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ORIGINAL ARTICLE

Semi Active Seat Suspension System using Modified Intelligent Active Force Control

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ABSTRACT – This paper presents a modified intelligent active force control (AFC) control strategy in a semi active seat suspension system. The main actuator studied in this research is the Magneto-rheological (MR) damper. Since a semi-active device like MR damper can only dissipate energy so a modified version of AFC controller is needed. The modified AFC controller main function is to determine the appropriate control force. A Heaviside Step Function (HSF) is used to ensure the MR damper produce the desired damping force according to the control force generated by AFC controller. The phenomenological Bouc-Wen is used to study the effectiveness of the new AFC controller taking into account the dynamic response of the damper. Sinusoidal signals simulated as vibration sources are 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 up to to 45% improvement to the occupant ride comfort under different vibration intensities.

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INTRODUCTION

Ride comfort has become one of the criteria in designing the seat for heavy machinery in the workspace which exposed to the prolonged low-frequency vibration. Several researches previously has alarm the society about the threat coming from the exposure to low-frequency vibration to the human body [1, 2]. Conventionally, passive seat suspension with fixed damping and spring rate was designed to compensate for the vibration, but having a fixed dynamic properties means the passive suspension has to compromise between ride comfort and handling [3].

Development in sensors and electronics industry has allowed seat suspension designer to adopt the active suspension system in the seat suspension system [4-6]. Active actuators deployed in the active seat suspension system usually limited by the power input, and also vulnerable to fatal failure in the case of power loss or controller malfunction occasion. Semi-active system offers sufficient amount of control force to the seat suspension system and proven to be fail-safe in the event of power loss, as it can perform as passive seat suspension when the power is cut off. Semi-active system can be applied by either controlling the damping force or the spring rate of the suspension system. Many researchers utilise the magnetorheological, MR damper as a vibration isolation system in various engineering applications as it offers a large amount of control force with a very minimum power input [7-10].

The MR damper dynamic properties can be manipulated by changing the current supplied to the damper. The current intensity will affect the magnetic field in the magnetorheological fluid inside the damper thus varying its damping rate to the desired value. This highly non-linear actuator however needs a proper controller scheme to operate effectively. Numbers of researchers have successfully studied the proper controller scheme for the MR damper in recent years [11-13].

This article proposes an intelligent active force control, AFC scheme to provide the input signals for the MR damper. The controller scheme is applied to a semi-active seat suspension system in a simulation work under the Matlab Simulink environment. The performance of the proposed controller is measured base on its seat transmissibility value, compared to the passive seat suspension system.

SEAT SUSPENSION MODEL

In this paper, the study on the single-seat suspension system is done using a single-seat suspension model. The 2 degree-of-freedom model consists of a spring, a mass, and a damper with vibration sources generated using sine-wave signals at the different amplitude and different frequency. The model is shown in Figure 1 for a passive seat suspension system with fixed damping rate, c_s, and spring constant, k_s, while semi-active suspension system has a different value of damping rate, c_s. Previous researcher has shown that a passive suspension system fails to attenuate the amplitude of given vibration sources when the frequency of the vibration is at low frequency [9]. An active suspension system is designed by replacing the spring and damper between the seat and the floor with an actuator model which means the actuator work without any support from any spring element. In all three types of suspension system, the goal is to isolate the vibration sources at the vehicle floor from transmitted to the seat pan, thus attenuate the vibration experienced by the seat occupant.

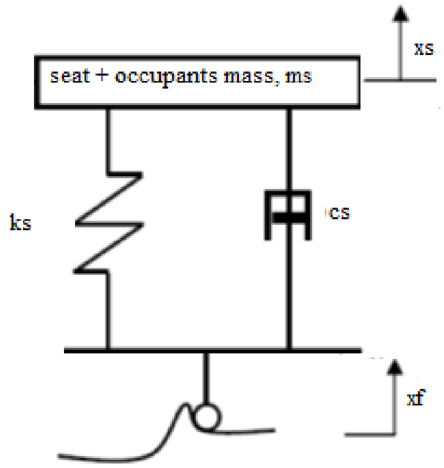

Figure 1. Conventional passive seat suspension model.

The equations of motion for the single-seat model are given by Eq. (1). Parameters m_s, representing the seat mass + occupant body. While k_s and c_s, are the stiffness of the spring and the damping rate of the seat suspension respectively. The vertical motion of the seat and the vehicle floor is represented by x_s and x_f respectively.

$$-k_s(X_s - X_f) - c_s(\dot{X}_s - \dot{X}_f) = m_s\ddot{X}_s \tag{1}$$

In a semi-active seat suspension system, the damping force generated by the damper is controllable which means the damping rate, c_s is not a constant. The damping force generated by the conventional damper depends on the stroke velocity, while in the MR damper, the damping force is also affected by the magnetic field form from its current input.

MR DAMPER MODEL

The MR damper is a highly nonlinear device so the mathematical model is very complex [14-16]. Researchers have turned their attention to the parametric model with hysteresis behaviour to model the MR damper [17-20]. The modified Bouc-Wen model has been used to represent the MR damper in this study. The model is shown in Figure 2, and the governing equations are given by Eq. (2) to (4). The hysteresis parameters of the modified Bouc-Wen model have been tuned to follow the characteristic of an MR damper model produced by Lord Company.

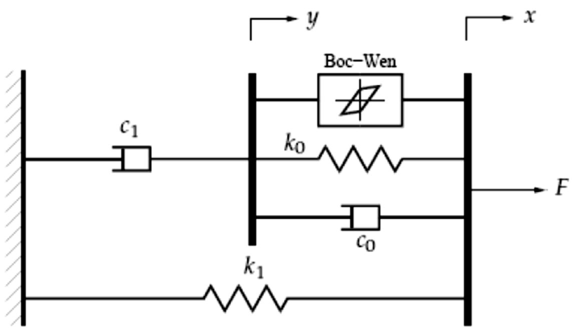

Figure 2. The modified Bouc-Wen model.

$$F = c_1\dot{y} + k_1(x - x_0) \tag{2}$$

$$\dot{z} = -\gamma|\dot{x} - \dot{y}|z\,|z|^{n-1} - \beta(\dot{x} - \dot{y})\,|z|^n + A(\dot{x} - \dot{y}) \tag{3}$$

$$\dot{y} = \frac{1}{c_0 + c_1}[\alpha z + c_0\dot{x} + k_0(x - y)] \tag{4}$$

In the modified Bouc-Wen model, there are nine hysteresis parameters need to be tuned including three current dependent parameters in order to produce hysteresis behaviour with minimum error compared to real hysteresis behaviour of a physical MR damper. Optimisation method using CS algorithm had been used to determine the modified Bouc-Wen hysteresis parameters value. The final values of each parameter are given in Table 1 while the current dependent parameters were derived from Eq. (5) to (7).

$$\alpha = 737.8i + 112.4 \tag{5}$$

$$c_0 = 13.73i + 8.15 \tag{6}$$

$$c_1 = 398i + 24 \tag{7}$$

These three equations are constructed by curve fitting the parameters' values when the model optimised at different current input values.

Table 1. The modified Bouc-Wen parameters values.

Parameters	Values
β (cm^{-2})	20
γ (cm^{-2})	29.12
A	30
k_0 (N/cm)	5
x_0 (cm)	-2
k_1(N/cm)	5

INTELLIGENT ACTIVE FORCE CONTROL

The controller scheme used in this research is based on the intelligent active force control (AFC) scheme. The idea of AFC controller scheme was introduced by Hewit [21], with most of its applications is in rotational mechanical movement such as in [22, 23]. Many other researchers have also successfully applied the AFC controller in mechanical translational movement studies [5, 24]. The basic concept of the AFC controller is the second Newton's Law where the total force has the same direction with the acceleration and proportional to the mass. The total force in this research refers to the force from the suspension component total up with the force generated by the disturbances subjected to the mass. The AFC controller calculates the disturbance force by using the acceleration and force inputs as shown in Eq. (8). The original AFC controller scheme can be divided into two parts. The first part is to calculate the disturbance force acting on the system, while the second part is where an inverse actuator function is modelled to convert the calculated disturbance force into appropriate signals so the actuator could produce the required force to level out the disturbance force.

$$f_d = m'\ddot{x} - f_{ai} \tag{8}$$

For this research, the MR damper is not a fully active actuator as the actual force, f_{ai}, generated by the damper not only controlled by the current, i, but also depends on the velocity value $\dot{x}$. The second part of the original AFC scheme which to generate the current signals to the modified Bouc-Wen model has been replaced with Heaviside Step Function (HSF) as shown in Figure 3.

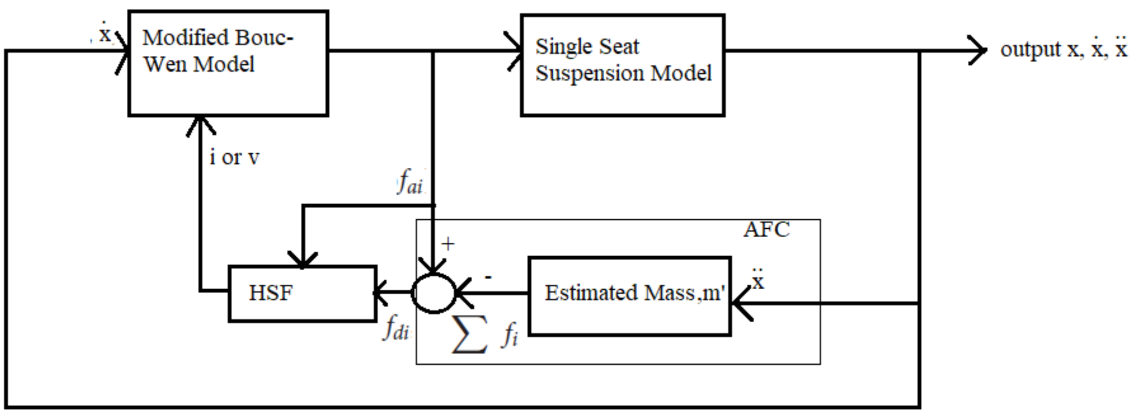

Figure 3. Semi-active seat suspension controller scheme using AFC.

The first part of the AFC controller scheme was used to estimate the disturbance force acting on the seat suspension system at any instant as shown in Eq. (8). But the estimated mass parameter needs to be tuned in order to optimise the AFC controller. The desired force, f_{di}, to cancel out the disturbance force is equal to the disturbance force but in opposite direction. HSF function described by Dyke [25] work based on the following rules.

i. The output current, i is either 0 or i_{max}.

ii. If the actual force generated by the MR damper, f_{ai} is equal to the desired force, f_{di} calculated by AFC controller, the current, i, value remains constant ($f_{ai} = f_{di}$).

iii. If the actual force generated by the MR damper, f_{ai} is smaller than the desired force, f_{di} calculated by AFC controller and they are in the same direction, then the current value is set to i_{max} until the actual force equal to desired force ($f_{ai} < f_{di}$).

iv. Other than that, the current value is set to zero.

Introducing the HSF function into the AFC controller eliminates the mathematical burden of composing the inverse actuator function as the inverse function for a nonlinear two input, one output actuator like MR damper is actually very complex to model. This means the only one parameter left to be tuned in the AFC scheme is the estimated mass. Eq. (9) describes the HSF function.

$$i_i = i_{max}(H((f_{di} - f_{ai})f_{ai}) \quad i = 1,2 \tag{9}$$

SIMULATION SETUP

Sets of the sine wave generated by Eq. (10), with different amplitude and various frequencies representing the vibration of the road profiles, has been subjected to the seat suspension system to study the effectiveness and the robustness of the proposed semi-active seat suspension system. The sinusoidal signals with frequencies (1-4.5 Hz) are chosen to see how the seat suspension performs at low frequencies around the natural frequency range. Table 2 shows the sine wave amplitude and frequency values.

$$x_f = A_m \sin(2\pi f) \tag{10}$$

From the response, the seat effective acceleration transmissibility, SEAT value then calculated using the root mean square of the body acceleration by using Eq. (11) and Eq. (12).

$$RMS(a) = \sqrt{\frac{1}{N}\sum_{i=1}^{N} a_i^2} \tag{11}$$

$$T = \frac{a_s}{a_v} \tag{12}$$

For the response obtained from the seat suspension system, the RMS value represented by a_s, while a_v denotes the RMS value for floor acceleration values. Transmissibility calculated by comparing the ratio of RMS seat suspension acceleration to the floor acceleration. Any value lower than one means that the seat suspension is isolating the vibration from the seat occupant while if the value exceeded one tells us that the seat is amplifying the vibration instead. The SEAT value in some research had been represented in percentage by multiplying the transmissibility value by 100 [26].

Table 2. Sinusoidal signals.

Amplitude, A_m (mm)	Frequency, f (Hz)
6	1.00, 1.25, 1.5, 1.75
5	2.00, 2.25, 2.50
4	2.75
3	3.00, 3.25, 3.50
2.5	3.75
2	4.00, 4.25, 4.50

RESULTS AND DISCUSSION

The results for this research mainly focus on minimising the vertical seat acceleration to reduce the vibration effect experienced by the seat occupant. The acceleration transmissibility over a various range of frequency is plotted as shown in Figure 4 to see the proposed controller performance at different frequencies. From the figure, the acceleration transmissibility value for passive seat suspension is higher than 1 at lower frequency range (near the resonance frequency) which shows that the passive seat at this working frequency is amplifying the vibration from the floor instead of suppressing it.

The proposed AFC controller manages to improve the transmissibility at lower frequencies but the passive seat suspension performs better at higher frequency range. The proposed controller acceleration transmissibility value at a frequency higher than 3 Hz is still below 1 showing that the semi-active seat suspension is still suppressing the disturbance generated by the sine wave.

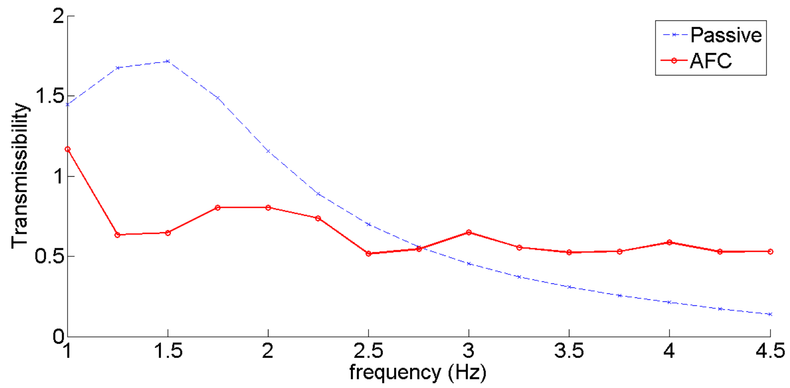

Figure 4. Acceleration transmissibility comparison over different vibration frequency.

From this transmissibility graph, the passive seat response shows a poor performance between 1-2 Hz of vibration. This is because vibration at these frequency values is near to the natural frequency of the seat suspension system which is around 1.5 Hz. The seat acceleration value for both passive and the AFC controller at vibration excitation of 6 mm 1.5 Hz are shown in Figure 5. Overall peak-to-peak acceleration value of the proposed AFC controller is lower than the acceleration value obtained from the passive seat suspension system. The results obtained from both simulations are out phase due to the delay introduced in the AFC loop.

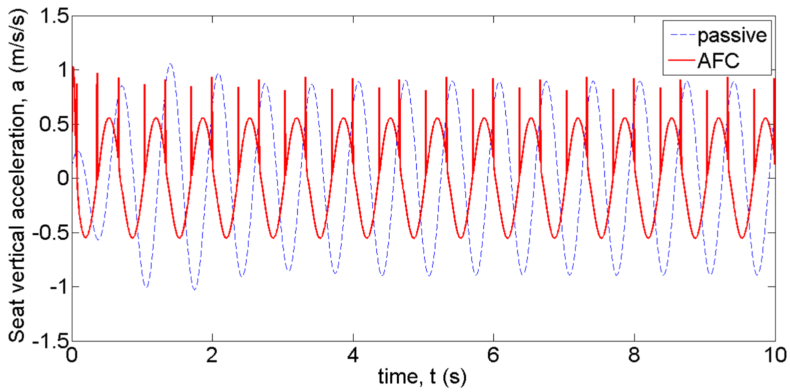

Figure 5. Seat acceleration for excitation at 6 mm 1.5Hz.

The current profile generated by the HSF function is shown in Figure 6 and the corresponding damping force generated by the MR damper is shown in Figure 7. As mention in section 4, the HSF function set the current signals to 0 at most of the condition and only set to a maximum value when the actual force generated by the MR damper is lower than the desired force calculated by the AFC controller and they both in same directions. This small current usage means the power consumption required by the MR damper to operate is low. The damping force generated is also relatively low due to the fact that the damper only compensates the disturbance force with the help of a spring which is missing in the active control strategy.

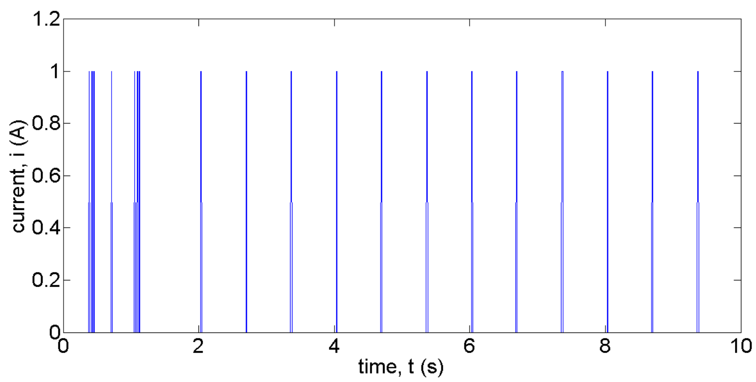

Figure 6. Current profile.

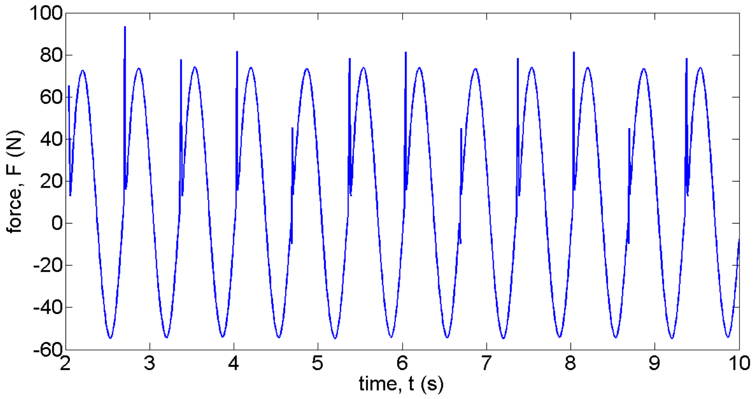

Figure 7. Damping force generated by MR damper.

CONCLUSION

This article focuses on developing the control strategy for an MR damper in a semi-active seat suspension system. The controller scheme suggested in this article is an active controller which need some modification in order to operate with a semi-active actuator like MR damper. The HSF function introduced in this article has successfully fed the signal calculated by the controller into the actuator model. The AFC based controller scheme has been successfully applied to the seat suspension system and the simulation results show that the proposed control strategy improves the seat transmissibility in low-frequency vibration application hence improving the ride comfort. A passive system performance to attenuate the vibration effect usually reduced when the system exposed to vibration signals near to its natural frequency value. This explained why the passive counterpart amplified the amplitude instead of attenuating it when the passive system is exposed to a vibration source at between 1-2 Hz frequency. The proposed controller has successfully compensated for the weakness of the passive suspension system at the mentioned frequency thus improving the transmissibility of the seat suspension system.

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