Integral sliding mode control and its application on Active suspension system

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Abstract–Active suspension system is a hot issue in the recent research of automotive industry, which is to generate active force to suppress the variation of the vehicle and improve safety and comfort for the passengers. By observing the dynamics of the vehicle, the controller output forces through linear actuators. Integral sliding mode control is applied here to calculate the output forces and then reject the disturbance. In this paper, an accurate nonlinear active suspension system model is developed first; decoupling of the sprung mass subsystem is investigated hereafter to isolate the dependence of the variables, then integral sliding mode control is used to reject the disturbance and reach the referenced surface for each subsystem. PID control, as a conventional control method, is compared here with the integral sliding mode control. The simulation results show the effectiveness of the proposed control method.

Keywords–Active suspension system, Nonlinear model, Integral sliding mode control

I. INTRODUCTION

Active suspension system has been focused by academic research and industrial application, which is supposed to eliminate or minimize vibrations greatly which ranged from 0.5 to 80 Hz [1], then isolate the vehicle body from road irregularities.

The active suspension system is a complex nonlinear system. The electrical active suspension system with linear actuators is proposed with the advantages of less mechanical connection and flexible control. Martins et al. (2006) proposed the potential of electrical active suspension system, and specified the "reduced comfort boundary" with one-third active force of the weight of the sprung mass [2]. Ikenaga et al (2000) has simplified the complicated full-car active suspension system into a linear model with filtered feedback control scheme under small vibration of the sprung mass.[3] An accurate full car ride model using model reducing techniques is proposed, and linearization of a full car MBD (multi-body dynamics) model is developed to obtain a large-order vehicle model. [4] However, linearization of the full vehicle is only effective based on the assumption that the vertical vibration is restricted within a small range, which is hard to satisfy under moderate road irregularities. An accurate nonlinear full car ride model is necessary to develop for accurate control of the actual car suspension system. A nonlinear active suspension system with four electro-hydraulic actuators is proposed by Chamseddine (2006) to illustrate the dynamic performances of the hydraulic active suspension system. [5]

Various control methods for active suspension system with electromagnetic actuator have been investigated. Multi-objective optimal control strategy is pursued for finding feedback control laws via mixed H_2/H_∞ control [6]. A simple and effective linear model following control method is used to reduce the calculation burden [7]. For the purpose of fault diagnosis and tolerance, sliding mode controller (SMC) had been developed by Yagiz (2000) for a nonlinear vehicle model [8]; a modified SMC was designed for a linear full vehicle active suspension system with partly knowledge of the system states [9]. For the conventional SMC method, the desired dynamic can be obtained only when the sliding mode occurs. A novel type PID type sliding mode control, so-called integral sliding mode control (ISMC) is introduced by Utkin [10], in which the sliding mode occurs at the initial instant and therefore the robustness of the system can be guaranteed during the entire process.

In this paper, an accurate nonlinear model is proposed in Section 2, to describe the dynamic response of the active suspension system because the dynamic of the vertical vibration of the car body is beyond the assumption of the linear model. A brief introduction of integral sliding mode control method is presented in Section 3, and its application on active suspension system is developed to reject road irregularities and parameter variations. Simulation results against road irregularities and parameter variations are shown in Section 4, to verify the proposed ISMC controller. In last section, the conclusions are given.

II. NONLINEAR MODEL OF ACTIVE SUSPENSION SYSTEM

The nonlinear model of the active suspension system is a 7-DOF system. Appling the principle of force and torque balance, with the consideration of the gravity of the sprung mass and unsprung mass and the fact of rigid body, the model is summarized as follow:

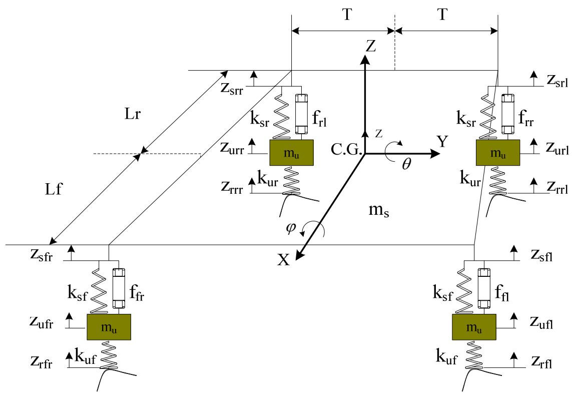


Fig.1: Model of the active suspension system

$$
\begin{aligned}
M_s\ddot{z} &= -\left(2K_{sf}+2K_{sr}\right)z-\left(2K_{sf}L_f-2K_{sr}L_r\right)\sin\theta+K_{sf}\left(z_{ufl}+z_{ufr}\right)+K_{sr}\left(z_{url}+z_{urr}\right)\\
&\quad -M_sg+\cos\theta\cos\varphi\left(F_{fl}+F_{fr}+F_{rl}+F_{rr}\right)\\
I_p\ddot{\theta} &= -\left(2K_{sf}L_f-2K_{sr}L_r\right)\cos\theta z-\left(2K_{sf}L_f^{\,2}+2K_{sr}L_r^{\,2}\right)\cos\theta\sin\theta+K_{sf}L_f\cos\theta\left(z_{ufl}+z_{ufr}\right)\\
&\quad -K_{sr}L_r\cos\theta\left(z_{url}+z_{urr}\right)+L_f\cos^2\theta\cos\varphi\left(F_{fl}+F_{fr}\right)-L_r\cos^2\theta\cos\varphi\left(F_{rl}+F_{rr}\right)\\
I_r\ddot{\varphi} &= -\left(2K_{sf}+2K_{sr}\right)T^2\cos\varphi\sin\varphi+K_{sf}T\cos\varphi\left(z_{ufl}-z_{ufr}\right)+K_{sr}T\cos\varphi\left(z_{url}-z_{urr}\right)\\
&\quad +T\cos\theta\cos^2\varphi\left(F_{fl}-F_{fr}\right)+T\cos\theta\cos^2\varphi\left(F_{rl}-F_{rr}\right)\\
M_u\ddot{z}_{ufl} &= K_{sf}z+K_{sf}L_f\sin\theta+K_{sf}T\sin\varphi-\left(K_{uf}+K_{sf}\right)z_{ufl}+K_{uf}z_{rfl}-M_ug-F_{fl}\cos\theta\cos\varphi\\
M_u\ddot{z}_{ufr} &= K_{sf}z+K_{sf}L_f\sin\theta-K_{sf}T\sin\varphi-\left(K_{uf}+K_{sf}\right)z_{ufr}+K_{uf}z_{rfr}-M_ug-F_{fr}\cos\theta\cos\varphi\\
M_u\ddot{z}_{url} &= K_{sr}z-K_{sr}L_r\sin\theta+K_{sr}T\sin\varphi-\left(K_{ur}+K_{sr}\right)z_{url}+K_{ur}z_{rrl}-M_ug-F_{rl}\cos\theta\cos\varphi\\
M_u\ddot{z}_{urr} &= K_{sr}z-K_{sr}L_r\sin\theta-K_{sr}T\sin\varphi-\left(K_{ur}+K_{sr}\right)z_{urr}+K_{ur}z_{rrr}-M_ug-F_{rr}\cos\theta\cos\varphi
\end{aligned}
$$
(1)

where, z,θ,φ are the heave position, pitch angle and roll angle of the sprung mass; $z_{ufl},z_{ufr},z_{url},z_{urr}$ are the unsprung mass heights of four corners; $z_{rfl},z_{rfr},z_{rrl},z_{rrr}$ are the terrain disturbance heights of four corners; M_s, M_u are the sprung mass and unsprung mass respectively; $K_{sf},K_{sr},K_{uf},K_{ur}$ are the stiffness of front springs, rear springs, front tires and rear tires, respectively; I_p, I_r are the pitch axis and roll axis moment of inertia; L_f, L_r are the distance from C.G. to the front axle and rear axle, T is the half track width of the sprung mass; g is the acceleration due to gravity; $F_{fl},F_{fr},F_{rl},F_{rr}$ are the active forces generated by the electromagnetic actuators.

Assume that the nominal values of mass and inertia of sprung mass subsystem are M_{s0},I_{p0},I_{r0} the range of the variation of the parameters of the active suspension system are bounded as follows

$$M_{s\min}<M_s=M_{s0}+\Delta M_s<M_{s\max}$$
$$I_{p\min}<I_p=I_{p0}+\Delta I_p<I_{p\max}$$
$$I_{r\min}<I_r=I_{r0}+\Delta I_r<I_{r\max}$$

where $\Delta M_s,\Delta I_p,\Delta I_r$ are their deviations, M_s,I_p,I_r are the actual values.

The nominal model of the sprung mass subsystem is obtained without unmodeled dynamic and external disturbances as follow:

$$
\begin{cases}
\ddot{z}=-\dfrac{2\left(K_{sf}+K_{sr}\right)}{M_{so}}z-\dfrac{2\left(K_{sf}L_f-K_{sr}L_r\right)}{M_{so}}\sin\theta+\dfrac{1}{M_{so}}u_z\\
\ddot{\theta}=-\dfrac{2\left(K_{sf}L_f-K_{sr}L_r\right)\cos\theta}{I_{po}}z-\dfrac{2\left(K_{sf}L_f^{\,2}+K_{sr}L_r^{\,2}\right)\cos\theta}{I_{po}}\sin\theta+\dfrac{1}{I_{po}}u_\theta\\
\ddot{\varphi}=-\dfrac{2\left(K_{sf}+K_{sr}\right)T^2\cos\varphi}{I_{ro}}\sin\varphi+\dfrac{1}{I_{ro}}u_\varphi
\end{cases}
\quad (2)
$$

where $u_z=\left(F_{fl}+F_{fr}+F_{rl}+F_{rr}\right)\cos\theta\cos\varphi$,
$u_\theta=\left[L_f\left(F_{fl}+F_{fr}\right)-L_r\left(F_{rl}+F_{rr}\right)\right]\cos^2\theta\cos\varphi$,
$u_\varphi=T\left(F_{fl}-F_{fr}+F_{rl}-F_{rr}\right)\cos\theta\cos^2\varphi$.

III. INTEGRAL SLIDING MODE CONTROL AND ITS APPLICATION ON ACTIVE SUSPENSION SYSTEM

1. ISMC scheme

The design concept of ISMC is that a discontinuous term is added to the existing feedback controller for a nominal plant model to ensure the desired performance despite parametric uncertainty and external disturbances. The design procedure of ISMC [10] is described briefly as follow.

An actual system with uncertainty conditions is

$$\dot{x}=f(x)+B(x)u+h(x,t) \quad (3)$$

and its ideal closed-loop system is

$$\dot{x}_0=f(x_0)+B(x_0)u_0 \quad (4)$$

where $x\in R^n$ is the state vector and $u\in R^m$ is the control input vector, $B(x)$ is an appropriate matrix with $rankB(x)=m$, $h(x,t)$ represents uncertainty conditions such as parameter variations, unmodeled dynamics and external disturbances; x_0 is the state trajectory of the ideal system under ideal feedback control u_0 . Assume that the perturbation $h(x,t)$ is bounded and is of the form:

$$\left|h_i(x,t)\right|\le h_i^u(x,t),\ i=1,\cdots,n$$

where $h_i^u(x,t)$ is the upper bound of $h_i(x,t)$ and being known positive scalar functions.

To ensure the sliding surfaces from the initial time instant, i.e. $x(0)=x_0(0)$, the control law is redesigned as follow

$$u=u_0+u_d \quad (5)$$

with $u_d\in R^m$ is designed to reject the perturbation term $h(x,t)$.

An integral item is added to the conventional sliding manifold to consist the novel sliding surface

$$s=s_0(x)+s_d$$

with $s,s_0(x),s_d\in R^m$.

The first control part u_0 is to ensure the ideal system trajectory; the second control part u_d is to reject the perturbation and enforce the sliding manifolds that

$$\dot{s}=\dot{s}_d+\dot{s}_0(x)=\frac{\partial s_0}{\partial x}\left[f(x)+B(x)u_0+B(x)u_d+h(x,t)\right]+\dot{s}_d=0$$

The integral item is defined to meet the requirement $s(0)=0$

$$\dot{s}_d=-\frac{\partial s_0}{\partial x}\left[f(x)+B(x)u_0\right],\ s_d(0)=-s_0\left(x(0)\right)$$

The control law u_d is defined to enforce sliding mode along the manifold via discontinuous

$$u_d=-M(x)sign(s) \quad (6)$$

where $M(x)$ is the control gain and selected as scalar function or even a constant to simplify the control design.

The Lyapunov function is defined as $V=\frac{1}{2}s^Ts$, the time derivative of V is $\dot{V}=s^T\dot{s}$. The goal of the control is enforce the sliding mode to the desired performance, i.e. $s=0$, this lead V converge to zero, hence its time derivative $\dot{V}$ should be $\dot{V}\le 0$, the sliding surface s and its time derivative $\dot{s}$ have different sign. The time derivative of s can be calculated as

$$\dot{s}=\frac{\partial s_0}{\partial x}\left[B(x)u_d+h(x,t)\right]=\frac{\partial s_0}{\partial x}\left[-B(x)M(x)sign(s)+h(x,t)\right]<0$$

The above inequality is guaranteed by $M(x)>h^u(x,t)$,

i.e. $M_i(x)>h_i^u(x,t), i=1,\cdots,m$.

2. Application of ISMC on active suspension system

The main goal of the active damper system is to isolate the vehicle body from road irregularities. Hence, for the state equations of the system, the control goal is to keep the height, pitch and roll angle of the car body constant.

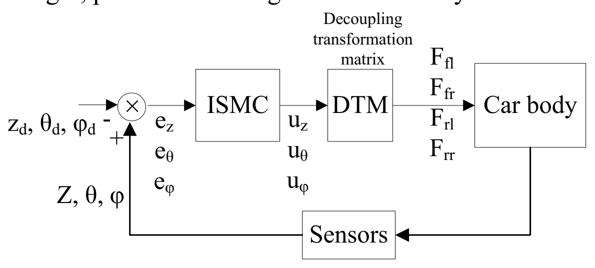


Fig.2: ISMC control scheme

Let z_d,θ_d,φ_d be the desired value of z,θ,φ , the error vector between the actual and desired values is defined as $e^T=\begin{bmatrix}z_e & \theta_e & \varphi_e\end{bmatrix}=\begin{bmatrix}z-z_d & \theta-\theta_d & \varphi-\varphi_d\end{bmatrix}$. The control law of the nominal sprung mass subsystem is demanded to converge the error vector e to zero at the well-designed trajectory, hence the closed-loop system obtains its desired dynamic performance. The ideal state-error trajectory of the nominal sprung mass subsystem is given

$$\begin{aligned}&\ddot{z}_e+k_{zd}\dot{z}_e+k_{zp}z_e=0\\&\ddot{\theta}_e+k_{\theta d}\dot{\theta}_e+k_{\theta p}\theta_e=0\\&\ddot{\varphi}_e+k_{\varphi d}\dot{\varphi}_e+k_{\varphi p}\varphi_e=0\end{aligned}\tag{7}$$

where $k_{zd},k_{zp},k_{\theta d},k_{\theta p},k_{\varphi p}$ and $k_{\varphi d}$ are the appropriate parameters to stable the plant and achieve the desired dynamic performance.

The first part of control vector u_0 is obtained from (2) and (5) as

$$u_0=\begin{bmatrix}u_{z0}\\u_{\theta 0}\\u_{\varphi 0}\end{bmatrix}=\begin{bmatrix}M_{s0}\begin{pmatrix}\frac{2(K_{sf}+K_{sr})}{M_{s0}}z+\frac{2(K_{sf}L_f-K_{sr}L_r)}{M_{s0}}\sin\theta\\+\ddot{z}_d-k_{zd}\dot{z}_e-k_{zp}z_e\end{pmatrix}\\I_{p0}\begin{pmatrix}\frac{2(K_{sf}L_f-K_{sr}L_r)\cos\theta}{I_{p0}}z+\frac{2(K_{sf}L_f^2+K_{sr}L_r^2)\cos\theta}{I_{p0}}\sin\theta\\+\ddot{\theta}_d-k_{\theta d}\dot{\theta}_e-k_{\theta p}\theta_e\end{pmatrix}\\I_{r0}\begin{pmatrix}\frac{2(K_{sf}+K_{sr})T^2\cos\varphi}{I_{r0}}\sin\varphi\\+\ddot{\varphi}_d-k_{\varphi d}\dot{\varphi}_e-k_{\varphi p}\varphi_e\end{pmatrix}\end{bmatrix}\tag{8}$$

The integral type sliding manifold of the sprung mass subsystem is defined as

$$s=\begin{bmatrix}s_z\\s_\theta\\s_\varphi\end{bmatrix}=\begin{bmatrix}s_{z0}+s_{zd}\\s_{\theta 0}+s_{\theta d}\\s_{\varphi 0}+s_{\varphi d}\end{bmatrix}\tag{9}$$

in which the nominal sliding surface s_0 is selected as

$$s_0=\begin{bmatrix}s_{z0}\\s_{\theta 0}\\s_{\varphi 0}\end{bmatrix}=\begin{bmatrix}\dot{z}_e+\lambda_z z_e\\\dot{\theta}_e+\lambda_\theta\theta_e\\\dot{\varphi}_e+\lambda_\varphi\varphi_e\end{bmatrix}$$

and the integral item s_d is designed to enforce the sliding mode at the initial time instant

$$\dot{s}_d=\begin{bmatrix}\dot{s}_{zd}\\\dot{s}_{\theta d}\\\dot{s}_{\varphi d}\end{bmatrix}=-\begin{bmatrix}\dot{s}_{z0}\\\dot{s}_{\theta 0}\\\dot{s}_{\varphi 0}\end{bmatrix}=\begin{bmatrix}-\ddot{z}_d+(k_{zd}-\lambda_z)\dot{z}_e+k_{zp}z_e\\-\ddot{\theta}_d+(k_{\theta d}-\lambda_\theta)\dot{\theta}_e+k_{\theta p}\theta_e\\-\ddot{\varphi}_d+(k_{\varphi d}-\lambda_\varphi)\dot{\varphi}_e+k_{\varphi p}\varphi_e\end{bmatrix},\begin{bmatrix}s_{zd}(0)\\s_{\theta d}(0)\\s_{\varphi d}(0)\end{bmatrix}=-\begin{bmatrix}s_{z0}(0)\\s_{\theta 0}(0)\\s_{\varphi 0}(0)\end{bmatrix}$$

The time derivative of sliding surface s is

$$\dot{s}=\begin{bmatrix}\dot{s}_z\\\dot{s}_\theta\\\dot{s}_\varphi\end{bmatrix}=\begin{bmatrix}\left(\frac{M_{so}}{M_s}-1\right)\left(\ddot{z}_d-k_{zd}\dot{z}_e-k_{zp}z_e\right)+\frac{1}{M_s}u_{zd}+\frac{1}{M_s}d_z\\\left(\frac{I_{po}}{I_p}-1\right)\left(\ddot{\theta}_d-k_{\theta d}\dot{\theta}_e-k_{\theta p}\theta_e\right)+\frac{1}{I_p}u_{\theta d}+\frac{1}{I_p}d_\theta\\\left(\frac{I_{ro}}{I_r}-1\right)\left(\ddot{\varphi}_d-k_{\varphi d}\dot{\varphi}_e-k_{\varphi p}\varphi_e\right)+\frac{1}{I_r}u_{\varphi d}+\frac{1}{I_r}d_\varphi\end{bmatrix}\tag{10}$$

To ensure that the well-defined Lyapunov function $V=\frac{1}{2}s^Ts$ has negative time derivative $\dot{V}=s^T\dot{s}\le 0$, the discontinuous control law is decided as

$$u_d=\begin{bmatrix}u_{zd}\\u_{\theta d}\\u_{\varphi d}\end{bmatrix}=\begin{bmatrix}M_{so}\left(-K_z z_e sign(z_e s_z)-M_z sign(s_z)\right)\\I_{po}\left(-K_\theta\theta_e sign(\theta_e s_\theta)-M_\theta sign(s_\theta)\right)\\I_{ro}\left(-K_\varphi\varphi_e sign(\varphi_e s_\varphi)-M_\varphi sign(s_\varphi)\right)\end{bmatrix}\tag{11}$$

where

$$K_z>Max\left|\left(1-\frac{M_s}{M_{s0}}\right)k_{zp}\right|,\ M_z>Max\left|\left(1-\frac{M_s}{M_{s0}}\right)\left(\ddot{z}_d-k_{zd}\dot{z}_e\right)+\frac{1}{M_{s0}}d_z\right|,$$

$$K_\theta>Max\left|\left(1-\frac{I_p}{I_{p0}}\right)k_{\theta p}\right|,\ M_\theta>Max\left|\left(1-\frac{I_p}{I_{p0}}\right)\left(\ddot{\theta}_d-k_{\theta d}\dot{\theta}_e\right)+\frac{1}{I_{p0}}d_\theta\right|,$$

$$K_\varphi>Max\left|\left(1-\frac{I_r}{I_{r0}}\right)k_{\varphi p}\right|,\ M_\varphi>Max\left|\left(1-\frac{I_r}{I_{r0}}\right)\left(\ddot{\varphi}_d-k_{\varphi d}\dot{\varphi}_e\right)+\frac{1}{I_{r0}}d_\varphi\right|.$$

The chattering phenomenon may be alleviative by two methods: (i) low-pass filter and (ii) saturation function. A LPF module with the form of $\frac{1}{0.02s+1}$ is used to smooth the discontinuous control generated by (11); the saturation function introduced here to replace the sign function and or even avoided

$$sat(s)=\begin{cases}1, s>\Delta\\s/\Delta, -\Delta<s<\Delta\\-1, s<-\Delta\end{cases}$$

where Δ is the boundary layer which is a positive constant.

In addition, the transformation of inputs u_z, u_θ, u_φ and the actuators output forces is represented here:

$$\begin{pmatrix}u_z\\u_\theta\\u_\varphi\end{pmatrix}=\cos\theta\cos\varphi\begin{pmatrix}1 & 1 & 1 & 1\\L_f\cos\theta & L_f\cos\theta & -L_r\cos\theta & -L_r\cos\theta\\T\cos\varphi & -T\cos\varphi & T\cos\varphi & -T\cos\varphi\end{pmatrix}\begin{pmatrix}F_{fl}\\F_{fr}\\F_{rl}\\F_{rr}\end{pmatrix}$$

The required generated forces of electromagnetic actuators are

$$\begin{pmatrix} F_{fl} \\ F_{fr} \\ F_{rl} \\ F_{rr} \end{pmatrix} = \frac{1}{\cos\theta\cos\varphi} \begin{pmatrix} \frac{L_r}{2(L_f+L_r)} & \frac{1}{2(L_f+L_r)\cos\theta} & \frac{1}{4T\cos\varphi} \\ \frac{L_r}{2(L_f+L_r)} & \frac{1}{2(L_f+L_r)\cos\theta} & \frac{-1}{4T\cos\varphi} \\ \frac{L_f}{2(L_f+L_r)} & \frac{-1}{2(L_f+L_r)\cos\theta} & \frac{1}{4T\cos\varphi} \\ \frac{L_f}{2(L_f+L_r)} & \frac{-1}{2(L_f+L_r)\cos\theta} & \frac{-1}{4T\cos\varphi} \end{pmatrix} \begin{pmatrix} u_z \\ u_\theta \\ u_\varphi \end{pmatrix}$$

IV. SIMULATION RESULTS

The parameters of the active suspension system of the vehicle are shown in Table 1. The parameters of the SMC controller are listed in Table 2. Without generality, most of the gains are selected to be identical for simplicity.

Table 1: System parameters

Parameter	Value
M_s	1500 (kg)
M_u	59 (kg)
K_{sf}	20000 (N/m)
K_{sr}	21000 (N/m)
K_u	190000 (N/m)
I_p	460 (kg*m^2)
I_r	2160 (kg*m^2)
L_f	1.4 (m)
L_r	1.7 (m)
T	0.8 (m)

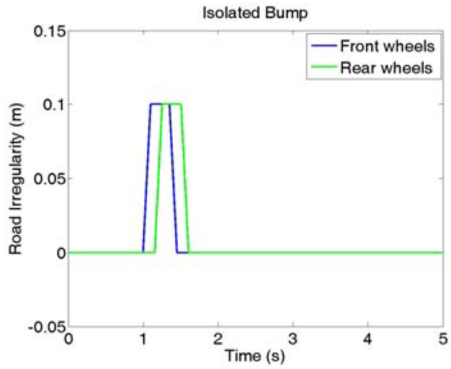
Table 2: Controller parameters

Parameter	Value
$k_{zp}, k_{\theta p}, k_{\varphi p}$	1
$k_{zd}, k_{\theta d}, k_{\varphi d}$	10
K_z, K_θ, K_φ	1
M_z, M_θ, M_φ	4.5,3.5,6
$\lambda_z, \lambda_\theta, \lambda_\varphi$	5
$\Delta_z, \Delta_\theta, \Delta_\varphi$	0.001

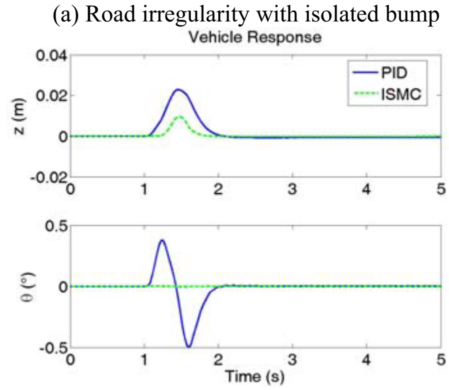
The road irregularities contain an isolated trapezoidal bump and sinusoidal two-track roads, as illustrated in Fig. 3(a) and 4(a). The time delay τ could be calculated under certain velocity, due to the track length between the front and rear axles which is $L = L_f + L_r = 3.1m$. Once the longitudinal velocity of the vehicle is assumed $v = 20\,m/s$, the time delay can be calculated as $\tau = L/v = 0.155\,s$, which is shown in Fig.3 (a). The response of car body indicates that the ISMC can absorb the road irregularity rapidly, compared with PID control as shown in Fig.3 (b).

The wavelength of the sinusoidal road is assumed to be 20m and the car runs over it under constant speed $v = 20\,m/s$ .Fig. 4 (b) shows the response of the car body under sinusoidal road which the phase-shift angle equals 90°. The results show that ISMC greatly reduces the oscillations due to the continuous sinusoidal road without the knowledge of the car body and road disturbance. It is convinced that the active suspension system with SMC controller improves the ride comfort. The corresponding generated forces are shown in Fig. 3(c) and 4(c). The required maximum forces are about 2000N, it is available for electrical actuator to generate the active force.

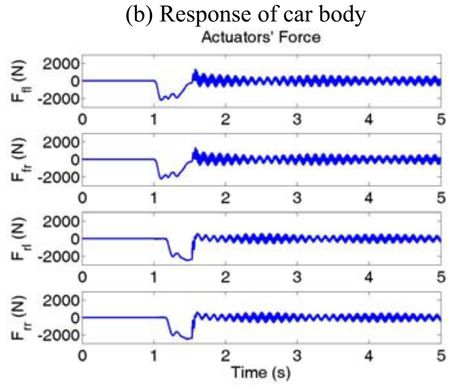
The actual parameters of the active suspension system, such as mass and inertia of sprung mass subsystem, will vary with different load. The robustness of the ISMC controller can guarantee the desired performances of the active suspension system with the variations of the system parameters reach up to 50%, i.e. $\Delta M_s = \pm 0.5 M_{s0}$, $\Delta I_p = \pm 0.5 I_{p0}$, $\Delta I_r = \pm 0.5 I_{r0}$, as shown in Fig. 5.



(a) Road irregularity with isolated bump

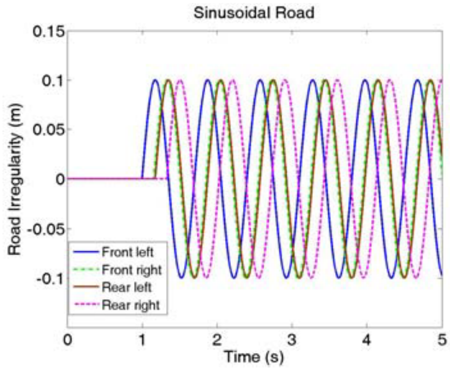


(b) Response of car body

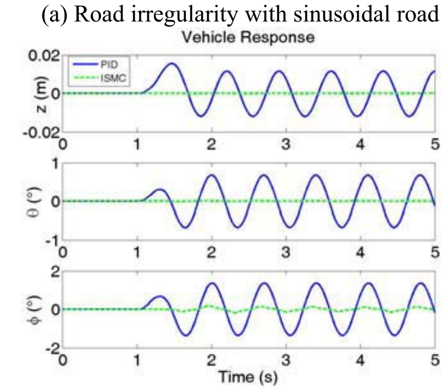


(c) Demanded actuators' force

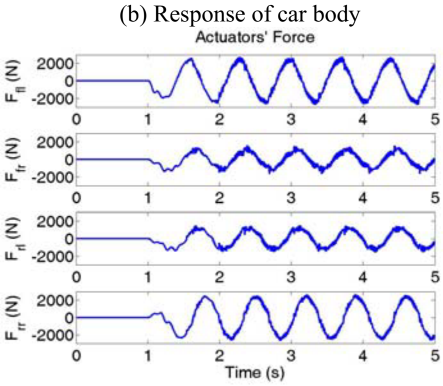
Fig.3: ISMC controller under isolated bump road



(a) Road irregularity with sinusoidal road



(b) Response of car body



(c) Demanded actuators' force

Fig.4: ISMC controller under sinusoidal road

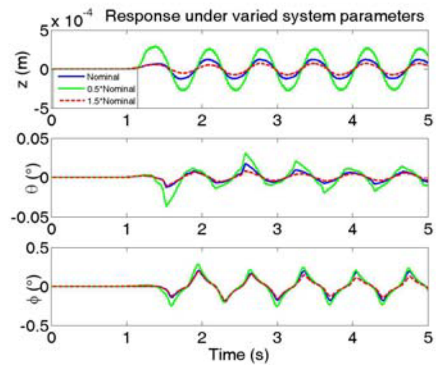


Fig.5: Response under sinusoidal road with varied system parameters

V. CONCLUSION

An accurate nonlinear active suspension model is developed in this paper, to track the accurate dynamic of the vertical motion of the vehicle. The decoupling transformation matrix is used to decouple the sprung subsystem and reduce the order of the whole system. Integral sliding mode control method is studied then to reject the road irregularities without the knowledge of the vehicle. As shown in Section IV, the dynamic responses of car body with ISMC scheme can be enhanced greatly comparing with conventional PID controller, hence the ride comfort and safety can be improved despite road irregularities.

The required force generated is constrained within 2000N by optimizing the SMC controller, as shown in the simulation results. It is below the output range of present linear electrical actuators; hence developing of linear electrical actuators and equipping into the active suspension system for implementation are available in future.

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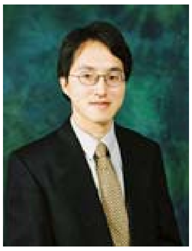

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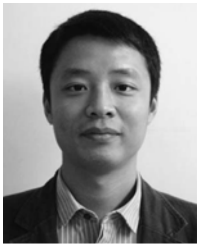

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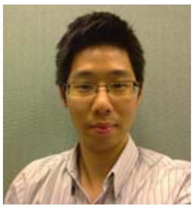

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