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ORIGINAL ARTICLE

OFF-ROAD VEHICLE SEAT VIBRATION CONTROL WITH SELF- TUNING PID CONTROLLER

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ABSTRACT

In general, in off road vehicles a seat suspension system is not used to control the vibrations which are induced from the road surface irregularities. The vibrations from road irregularities bypass the suspension system are transferred to seat and then to human body. Such vibrations, if not controlled causes the back pain and other spine related problems over the course of time. Use of a suspension seat is a common way to isolate the vehicle operator from the adverse effects of vibration exposure. In this research, semi active PID control of seat suspension with MR dampers is discussed. The proposed controlled quarter vehicle model is compared withuncontrolled system using simulation work under different type of roadexcitation. Results obtained by simulation work shows theeffectiveness of PID controlled suspension system in improving theride comfort and safety of driver compared touncontrolled suspension system.

Key words: *PID, Matlab, Semi-Active Seat, MR damper.*

INTRODUCTION

SUSPENSION system has to fulfil conflicting multipletasks related to passenger ride comfort, controlling thesuspension deflection within designed limits as well as toenhance and provide road holding capability to travellingvehicles. Depending on the effectiveness and performancedelivering behaviour of the vehicle system, a suspension system can be categorized into three types generally known as passive or uncontrolled, semi-active and active. Normally, passive suspension system is assembled with uncontrollable shock absorber having no feedback control and represents cost effective technology. However, optimum ride comfort andrelated requirements cannot be achieved in a desired way dueto uncontrollable damping force generation nature of shockabsorber thus limiting its performance within certainfrequency range as well as under changing vehicle travellingconditions. While active suspension system technology canprovide best results in a wider frequency range when theproblems related to ride comfort, safety of vehicle andworking of suspension system are considered due to assemblyof mechatronics based devices such as sensors and actuators.

Off – road vehicle causes severe vibrations which are usually of low frequency ranging below 5 Hz as are driven and work on gravel or rough surfaces. Vibrations at lowexcitation frequencies (0.5–5Hz) are the main risk factors for lumbago or backache, which seriously affects mental and physical health and reduces the work efficiency of drivers and passengers [1–5].

Hence, such vibrations require isolation.One way to lessen the adverse effects of vehicle vibration to the operator is through the use of a seat suspension system. Suspension seats are included in most off-road vehicles are passive in nature. Moreover, during operation, these vehicles are subjected to significant energy in the 2 to 4 Hz region where conventional seats tend to amplify vertical vibration. A common method used to reduce vehicle operators' exposure to vehicle vibration is to design the natural frequency of the seat suspension so that it is considerably lower than the typical operating frequency generated by the vehicle. However, a large decrease in seat stiffness ("softer" or lower spring rate) is required to achieve this.

Researchers have introduced several methods, such as the active, semi active, and nonlinear seat suspensions, to isolatelow-frequency vibrations fromoff-road vehicles.Maciejewskistudied the active vibration control strategy based on thereverse dynamics of a force actuator and a primary controller.Primary controller settings were calculated via multicriteria optimization, which subsequently defined vibroisolationcharacteristics of active suspension. Sun et al. investigated the problemof$H\infty$ control for active seat suspension systemsvia dynamic output feedback control. Stein et al. studiedthe seat vertical suspension system of a locomotive driverusing an adjustable damper. Choi et al. studied a semi activeseat suspension using an electrorheological fluid damper. Bouazara et al. studied the safety and comfort of a 3Dvehicle model with optimal nonlinear active seat suspensionand found that the comfort performance of a suspension seatwith semi active and active dampers could be considerablyenhanced by 20–30%.Active and semiactive seat suspensions possess complicatedstructures and require outside energy. By contrast,nonlinear seat suspensions do not require outside energyand exhibit simple structures. Le and Ahn designed andfabricated a vibration isolation model to improve vibration. According to the isolation theory the isolation system should have a lower natural frequency if the excitation frequency is low. The natural frequency formula is$\sqrt{k/m}$, where the body mass m of the driver is constant, while stiffness k should be small. A small system natural frequency may be obtained, but the static deformation of the system is large. A four-fold decrease in seat stiffness is necessary to reduce the natural frequency by 50%. Doing this however, now increases the likelihood that the suspension system will "bottom out."

Moreover, a high vehicle torque requirement also contributes to vibration at the operator deck. Passive seat suspension design includes selecting an appropriate natural frequency and optimization of damping according to a set stiffness. The consequence is a trade-off between isolation of vibration peak amplitudes at resonance and isolation of higher frequency vibrations. The potential benefit of a semi-active system is the removal of this trade-off since the system is designed to adjust the damping in real time.

In present research, for tracing the experimental results ofMR damper, polynomial model is selected. PID control strategy is used in present paper for semi-activecontrol of quarter vehicle model having three degrees of freedom.Theeffectiveness of the presented semi-active quarter vehicle systemin controlling the seat vibrations taking into accountis evaluated using simulation work in time domain while thevehicle travels over the bump type of input road profile.Simulation results are presented in graphical form andmathematical form which shows that the designed semi-activevehicle system with MR Bouc-Wen damper model provide much betterperformance in terms of passenger ride comfort and safetycompared to uncontrolled or passive suspension system.

2. MR DAMPER

MR damper typically consistsofa piston rod, electromagnet , accumulator, bearing, seal, and dampercylinder filled with MR fluidasshownin Fig. 1. The magneticfield generated by the electromagnet changes the characteristics of the MR fluid, which consists of small magnetic particles in non-conducting (magneti callyinert) afluidbase. Consequently,the strength of the electromagnet'sinputcurrent determines the physical characteristics of the MR dampers.The damperused in this research is the Lord RD-1005-3.Continuously variable damping is controlled by the increase in yield strength of the MR fluidin response to magnetic fieldstrength. In this damper,MR fluid flows from a high pressure chamber to alow pressure chamber throughanorificeinthepiston head. The damperis 209 mm long initsextended position,andthemain cylinderis38mmindiameter.The maincy linder houses the piston, the magnetic circuit, anaccumulator, and 50mlof MR fluid.

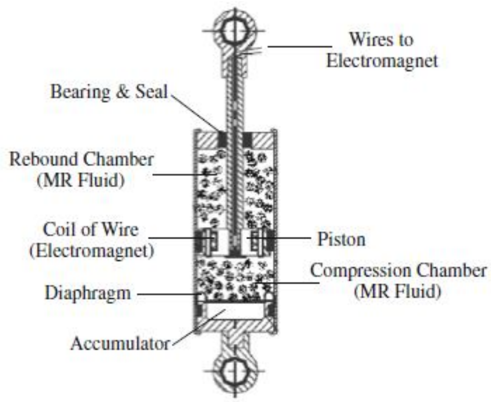


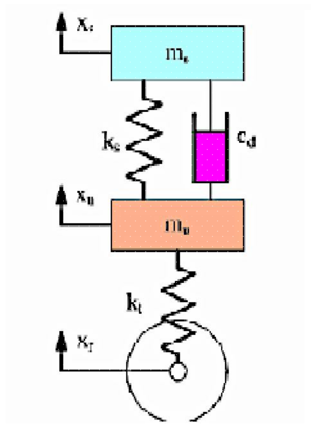
Fig 1. MR fluid damper

The damper has a752mm stroke. The magnetic field, which is perpendicular to the fluid flow, is generated by a small electromagnet in the piston head. The concept of MR damper operation is based on the MR effect which involves quick changes in viscosity of an MR fluid under the action of magnetic field (Rainbow, 1948). In this study, we employ an MR damper operating in the flow mode, which means that the produced damping force is controlled by the flow resistance of the MR fluid portion contained in the gap (Jolly *et al*., 1999). A schematic diagram of the MR damper is depicted in Fig. 1.

3. MATHEMATICAL FORMULATION OF AQUARTER- TRACTOR MODEL

To derive an equation of (vertical) motion of a vehicleinduced by terrain irregularities, we consider a quarter tractormodel by assuming that terrain roughness is evenlydistributed under all wheels of a vehicle and loading fromthe whole vehicle body is equally distributed across all ofits axles. In addition, we consider that a tire has small damping effect. With these preconditions, we draw thenext physical model (Figure 2 & 3) of the system for passivelyand semi-actively controlled systems of a quarter-tractormodel.a) Passive suspension designb) Semi-active suspension design

Most of the agricultural tractors in India do not have a suspension system. Between body of tractor and road there is only tires of tractor which damp out vibrations induced from road surface according to their stiffness. But in between driver seat and body of tractor, spring or spring and damper system is there to damp the vibrations. Quarter model of an agricultural tractor is shown in figure 2.



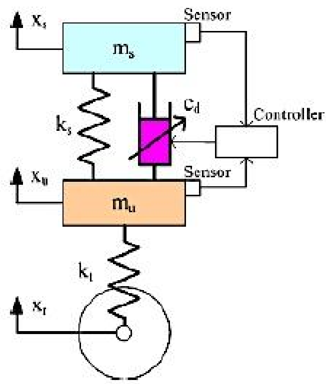


Fig2. Passive suspension system **Fig3. Semi-active suspension system**

From the passive and semi-active suspension designs of quarter-tractor model shown in Figure 2 and Figure 3, the equation of motion of the two mass bodies which are un-sprung mass (m_u) (i.e. half of axle mass and one wheel) and sprung mass (m_s) (i.e seat mass) are :

(a) Passive system

$$m_s\ddot{x}_s + c_d(\dot{x}_s - \dot{x}_u) + k_s(x_s - x_u) = 0$$
$$m_u\ddot{x}_u + c_d(\dot{x}_u - \dot{x}_s) + k_s(x_u - x_s) + k_t x_u = k_t x_t \quad \text{------(1)}$$

(b) Semi active system

$$m_s\ddot{x}_s + c_d(\dot{x}_s - \dot{x}_u) + k_s(x_s - x_u) = U_c \quad \text{\_\_\_\_\_\_\_(2)}$$
$$m_u\ddot{x}_u + c_d(\dot{x}_u - \dot{x}_s) + k_s(x_u - x_s) + k_t x_u = -U_c + k_t x_t$$

Wherex_s , $\dot{x}_s$ *and* $\ddot{x}_s$are displacement, velocity and acceleration of the sprung seat mass respectively: x_u , $\dot{x}_u$ *and* $\ddot{x}_u$are displacement, velocity and acceleration of the un-sprung mass (half of axle mass and one wheel), respectively;$c_d$is the damping coefficients ofseat suspension ; k_{se} and$k_t$are stiffness of seat suspension andtire; $r$is the terrain roughness (disturbancedisplacement;U_c is the force generated by the controller thattakes into account terrain irregularities r , that is thevertical displacement, the function of time and dependent ofthe vehicle speed. In the model (2), $U_c$is the control forceexerted by the controller.Our objective is to design such a controllable damper, capable of generating such control force$U_c$that responds to profile irregularities adequately by providing comfort ride. For the controller model of thedamper, we apply Bouc-Wenhysteresis effect model, and design numerical simulation model in MATLAB/Simulink.Mathematical formulations and simulation models of this model is expressed in the following section withrespect to a tractorseat suspension system.

4. SIMULATION MODELS OF THE BOUC-WENMRDAMPERS AND PID CONTROLLER

The MR damper with the Bouc-Wen model is composed ofstiffness (spring) element, passive damper and Bouc-Wenhysteresis loop elements. The schematic representation of theBouc-Wen model of the MR damper is depicted by the nextschematic view – Figure 4.

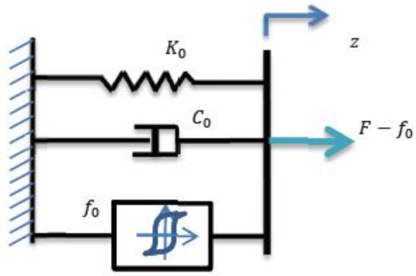


Fig4. Schematic diagram of Bouc - Wen model of MR damper

The hysteresis loop has aninternal variable γ that represents hysteretic behaviour andsatisfies the next expression (3). The model equation of theBouc-Wen model [13, 14] is expressed by the following.

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

Where y is the evolutionary variable that can vary from a sinusoidal to a quasi-rectangular function of the time depending on the parameters γ,β and A.

The force exerted by the MR damper is the function of the relative displacement and velocity, and the parameter α defined by the control voltage u and is given by

$F_{mr} = C_0(u)\dot{z} + K_0 z + \alpha(u)y + f_0$ ______________ (4)

In the model $K_0$is the stiffness of the spring element of the MR damper and the values of the parameters (coefficients) $C_0(u)$ and $\alpha(u)$have a linear relationship with the control voltage u and determine the influence of the model on the final forceF_{mr}. The force$f_0$takes into accountpre-yield stress of the damper. The values of the parameters (coefficients) $C_0(u)$ and $\alpha(u)$ are determined from the following expressions:

$C_0(u) = C_{0a} + C_{0b}u, \alpha(u) = \alpha_{0a} + \alpha_{0b}u$____________ (5)

The best-fit parameter values of these parameters are determined by fitting to the experimentally measured response of the system.

The simulation model of the system from Bouc-Wen model shown in Figure 5 is built in Simulink by using the equations (3), (4) and (5). The simulation model has two input sources x(t) displacement and dx(t) velocity of the seat mass of the system, and two output signals for control force F_{mr}going to the seat mass (m_s) with (-) sign and the unsprung mass (m_u) with the (+) sign. In this research, *dx(t)* is equal to $\dot{x}_t$ and F_{mr} is equal to $U_c$in equation (2) .

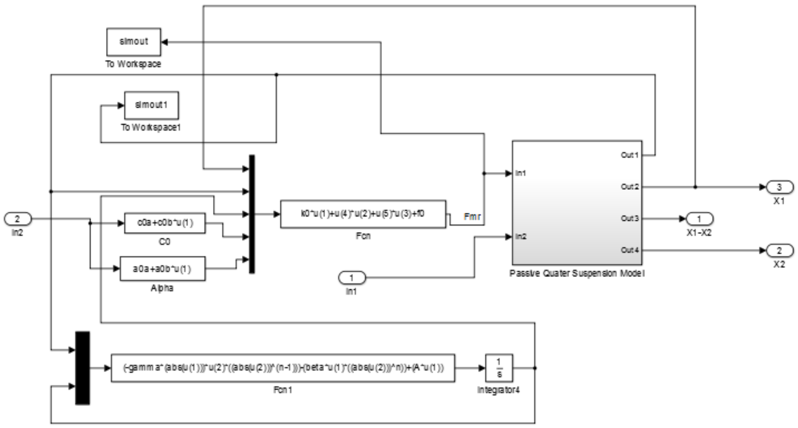


Fig5. Bouc-Wen model

PID controller has proved to be remarkably effective in controlling a wide range of processes. The use of PID controller does not require an exact process model and hence it is effective in controlling processes whose models are difficult to derive. Moreover, PID controller s are based on linear control theory andare much easier to understand, so they can be implemented very easily.Ziegler and Nichols method to tune the coefficients of a PID controller became the focus of research and become better understood, but cannot guarantee to be always effective. For this reason, this paper investigates the design of self-tuning for a PID controller. The transfers function of the PID controller given as:

$$k(s) = k_p + \frac{k_i}{s} + k_d s \qquad (6)$$

Where, $k_p$is proportional gain, $k_i$is the integral gain, $k_d$is the derivative gain.

Figure 6 illustrates the matlab/Simulink model of semi active suspension system with PID controller. The proportional, integral and derivative terms are summed to calculate the output of PID controller.

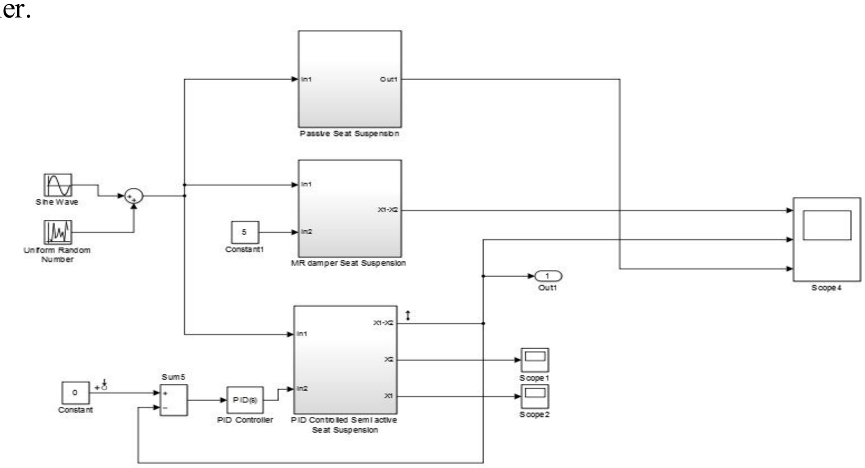


Fig 6. Passively controlled system vs MR damper controlled vs PID Controlled subsystem model

The final output defined by u(t) is given by

$$u(t) = k_p e(t) + k_i \int_0^t e(x)\, dx + k_d \frac{de(t)}{dt} \qquad (7)$$

Where, is the error present in the controller, t is the time or instantaneous time, and *x* is the variable of integration, taken from zero to present one. *E*

Simulation Results and Discussion

Selected parameters for simulation work of quarter tractor model with seat three-degrees-of-freedom with symbols and valuesare listed in Table 1. Random and sine wave of road input excitation,responses of the secondary suspension system havingpassenger seat using passive and semi-active suspensionsystem are shown in Fig. 7 while the calculate RMS and Peakvalues are tabulated in Table 1.

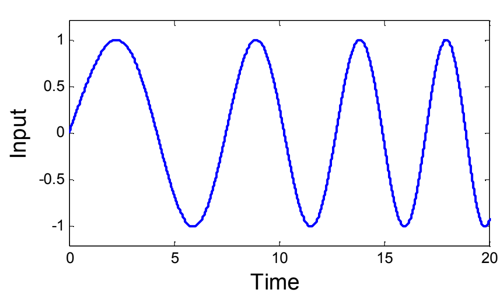
Table1: Parameters of quarter tractor model

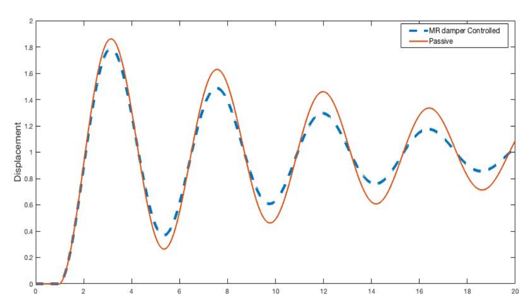
Driver seat mass, m_s	**75 [kg]**
Unsprung mass, m_u	**2800[kg]**
Seat suspension damping, c_d	**130 [N/m/s]**
Seat suspension stiffness, k_s	**8,000 [N/m]**
Tire stiffness, k_t	**90,000 [N/m]**

Table2: Data for Bouc-Wen model for simulation

Parameter of hysteresis shape; γ ,	**1,0,1.5,2**
β, A, n	**300 [N/m]**
Stiffness of spring element, K_0	**5 [V]**
Input voltage, *u*	**4400,442,10872,49616**
Other parameters; c_{0a} , $c_{0b}, \alpha_{0a}, \alpha_{0b}$	**0 [N]**
Pre-yield stress , f_0	

From Figs. 7 (b)-(e), it can be seen that controlled semiactivesuspension system integrated with MR damper and PID controller successfully suppresses the effects of vibrations as transmittedfrom unsprung mass to passenger seat resulting into improvedperformance related to passenger seat acceleration ($\ddot{x}_s$),passenger seat displacement (x_s)and secondary suspensionsystem deflection ($x_s - x_u$) compared to the uncontrolled orpassive suspension system showing the advantages of usingMR damper in semi-active suspension systems for vibration control system applications when passenger ridecomfort and safety issues are taken into account.





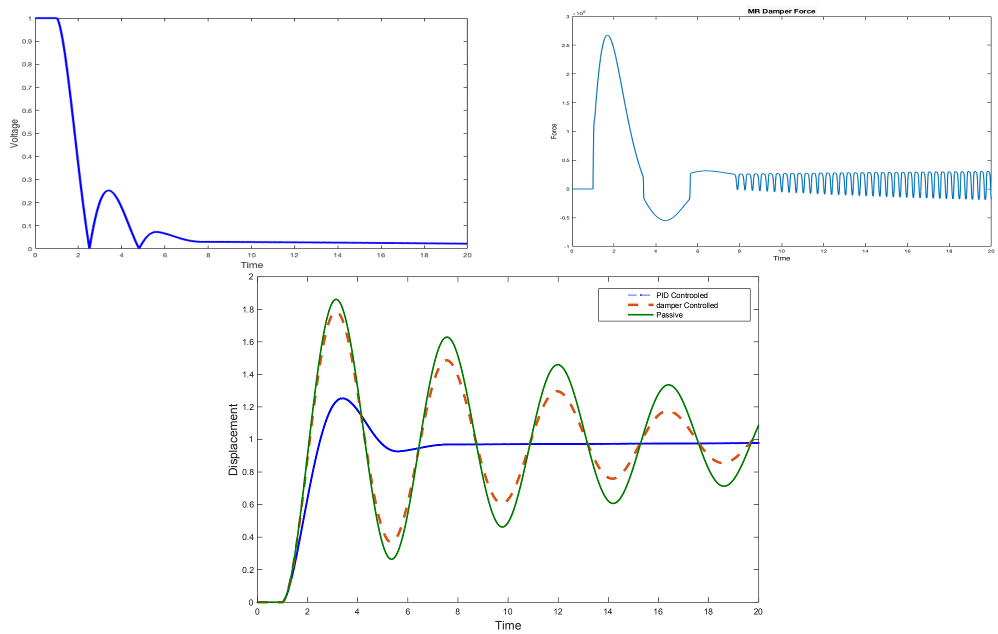


Fig.7 Time history plots of seat response under step road excitation with vehicle speed 8 km/hr (a) random road excitation (b) seat displacement (passive vs MR damper) (c) supplied voltage (d) desired damping force (e) seat displacement (uncontrolled vs PID controlled)

CONCLUSION

In present paper, passenger ride comfort issues has beenstudied through simulation work using quarter car model withthree degrees of freedom by taking acceleration anddisplacement factors into account. It has been observed thatsemi-active vehicle system integrated with MR damper provides better performance in terms of controlling vibrationeffects on driver when different parameters suchas passenger mass and road height are considered forsimulation purpose. The designed PID controller can besuccessfully implemented for proper working of assembledMR damper. Finally, it can be concluded that semiactivesuspension system provides much better performancecompared to passive suspension system for passenger ridecomfort and safety in travelling vehicles.

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