - \dot{x}_u) \geq 0 \\ C_{min}(\dot{x}_s - \dot{x}_u) & \text{if } -\dot{x}_u(\dot{x}_s - \dot{x}_u) < 0 \end{cases}$$

3.3 Hybrid

The combination of sky-hook and ground-hook is called hybrid control technique and the corresponding semi-active damping force can be written as

$$F_{hyb} \stackrel{\triangle}{=} B(t)(\dot{x}_s - \dot{x}_u) = \beta F_{sky} + (1-\beta)F_{grd} \qquad (1)$$

where, $C_{sky}, C_{grd} > C_{min} = B_{min} > 0, \quad \beta \in [0, 1]$. The value of β determines the contribution of the individual sky-hook and ground-hook control actions buried inside the hybrid control algorithm.

4. RULE BASED ADAPTIVE CONTROL STRATEGY

The aim of the rule based adaptive control strategy is to improve the stability or comfort based on the driving and road based requirements. To design such a strategy, various control methods have been studied in this section using standard maneuvers. The roll stability and yaw stability of the vehicle are investigated using fish-hook and μ-split maneuvers respectively. Semi-active suspensions cannot actively stabilize the roll and yaw motions but they can improve the performance of the active controllers such as anti-roll bars and Electronic Stability Program (ESP). This section dedicated to study the roll and yaw stability improvement using various controller configuration.

4.1 Controller configuration

Let the damping factor of each damper be represented as

$$B(t)_{FL}, \quad B(t)_{FR}, \quad B(t)_{RL}, \quad B(t)_{RR}$$

where

$$0 < B_{min} < B(t) < B_{max}.$$

Similarly, the β value of individual dampers can be expressed as

$$\beta(t)_{FL}, \quad \beta(t)_{FR}, \quad \beta(t)_{RL}, \quad \beta(t)_{RR}$$

where

$$\beta(t) \in [0, 1]$$

(1) Configuration-1
All passive dampers, that is, $B(t)_{FL} = B(t)_{FR} = B(t)_{RL} = B(t)_{RR} = B_{pas} \stackrel{\triangle}{=} \frac{B_{min}+B_{max}}{2}$
(2) Configuration-2
All soft dampers, that is, $B(t)_{FL} = B(t)_{FR} = B(t)_{RL} = B(t)_{RR} = B_{min}$
(3) Configuration-3
All hard dampers, that is, $B(t)_{FL} = B(t)_{FR} = B(t)_{RL} = B(t)_{RR} = B_{max}$
(4) Configuration-4
All sky-hook, that is, $\beta(t)_{FL} = \beta(t)_{FR} = \beta(t)_{RL} = \beta(t)_{RR} = 1$ in (1)
(5) Configuration-5
All ground-hook, that is, $\beta(t)_{FL} = \beta(t)_{FR} = \beta(t)_{RL} = \beta(t)_{RR} = 0$ in (1)
(6) Configuration-6
IDC-1(for roll stability)

$$B_{FR}(t) = B_{RR}(t) = \left[\frac{1 + sign(\dot{\phi} - \dot{\phi}_{set})}{2}\right] B_{max} + \left[\frac{1 - sign(\dot{\phi} + \dot{\phi}_{set})}{2}\right] B_{min} + \left[\frac{1 + sign(\dot{\phi}_{set} - |\dot{\phi}|)}{2}\right] B_{pas}$$

$$B_{FL}(t) = B_{RL}(t) = \left[\frac{1 + sign(\dot{\phi} - \dot{\phi}_{set})}{2}\right] B_{min} + \left[\frac{1 - sign(\dot{\phi} + \dot{\phi}_{set})}{2}\right] B_{max} + \left[\frac{1 + sign(\dot{\phi}_{set} - |\dot{\phi}|)}{2}\right] B_{pas}$$

(7) Configuration-7
IDC-2(for yaw stability)

$$B_{FL}(t) = B_{RR}(t) = \left[\frac{1 + sign(r - r_{set})}{2}\right] B_{max}$$
$$+ \left[\frac{1 - sign(r + r_{set})}{2}\right] B_{min} + \left[\frac{1 + sign(r_{set} - |r|)}{2}\right] B_{pas}$$
$$B_{FR}(t) = B_{RL}(t) = \left[\frac{1 + sign(r - r_{set})}{2}\right] B_{min}$$
$$+ \left[\frac{1 - sign(r + r_{set})}{2}\right] B_{max} + \left[\frac{1 + sign(r_{set} - |r|)}{2}\right] B_{pas}$$

IDC1

This configuration is designed to improve the roll stability by individually changing the damping coefficient of the vehicle based on the roll rate. If the roll rate is more than the threshold value and positive, that is, in the clock-wise direction with respect to the x-axis, as shown in Fig. 5, the right side dampers are made hard and left side dampers are made soft and visa versa for negative roll rate.

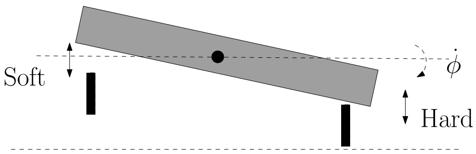

Fig. 5. IDC1

IDC2

The aim of IDC2 is to reduce the yaw rate of the vehicle using semi-active suspension dampers. The damping factor of the diagonal dampers are modified based on the yaw rate as shown in Fig. 6.

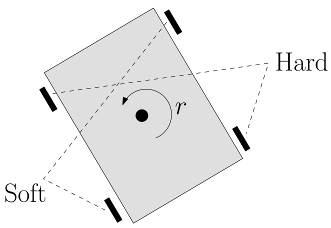

Fig. 6. IDC2

4.2 Roll stability

Roll over possibilities are more critical in commercial vehicles, especially in heavy trucks, which can be avoided using anti-roll bars. Semi-active suspension can also help prevent roll over accidents to some extent as transient roll behaviour can be improved. Properly designed semi-active suspension system can improve the roll stability compared with passive suspension. The main cause of roll over is the high speed with sudden steering of the vehicle. This situation can be detected with the steering angular rate together with speed of the vehicle or roll rate.

In this paper, the fish-hook maneuver is considered for roll stability analysis using various controller configuration presented in the previous section. The actual vehicle's parameters are entered into a high-fidelity, realistic vehicle simulator for this purpose. Fish-hook maneuver is tested with all the controller configurations. The simulation results are presented in Figs. 7, 8.

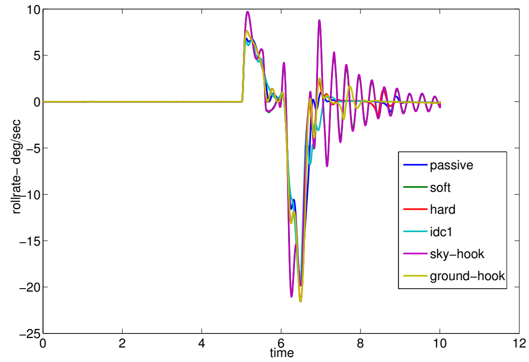

Fig. 7. Roll rate in fish-hook maneuver

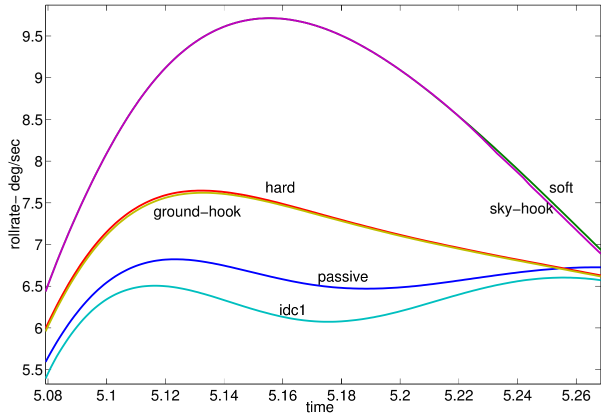

Fig. 8. Roll rate in fish-hook maneuver - zoomed

It can be seen from the figures that, IDC1 significantly reduce the roll rate of the vehicle in the fish-hook maneuver.

4.3 Yaw stability

Yaw stability of the vehicle is improved using ESP (Electronic Stability Program) which conventionally uses individual braking concept or recently using active steering or steer-by-wire. In this section, the yaw stability of the vehicle using steer-by-wire together with IDC2 is investigated.

Yaw stability controllers have been studied using steer-by-wire methods recently by [Oncu et al., 2007], [Karaman et al., 2006]. An ESP controller implementation approach based on the disturbance observer is shown in Fig. 9. IDC2 is tested in the vehicle with μ-split maneuver and the results are compared with passive suspension without ESP, with ESP and ESP+IDC2. The simulation results are presented in Figs. 10, 11. From the figures, it can be easily seen that, IDC2 improves the yaw stability of the ESP controller as compared with ESP alone.

4.4 Supervisory controller

Based on various controller configuration and simulation results, it is clearly seen that, IDC1 works better to improve the roll stability and IDC2 works better to improve the yaw stability. The aim is to design a rule based adaptive algorithm which switches between the control strategies in subsection 4.1 to achieve the necessary combination of comfort and holding. In other words, the algorithm has to switch to sky-hook for comfort, IDC1 for roll stability

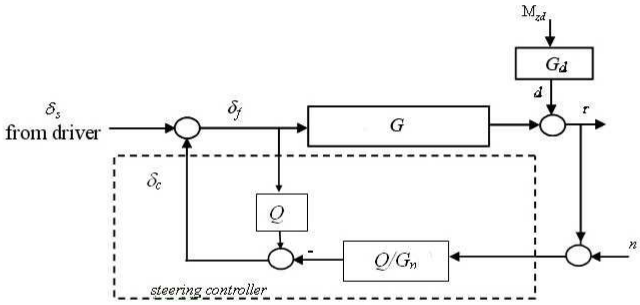

Fig. 9. Yaw stability control using steer-by-wire and disturbance observer

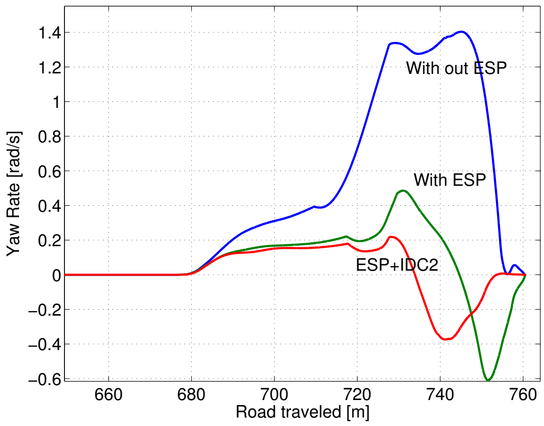

Fig. 10. Yaw rate comparison

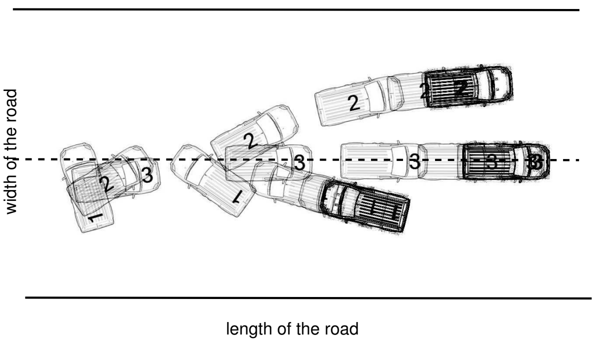

Fig. 11. Stroboscopic plots. 1-without ESP, 2-with ESP, 3-ESP+IDC2

and IDC2 for yaw stability. This can be represented as in a block diagram form as shown in Fig. 12 and can be written as a formula as follows

$$F_{sem} = \begin{cases} IDC1 & \text{if} \quad |\dot{\phi}| > \dot{\phi}_{set} \;\; \& \;\; |\dot{\phi}| > |r| \\ IDC2 & \text{if} \quad |r| > r_{set} \;\; \& \;\; |r| > |\dot{\phi}| \\ F_{sky} & \text{otherwise} \end{cases} \quad (2)$$

To simulate the rule based adaptive controller, a test road is created using the same software which is used for simulation. This road consists of road bumps to test the suspension for comfort, and a sudden turn and a μ-split maneuver to test handling of the vehicle. The simulation results with rule based adaptive controller are compared with passive suspension and the results are presented in Figs.14-17. The ride comfort of the vehicle is improved

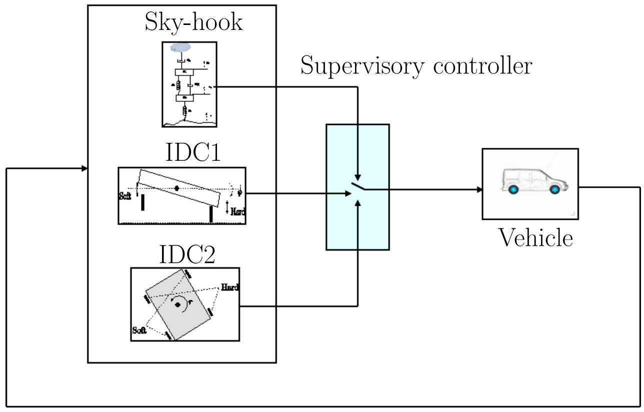

Fig. 12. Adaptive algorithm

whenever roll rate or yaw rate is more than the threshold value. It can be seen from Fig. 17 that, up to 12 sec, the sky-hook control force is applied since there is no roll rate and yaw rate at this time which can be seen from Figs.15, 16. But from 12 to 22 sec, the vehicle handling is improved through both IDC1 and IDC2 based in the magnitude of the roll and yaw rate. Similarly from 41 to 45 sec, the handling is improved. The roll rate and yaw rate of the vehicle is improved whenever the supervisory controller also called IDC controller switches from sky-hook to either IDC1 or IDC2 which can be clearly seen from Figs. 15, 16. The body acceleration is reduced whenever the controller switched to sky-hook which can be seen from Fig. 14.

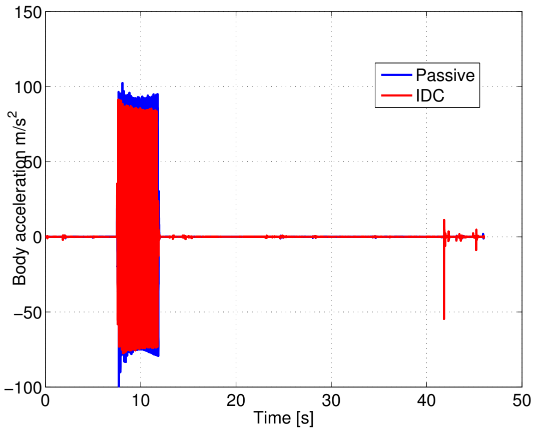

Fig. 13. Body acceleration

5. CONCLUSION

A rule based adaptive semi-active suspension controller algorithm is developed to improve the ride comfort and road holding of a light commercial vehicle. Various semi-active suspension control algorithms are studied to design the adaptive controller. A new method has been proposed to improve the handling of the vehicle by changing the damping factor of each dampers based on roll rate and yaw rate which is named Individual Damping Control. The rule based adaptive controller automatically switches between controllers to improve either ride comfort or road holding. The simulation results shows that, the proposed adaptive controller has better performance compared to a passive suspension system.

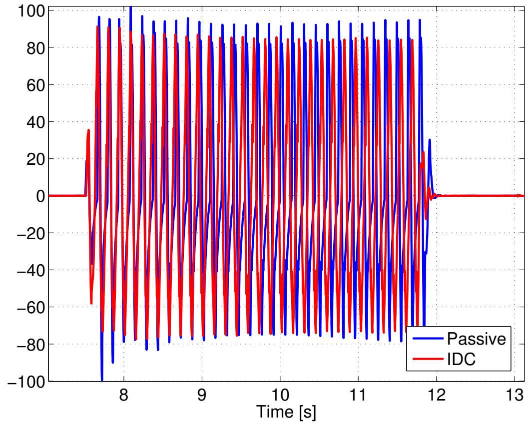

Fig. 14. Body acceleration-zoomed

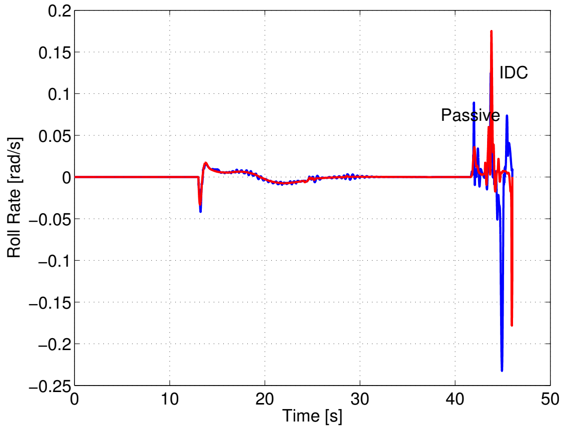

Fig. 15. Roll rate

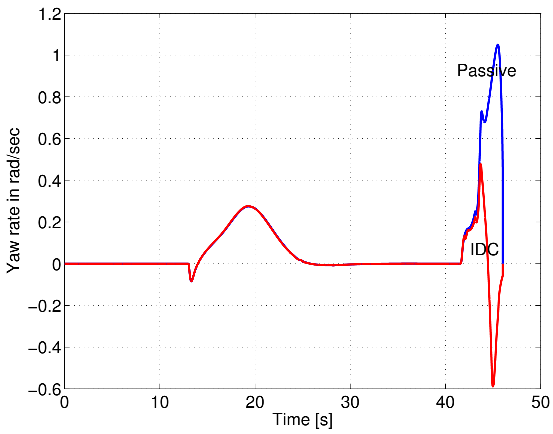

Fig. 16. Yaw rate

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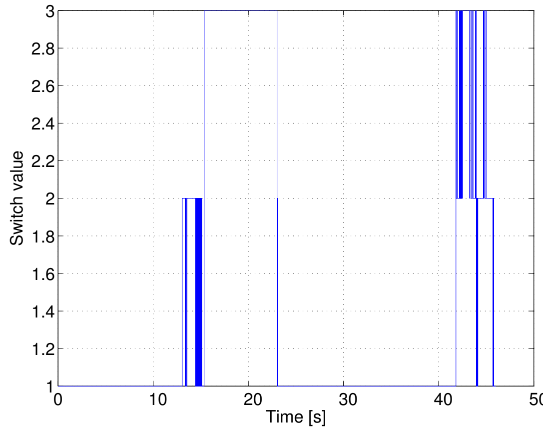

Fig. 17. Switch value. 1-Sky-hook, 2-IDC1, 3-IDC2