

Cooperative Optimization of Vehicle Ride Comfort and Handling Stability by Integrated Control Strategy

Wei Liu
Jilin University

Hao Ya
China FAW Group Corporation R&D Center

ABSTRACT

Vehicle needs suspension and steering systems with different features to fit different driving conditions. In normal straight driving condition, soft suspension and heavy steering systems are needed to achieve better ride comfort and straight line driving stability; in turning conditions, hard suspension and lightweight steering systems are needed to get better handling stability. The semi-active suspension system with Magneto-Rheological dampers can improve the ride comfort and handling performance of vehicle. Electrical power steering system is developed rapidly due to its portable and flexible operations as well as stable steering performance. In this paper, the features of electrical power steering and semi-active suspension systems have been done a more in-depth study, and an integrated control strategy was proposed to achieve the coordinated control of these two electric control systems arises in order to improve the ride comfort and ensure the steering stability in cornering situation. In a well on the basis of experimental data, the mathematical model of electrical power steering and semi-active suspension systems were adopted to predict the characteristics of these two electric systems. The numerical simulation results show that the integrated control strategy is effective and successful in improving the overall performances such as rapid electrical power steering and good body post regulation. The primary goal of this paper is to create an effective, reliable and safe integrated control strategy for steering and suspension systems that improves the ride comfort as well as handling stability.

CITATION: Liu, W. and Ya, H., "Cooperative Optimization of Vehicle Ride Comfort and Handling Stability by Integrated Control Strategy," *SAE Int. J. Commer. Veh.* 5(1):2012, doi:10.4271/2012-01-0247.

INTRODUCTION

Recently, Electric Power Steering (EPS) system has been developing rapidly and becomes one of the safe-critical systems in modern control systems[1]. To compare with the hybrid-power assisted steering system, EPS system promises weight reduction, fuel savings and package flexibility, at no cost penalty. EPS system for a vehicle is known in which the torque output from an electric motor for generating assisted steering effort is reduced by a gearbox in order to transmit proper torque and velocity to the output shaft of the steering system[2]. In addition, EPS system can provide good road feeling for the drivers, the road feeling is effectively defined the steering wheel torque the driver senses during steering maneuvers and by the vehicle response to the steering inputs. Semi-active suspension (SAS) has received more and more attention because of its special properties. One of the properties is SAS based on Magneto-Rheological (MR) Fluid can make fast regulation of the damping coefficient with

reaction time in even few milliseconds. The second significant feature is that little power is required to significantly change the suspension damping force, whose maximal value does not extend 20 W per damper[3,4].

Usually, EPS and SAS perform independently to improve maneuverability in steering and the ride comfort respectively. However, as the electronic control subsystems of the chassis in vehicle, they often work together, the performances of vehicle depend not only every subsystem but also the relationship of these subsystems strongly[5,6,7]. In this paper, a full multi-body dynamic vehicle model which describes the dynamics of the integrated system of EPS and SAS is established and the dynamic equations are derived. Considering the complexity of full-car mathematical modeling, the full-car model of the light bus, with detailed suspension and steering system, was carefully built in multi-body dynamics software SIMPACK. An integrated control strategy base on neural network is presented to control EPS

and SAS systems. The vehicle performances of the integrated system are compared with that of EPS and SAS only. The significant results are drawn out and future work is suggested.

MATHEMATICAL MODELING OF EPS

EPS is the mechanical steering mechanism that added electric power plant, and the basic components of EPS include power motor, gear box, electronic control unit (ECU), and the speed and torque sensors[8]. When driver manipulating the steering wheel, the torque sensor and speed sensor offers steering wheel torque and vehicle speed signals to the ECU simultaneously, and ECU adjusts the output power torque of EPS by controlling the input current of EPS. The photo of EPS used in this paper is shown in [figure 1](#). The schematic diagram of steering mechanism with EPS is shown in [figure 2](#).



Figure 1. Electric Power Steering system.

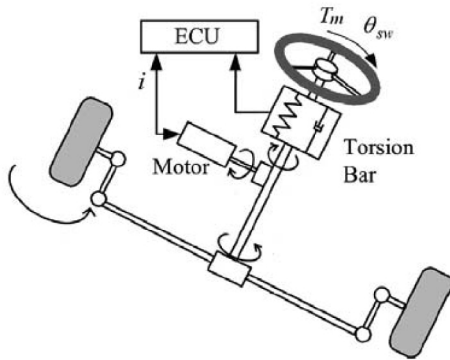


Figure 2. Steering mechanism with EPS.

The power torque characteristics of EPS can be described as:

$$I_{sw} \ddot{\theta}_{sw} + B_{sw} \dot{\theta}_{sw} + K_{sw} \theta_{sw} = T_s + K_{sw} \frac{x_r}{r_p} \quad (1)$$

$$I_m \ddot{\theta}_m + B_m \dot{\theta}_m = T_m - K_m (\theta_m - G \frac{x_r}{r_p}) \quad (2)$$

$$m_r \ddot{x}_r + B_r \dot{x}_r = \frac{GK_m}{r_p} (\theta_m - G \frac{x_r}{r_p}) + \frac{K_{sw}}{r_p} (\theta_{sw} - \frac{x_r}{r_p}) - \frac{2K_{kp}}{G_t} (\frac{x_r}{G_t} - \delta) \quad (3)$$

$$I_w \ddot{\delta} + B_{kp} \dot{\delta} = K_{kp} (\frac{x_r}{G_t} - \delta) - T_w \quad (4)$$

$$T_m = \frac{i_1 K_a}{R} (U - K_b i_1 \dot{\delta}_1) \quad (5)$$

$$U = L \dot{I} + RI + K_b n_m \quad (6)$$

$$T_m = K_a I \quad (7)$$

where, I_{sw} , I_m and I_w are the moment of inertia of steering wheel, motor and tires; B_{sw} , B_m , B_r and B_{kp} are the damping coefficient of steering wheel, motor and tires; θ_{sw} and θ_m are the steering wheel angle and angle of motor; K_{sw} , K_m and K_{kp} are spring constant of steering wheel, motor and tires; T_m is the output power torque of EPS; i_1 is the reduction ration of reducing mechanism; δ_1 is the turn angle of small gear; K_a is the torque coefficient of power motor; K_b is the back electromotive force coefficient of power motor; R is the electric resistance of power motor armature; L is the induction of power motor; n_m is the rotational speed of power motor; U and I are the input voltage and current of power motor. The characteristic curve of EPS is shown in [figure 3](#).

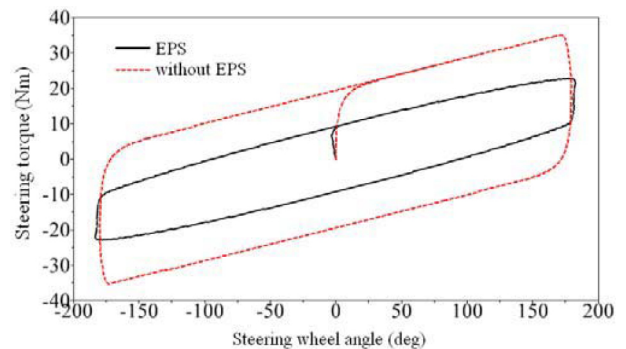


Figure 3. Characteristic curve of EPS.

MATHEMATICAL MODELING OF SAS

A prototype MR damper with the mixed mode was designed and manufactured in order to perform SAS system in the vehicle, shown in [figure 4](#). The nonlinear force-velocity and force-displacement responses of MR damper has been described by many mathematical models, such as Bingham model and nonlinear hysteretic bi-viscous model[9,10,11], etc. One of the most popular is the Bouc-Wen hysteresis models, originally deduced by Bouc in 1967 and later modified by Wen in 1976[12,13]. Later on, Spencer proposed a phenomenological model for MR dampers, a more accurate one based on the Bouc-Wen model[14], shown in [figure 5](#). This model predicts the force-displacement behavior of the damper well, and it expresses force-velocity behavior that more closely resembles the experimental data. The damping force of the MR damper can be derived as follow:

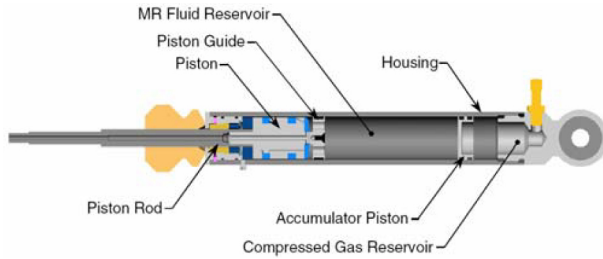


Figure 4. Schematic diagram of MR damper.

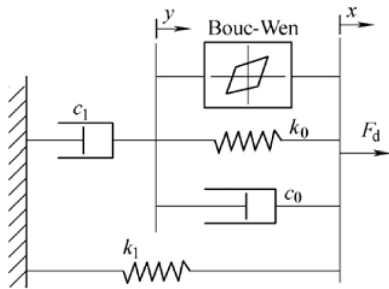


Figure 5. Phenomenological model of MR damper.

According to the nonlinear force-velocity and force-displacement responses, the mathematical model of MR damper is expressed as [Eq. \(13\)](#).

$$F_d = c_1 \dot{y} + k_1(x - x_0) \quad (8)$$

where the damping force is governed by

$$\dot{z} = -\gamma |\dot{x} - \dot{y}| z |z|^{n-1} - \beta(\dot{x} - \dot{y}) |z|^n + A(\dot{x} - \dot{y}) \quad (9)$$

Solving (8) for results in

$$\dot{y} = \frac{1}{c_0 + c_1} [\alpha z + c_0 \dot{x} + k_0(x - y)] \quad (10)$$

In order to verify the parameters of phenomenological model, a MR damper test was applied only to the experiment with $\pm 25\text{mm}$ of displacement amplitude. In this setup, a DC motor and a gearbox were employed to drive the MR damper. The current signal applied in the MR damper is controlled by a DC regulated power supply. And an electronic dynamometer was included in series with the damper to measure the damping force. The block length of MR damper is 147mm, and the Maximum input current is 2.0A.

The results of parameter identification process using the optimization algorithm are shown in the [Table 1](#).

Table 1. Parameters of Phenomenological model.

A	31
β	2.6 mm^{-2}
γ	2.9 mm^{-2}
n	2
c_0	$29 \text{ N}\cdot\text{s}\cdot\text{mm}^{-1}$
c_1	$1400 \text{ N}\cdot\text{s}\cdot\text{mm}^{-1}$
α	430 mm^{-2}
k_0	-300 Nm
k_1	-298 Nm

[Figure 6](#) shows the theoretical and experimental results. It can also be seen that the theoretical predictions are in close agreement with the experimental results. This means that [Eq. \(8\), \(9\), \(10\)](#) can be used to describe the characteristics of the prototype MR damper and decide the real-time input current.

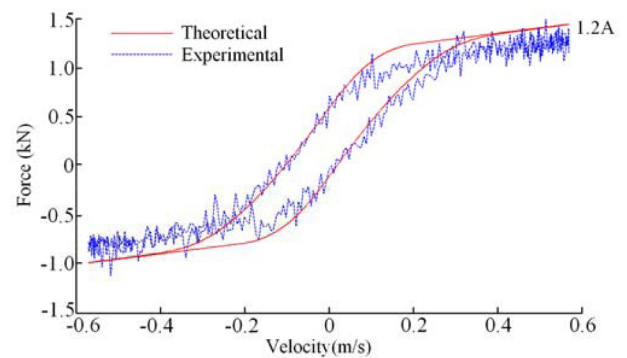


Figure 6. Damping characteristics of SAS.