Simulation of Active Force Control Using MR Damper in Semi Active Seat Suspension System

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Abstract. With the rapid development of electronic sensor and actuator technology, semi-active seat suspension system has become even more practical driven by lower power consumption. Magneto-rheological (MR) dampers are among the best and the most reliable semi active control devices that can produce controllable damping force in seat suspension system to further improve the ride comfort. This paper focus on a new controller scheme named Active Force Control (AFC) to control the damping force of the MR damper to achieve better ride comfort. The phenomenological Bouc-Wen model for MR damper has been simulated in Matlab Simulink to study the effectiveness of the new AFC controller. A sinusoidal signal simulated as vibration source is applied to the seat suspension system to investigate the improvement of ride comfort as well as to ascertain the new AFC controller robustness. Comparison of body acceleration signals from the passive suspension with AFC controller semi active seat suspension system shows improvement to the occupant ride comfort under different vibration intensities.

Keywords: Active Force Control; Magneto Rheological; Semi Active seat suspension

1. Introduction

The vehicle suspension system is responsible in providing the ride comfort to the passenger by optimally isolating the passenger from any vibration sources. The spring in a suspension system stores the energy from the vibration excitations as potential energy which in time is released back as kinetic energy as the spring bounce back and forth. Another element in the suspension system is the damper which can be installed in parallel with the spring to absorb the kinetic energy produced by the bouncing excitations of the spring [1].

Modern suspension system consists of three different categories which are passive, active, and semi-active suspension. Active and semi-active suspension systems are created to provide a better ride comfort for a wider range of ride conditions, with better adaptability to road profiles which passive suspension cannot do. With wide varieties of controller developed recently, it is also possible for the active and semi-active suspension systems to improve both handling and ride comfort without compromising either one [2].

A number of researchers have turn their attention to further improve ride comfort in this multiple degree of freedom system by introducing another level of vibrations control on top of standard suspension system which then contribute to the development of seat suspension system such as in [3] and [4]. With similar approach to the research done on suspension system, seat suspension systems are
divided to passive, active and semi-active system with some researchers have already come up with robust controller technique.

The quarter car seat suspension model used in this study is shown in Figure 1 where the system is assumed to be a single degree of freedom suspended spring mass damper system. Total mass of the passenger and seat is given by \( m_b \), \( k_s \) representing the stiffness of the seat, while \( c_s \) is the damping rate of the seat but in semi active seat system it is replaced by the \( f-v \) characteristic of MR damper.

The ride comfort which is the main objective of this research is quantified by the peak to peak vertical body acceleration experienced by the occupant of the seat. The degree of comfort is analysed by referring to the ISO standard as in Table 1 [5].

![Figure 1. Quarter car seat suspension model.](image)

### Table 1. Comfort evaluation [ISO 2631-4].

| Vertical Body Accelerations (ms\(^{-2}\)) | Comfort Evaluation     |
|-----------------------------------------|------------------------|
| < 0.315                                 | Not uncomfortable       |
| 0.315 – 0.63                            | A little uncomfortable  |
| 0.5-1                                   | Fairly uncomfortable    |
| 0.8-1.6                                 | Uncomfortable           |

Active and semi-active suspensions have control systems which force the system to achieve optimized conditions. Semi-active suspension uses semi-active dampers whose force is commanded indirectly through a controlled change in the dampers’ properties. This change is affected by a damper controller that receives information from the system controller. A semi-active suspension combines the advantages of both active and passive suspensions. It can be nearly as efficient as a fully active suspension in improving ride comfort and stability and is much more economical [6], and [7]. It is also safer to use since if the control system fails, the semi-active suspension can still work as a passive suspension system.

### 2. MR Damper

MR damper has been shown to be effective as semi-active damper that can satisfy the requirements of ride comfort and vehicle stability [8]. Moreover, an MR damper is an effectively fail-safe device from an electronic perspective. If any fault happened in the system, the MR damper still works as a passive damper within definite performance characteristics, depending on the off-state case of the MR damper [9].

Corresponding mathematical model for MR damper is still intensely researched as it is relatively complex because of its rheological and thermodynamics effects [10]. Complex model of MR damper had led the researcher to turn their heads onto numerous parametric phenomenological models, like the
Bingham plastic model [11,12], Bouc–Wen hysteretic model [13–16], Duffing’s equation [17] sigmoid-based model [18], hyperbolic tangent function [19] or even intelligent method approach like neural networks in [20].

3. MR Damper Modelling

A parametric approach had been used in this research to model the MR damper using existing model from previous researchers. A comprehensive review of parametric models can be found in [21]. This paper will utilize the phenomenological Bouc-Wen model modified by spencer as the mathematical modelling for MR damper.

This model was proposed firstly in [16], as shown in Figure 2. This modifies the Bouc-Wen representation through the introduction of an extra internal degree of freedom which differ from the original Bouc-Wen model. This model was proven to better in adaptively tracks the force-velocity hysteresis loop from the experimental measurements.

The force equation from Figure 2 is given by:

$$ f = c_1 \dot{y} + k_1 (x - x_0) $$

(1)

where the evolutionary equation to produce the hysteretic characteristics of MR damper is given by:

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

(2)

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

(3)

Conventional Bouc-Wen model basically introduce only equation (2) while the modified Bouc-Wen model introduce equation (3) as another additional evolutionary equation which surprisingly improve the tracking ability of the overall model. The three equations are simulated to compose the MR damper model in Simulink. This bouc-wen model will be then replaced the damping parameter of the seat, c_s, in Figure 1.

The optimized model of modified Bouc-Wen simulated in Simulink performance is shown in Figure 3. The F-v curve shows the hysteresis characteristics in MR damper for difference input current values. The peak to peak force values are matching with the datasheet provided by the manufacturer and the model is considered to track the real MR damper response.
The F-v curve shown in Figure 3 represents the significance of input current changes make to the force output in the MR damper at certain velocity value. The maximum and minimum values differences are about 1kN for the changes of only 1A in input current. This shows that small changes in input current has affected the changes of damping rate of the damper significantly which is much desired in controlling the damping force of the semi active seat suspension system.

4. Active Force Control

Active Force Control, AFC is known as a robust disturbance compensator and is superior compared to other conventional methods in dealing with various types of disturbances [22]. It is a very simple scheme that is suitable to be applied to a plant that we wish to control in the presence of disturbance. Applying a simple Newton Law, AFC will react to estimate the disturbance occur at some instant, and give an appropriate signal to the actuator to counter the corresponding disturbance force. There are two main parameters in AFC which needs some intelligent method to estimate. One is the estimated mass, and second is the inverse dynamic of the actuator.

Figure 4 shows the AFC loop in the controller scheme where it takes place to estimate the disturbance forces acting to the sprung mass of the vehicle. Base on the Newton Second Law we have the sum of all resultant forces including disturbance force acting to the sprung mass as:

\[ \sum F = m \ddot{z}_s \]  

The sprung mass acceleration, now is actually produced by both actuated force and also disturbance forces from various sources. The sprung mass acceleration can be taken from the accelerometer which will be attached to the sprung mass. The controller now needs the value of estimated mass to multiply with the acceleration value from the accelerometer to produce a calculated value of the resultant force in computer.

To get the estimated disturbance force, the controller now need to subtract the actuated force which can be measured easily using a force (or pressure) sensor, from the resultant force calculated by the controller.

\[ D_f = F' - m_s' \ddot{z}_s \]
Where

\[
\begin{align*}
D_f' & : \text{Estimated disturbance force} \\
F' & : \text{Actuator force taken by using force (or pressure) sensor} \\
\ddot{z}_s & : \text{Body acceleration taken using accelerometer.}
\end{align*}
\]

The calculated disturbance force, then will be converted into an appropriate electrical signal through the inverse dynamic of the actuator. The real pneumatic actuator then will convert this signal into the force and inject it into the suspension system to overcome the disturbance force.

There were numbers of research that had successfully applied AFC into dynamic mechanical system. Similarly, [23] implemented AFC into suspension system but combined with skyhook controller. In his work, the AFC-based controller scheme has successfully improved ride comfort. [24] implemented AFC into robotic arm to control its motion. In their work the proposed controller has improved the motion of the mobile robot under the effect of disturbances. Kwek et.al., 2003 implemented AFC into 5-link biped robot [22]. Their work was done via simulation study.

![Active force control loop](image)

**Figure 4.** Active force control loop.

In other application such as in robotics, AFC has been researched widely such as in [25-27] to mention a few but there is very limited research done on AFC in translational application such as suspension system. While in seat suspension system, AFC approach has only been done by [28] in active seat suspension. The project focused on a helicopter seat suspension where AFC is coupled with Iterative Learning Algorithm control as the function of inverse actuator feed.

### 5. Results and Discussion

This section discusses the output of the newly design controller scheme, which will determine the improvement in ride comfort level achieved by this research. The main indicators will be the peak to peak body acceleration experienced by the seat occupant and the acceleration transmissibility of the passive seat suspension system. The indicators are compared to controller scheme over different frequencies of vibration.

In this paper, the inverse actuator model in the AFC loop is treated as a linear function which is the inverse of the linearized function for the modified Bouc-Wen model. The actual model of the inverse function need another comprehensive literature to cover or by using another parametric approach to get a better result. The estimated mass of the body for this simulation has been determined by using heuristic approach as it is only involving a simple single degree of freedom mass spring damper configuration.
Figures 5 show the body acceleration experienced by the seat occupant for both passive seat suspension and controller scheme seat suspension when subjected to 4mm of excitation at 1Hz and a sine chirp signal with varying frequencies of 0.1 to 1 Hz. It is clear that comparison in peak to peak value of body acceleration shows that the controller scheme managed to give useful input to MR damper hence providing the suspension system with better damping force compared to the uncontrolled damping force of a passive system.

Based on comfort evaluation by ISO shown in Table 1, the degree of comfort has improved from fairly uncomfortable (nearly 1m/s²) to a little uncomfortable (0.3m/s²). Though the proposed semi active seat system shows better reduction in seat vertical acceleration in low frequency, but the opposite phenomenon happens at higher frequency where we can see the passive response has lower vertical seat acceleration. But since the lower frequency vibration that gives harm to human body, it is essential to better suppress vibration at low frequencies than at higher frequencies.

6. Conclusion
Active Force Control loop has been successfully adopted into a semi active seat suspension system in this paper. Using a modified Bouc-Wen hysteresis model, the MR damper has been shown to be able to act as actuator to the controller. Damping force has been successfully controlled in the simulation. Comfort evaluation of the semi active seat suspension system has been improved from fairly uncomfortable (nearly 1m/s²) to a little uncomfortable (0.3m/s²) when compared to passive system. Further improvement can be achieved if the inverse model of the MR damper can be formulated more accurately and further study on intelligent method to find estimated mass can be done in the future.
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