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Fuzzy controlled nonlinear Semi-active suspension systems

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ABSTRACT

Ride comfortability of vehicle is a big challenge for the automotive industries and design engineers. The suspension system is the major source of providing comfortability by absorbing vertical vibrations. This paper presents modeling of the non-linear semi-active suspension system equipped with Magneto Rheological damper. Simulation of the designed model is performed in Simulink. A comparative analysis of the Simulink response shows the difference between linear and non-linear model. It has also shown that fuzzy logic controller has improved the performance of the system.

Keywords: *Fuzzy Logic Controller, Magnetorheological damper, Matlab Simulink, Semi-active suspension, Sprung mass, Suspension systems, Unsprung mass.*

I. Introduction

Vibrations produced in vehicle because of the road irregularities is one of the main cause of ride uncomfortability. These vibrations are undesirable and should be reduced to a great extent. Suspension system is the main source of reduction in these vibrations. That is why the automotive manufacturing companies and engineers are paying much attention to the development of suspension system. The most widely used suspension system in the vehicles is the passive suspension system which cannot effectively suppress all the vibrations [1]. The stiffness and damping coefficient of passive suspension system are not adjustable for all types of vibrations [2]. On the other hand, active and semi-active suspension systems have the ability of adjusting their damping quality. All the basic components of real semi-active suspension system behave non-linearly for high velocities. Therefore non-linearities should be included in modeling for more realistic operation of vehicle [3]. In this study, a model for the non-linear semi-active suspension system is developed considering non-linearities in suspension stiffness, tire stiffness and MR damper. Non-linear model for MR damper is given by Bingham plasticity model as described by L. Kong et al [4]. The graphical results shows that proposed non-linear model deviates from the linear model. Also fuzzy controller shows an improvement in the vibration absorption.

II. Semi-active suspension system

2.1. Mathematical Modeling

A two degree of freedom quarter model for the semi-active suspension system equipped with Magnetorheological damper is shown in the Fig. 1.

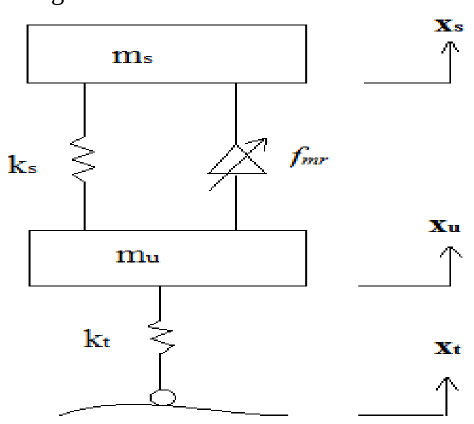


Figure 1: Semi-active suspension system

Vehicle mass is represented by m_s and tire mass mu k_s is the suspension stiffness, k_t is the tire stiffness and fmr is the force generated by MR damper. x_s is the sprung mass vertical displacement, x_u is the unsprung mass vertical displacement and x_t is the road displacement.

Dynamic equations of the system are derived from Fig.1 using Newton's second law of motion.

$$m_s\ddot{x}_s = -k_s(x_s - x_u) - f_{mr} \quad \textbf{(1)}$$

$$m_u\ddot{x}_u = k_s(x_s - x_u) - k_t(x_u - x_t) + f_{mr} \quad \textbf{(2)}$$

Considering non-linearities in these equations of motion, the equation of motion for the sprung mass becomes as

$$m_s\ddot{x}_s = -k_{s1}(x_s - x_u) - k_{s2}(x_s - x_u)^2 - k_{s3}(x_s - x_u)^3 - f_d \quad \textbf{(3)}$$

$$m_u\ddot{x}_u = k_{s1}(x_s - x_u) + k_{s2}(x_s - x_u)^2 + k_{s3}(x_s - x_u)^3 - k_{t1}(x_u - x_t) - k_{t2}(x_u - x_t)^2 - k_{t3}(x_u - x_t)^3 + f_d \quad \textbf{(4)}$$

where f_d is the non-linear damping force described by the Bingham plasticity model

$$f_d = f_{mr}sign(\dot{x}) + c_{p0}(\dot{x}) + f_0 \quad \textbf{(5)}$$

C_{p0} is the hysteretic damping coefficient, x is the displacement of the damper piston, f_{mr} is the force generated by MR damper and f_0 is the force of the accumulator.

2.2. Fuzzy Logic Controller

In this study a fuzzy logic controller is developed for the semi-active suspension system to control the damping force. Inputs to the fuzzy logic controller are the absolute velocity and relative velocity of the suspension system while output of the fuzzy logic controller is the damping force.

Trapezoidal membership functions are used for each of the input and triangular membership function is used for output. Each membership function is divided into three stages positive (P), zero (Z) and negative (N). Mamdani inference system is applied in this case and for transformation centroid method is used. Membership functions for input variables are shown in the fig. 2 and fig. 3. Triangular membership function for output damping force is shown in the fig. 4.

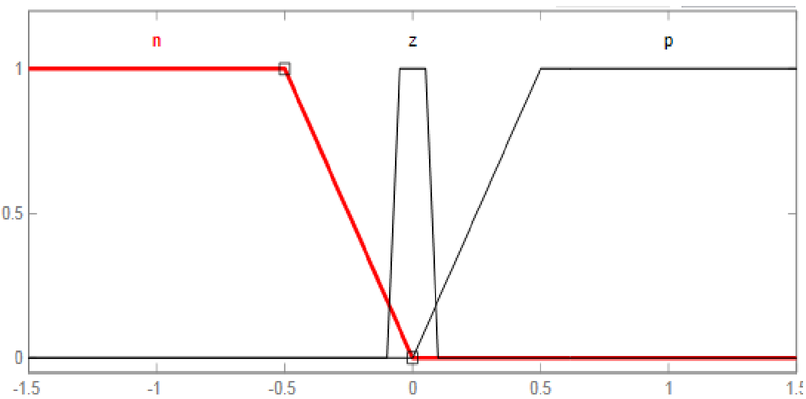


Figure 2: Absolute velocity membership function

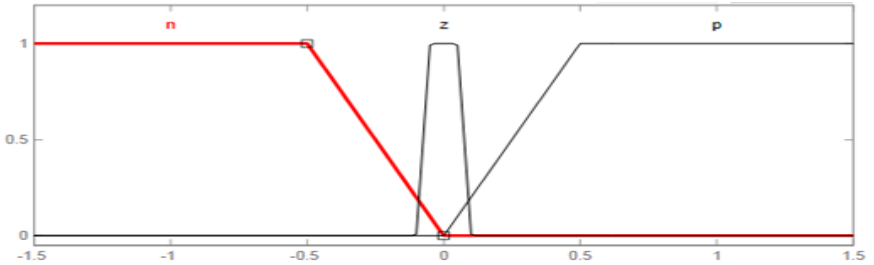


Figure 3: Relative velocity membership function

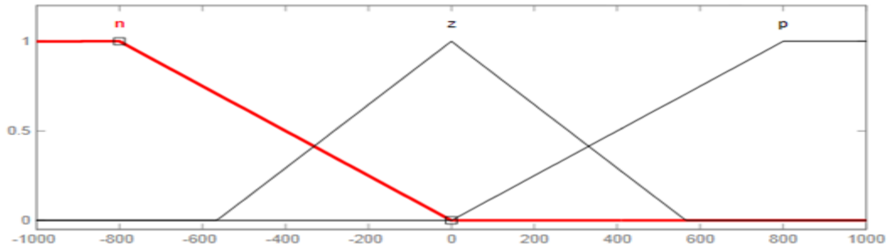


Figure 4: Output damping force membership function

2.3. Simulation

Simulation of quarter car model for both uncontrolled non-linear semi-active suspension system and fuzzy controlled semi-active suspension systems were carried using Matlab/Simulink. Both the models were excited by a step input of 0.5 m height from 0 to 1 second. Input used for the model excitation is step input shown in the Fig. 5.

$$x_t = \begin{cases} 0.5 & 0 \le t \le 1, \\ 0 & otherwise \end{cases} \qquad \textbf{(6)}$$

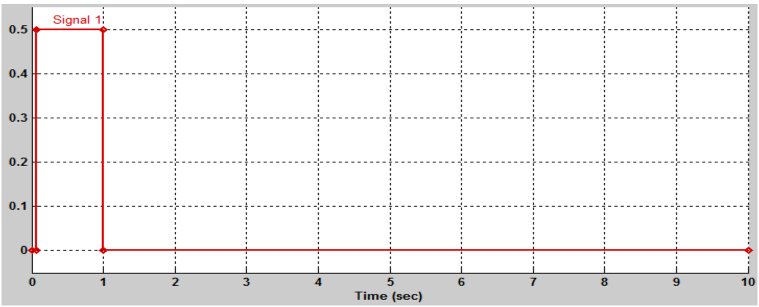


Figure 3: Step input

The values of suspension parameters are given in table 1 as described by [3], [5].

Table 1: Suspension parameters

Parameter	Values	Parameter	Values
k_{s1}	12.394 k N/m	k_{t3}	-0.1912 k N/m^3
k_{s2}	-73.696 k N/m^2	f_0	195.51 N
k_{s3}	3170.4 k N/m^3	c_0	735. 90 N-s/m
k_{t1}	5.5016 k N/m	m_s	240 kg
k_{t2}	0.520 k N/m^2	m_u	36 kg

III. Results and Discussions

The result of sprung mass displacement for the step input is shown in the Fig. 6 and Fig. 7. Fig. 6 shows a comparison between linear and non-linear semi-active suspension system and the result is tabulated in table 2. It is obvious that graph of the linear system deviates from the non-linear system.

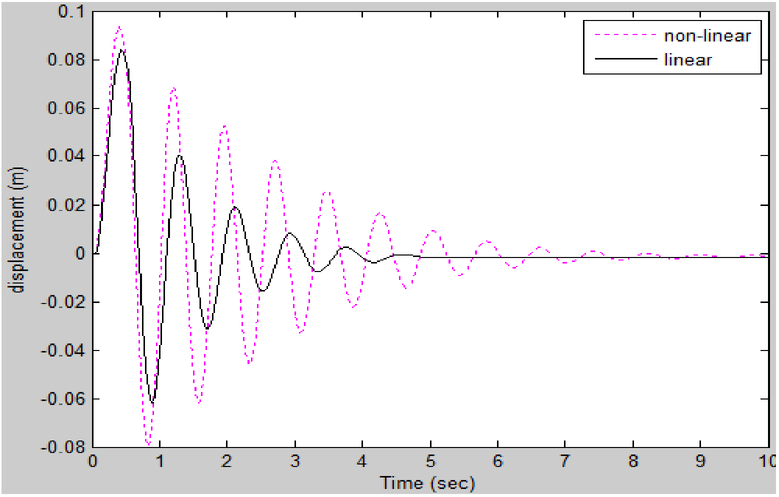
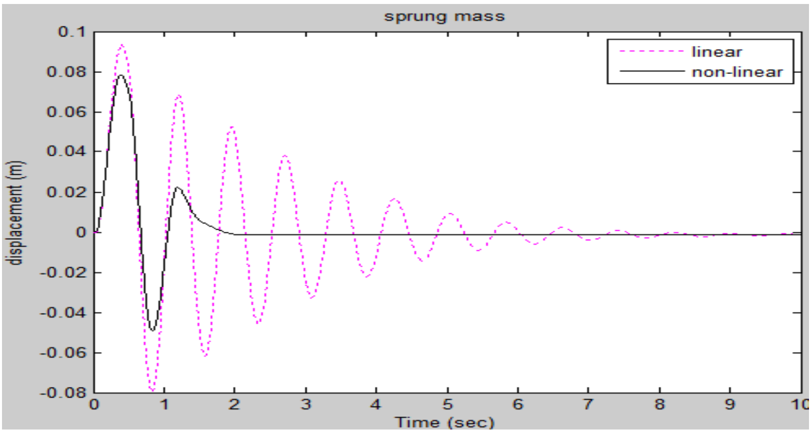


Figure 4: Linear vs nonlinear semi-active suspension system

Table 2: Comparative analysis of sprung and unsprung mass displacement

System	Max. Displacement (m)	Percentage Error
Linear	0.085	0.8 %
Non-linear	0.093	

Fig. 7 shows comparison of the graphical results of uncontrolled and fuzzy controlled non-linear semi-active suspension system. First, the vibration amplitude is reduced significantly and secondly the steady state response



of the system is achieved very early by the use of fuzzy controller.

Figure 7: Fuzzy controlled vs uncontrolled response

IV. Conclusions

The vibrations absorption capability of a semi-active suspension system was enhanced using fuzzy logic controller for continuous monitoring of the performance of the MR damper. The proposed controller has improved performance of the semi-active suspension system by reducing peak displacement and transient response of the vehicle body's mass.

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