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The Control of Seat Suspension System Using Fuzzy Controller on Vehicle Model

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Abstract: ***Matlab and Simulink are used as tools for developing the simulation model of the driver seat. The mathematical model was created according to the physical setup of the vehicle seat at the testing laboratory. Description of the seat, including its mechanical characteristics and mathematical model, is given in the paper. The seat value is used for the validation of control quality. The obtained simulations show that the developed suspension controllers provide superior passenger comfort for different types of road. To improve the ride comfort of car, this paper proposed a seat suspension with damper and designed a fuzzy controller with dynamics approximation of the damper. The performance of the feedback system with a fuzzy controller is tested in computer simulations and compared with the performance of an open loop system and the feedback system with controller. The obtained results are verified in laboratory experiments.The advantages of fuzzy control are proved in experimental investigations.***

Keywords: Vibration, Suspension, Transmissibility, Frequency

1. Introduction

All physical bodies, and structures, posses the mass and stiffness and are also subjected to motion which causes vibration. Vibration is always present in any operating machine and a vehicle is not an exception. Focusing on ride comfort of Occupants in a Bus, the vibration inputs coming directly from seat as well as the rattling body interiors are a major contributor to discomfort and fatigue. Issues like shaking rear view mirror combined with this fatigue, are also frustrating for Driver and is potential contributor to the accidents.In certain engineering cases vibration plays a paramount role, in others it is negligible and is of no interest from a technical point of view. To decide which type is the problem under consideration is a matter of vibration analysis and fine engineering judgment An automobile, is a combination of various dynamic systems that too working together under highly dynamic inputs. It is combination of tires, wheel and un-sprung masses, suspension, followed by flexible strength member like frame and body. There are internal inputs generators like engine and drive line. This complex system produces a vibration response based on road inputs, varying torque inputs and overall dynamic interaction of various element, which is finally perceived by subject receiver i.e. driver and passenger.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. 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.Vehicle drivers are frequently exposed to vertical vibration over a low frequency range, which is transmitted to the driver's seat from road roughness through the vehicle suspension and body.This vibration reduces ride comfort and can cause long-term health problems, as the human body is most sensitive to vertical vibration in the frequency range of 4–8 Hz. Consequently,seat suspensions are practicable and cost-effective solutions that are used in vehicles to overcome these effects. Three configurations of seat suspension exist in this field: passive, semi-active, and active. Passive systems are simple and cost-effective, consisting of spring and damper elements with fixed characteristics.They have limited vibration attenuation performance. Semi active suspensions are also comprised of spring and damper elements, but the properties of these elements can be adjusted using minimal power consumption. Their vibration attenuation performance is superior to passive systems. Their performance is still compromised, because they can only dissipate energy from the system. Active seat suspensions, which apply an external force based upon a control strategy, can significantly reduce vibration over a broadband frequency range.Therefore can be considered as an alternative to costly vehicle active suspension systems that are called upon to optimise both ride comfort and handling [2].

Fig.1. shows the flow of vibration in vehicles. In the bottom are shown the sources like road inputs, engine and Transmissibility components that Wheel and tires vibration in the system. While the road and rear axle and tyre wheel vibration area filtered to some extent by suspension properties, the engine vibration are filtered through the rubber fixes used for installations. Based on response of these rubber fixes and suspension the vibration inputs are transferred to the chassis frame from where it is transferred to the hood to same. Based on the mass and stiffness property and thus the natural frequency of body plus chassis frame structure and the internal/external components mounted on it like floor , seats ,windows, etc., the vibration level are amplified and thus finally perceived by the driver and passenger [3].

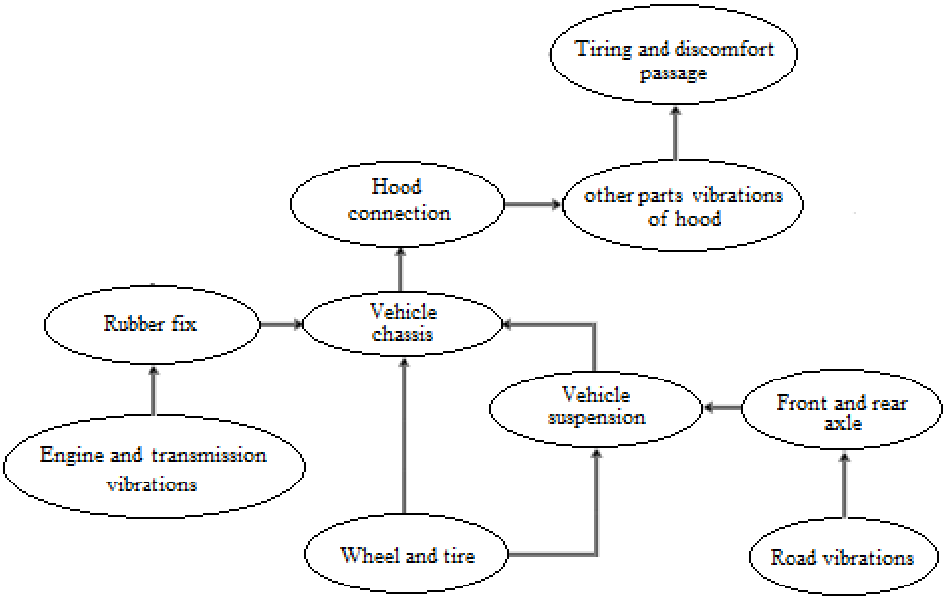

Figure 1: Follow diagram of vibration in vehicles [3]

2. Vibration and Transmissibility

Physical phenomena of solids in free vibration occurs when the vibrations are produced by instantaneous disturbances as a result of either a force or a deformation of the supporting spring.The initial disturbance does not exist after the vibration of the solid [4]. This can be shown by Fig.2.

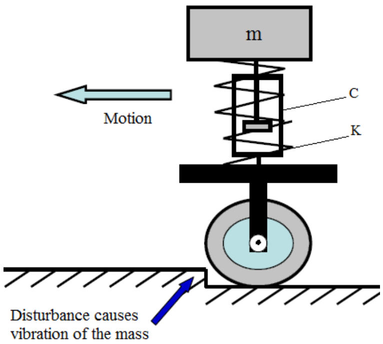

Figure 2: Physical model of a simple damped mass-spring system [4]

2.1. Vibration

What makes free-vibration of a mass-spring system to stop after some time t in reality is the damping effect. This is a more realistic phenomenon. Some of the sources of damping in mechanical vibrations include, resistance by the vibrating mass, and internal friction of the spring, during deformation. A simple damped mass-spring in vibration damping shown in Fig. 3. So this Fig.3. shows an application of spiral spring and amortiser in motorcycle [5].

Figure 3: Motorcycle suspension system [5]

Damper or dashpot of the other means Amortiser is automotive shock absorbe and damper, device that absorbs vibrations. A simple vibration and damping system shown in Fig.4. So this Fig.4. shows an application of ideal damping in amortiser [6].

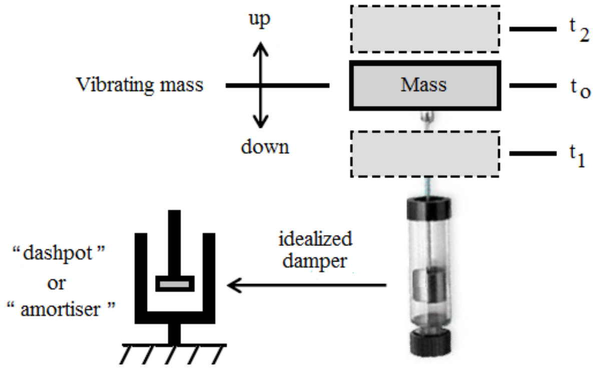

Figure 4: Vibrating and damper system[6]

2.1.1. Mathematical model

The corresponding damping force is related to air resistance to the movement of the mass. The resistance R is proportional to the velocity of the moving mass. Thus, mathematically equation ;

$$R(t) = c\,\frac{d\,y(t)}{dt} \tag{1}$$

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where *y(t)* is the distance the mass has travelled from its initial equilibrium position. Thus the damping force *R(t)*, is of the form. Where *c* = damping coecient.The mathematical expression of this physical model can be obtained by following similar procedure for the simple mass-spring, with the inclusion of the additional damping force. Newton's _first law in dynamic equilibrium gives [7]:

$$+\downarrow \sum Fy = -F(t) - R(t) - Fs + W = 0,$$

$$Fs = k\,[\,h+y\,(t\,)],\quad F(t) = m\,\frac{d^2\ y\,(t)}{dt^2}$$

Hence we have the following second order linear equation

$$m\,\frac{d^2\ y\,(t)}{dt^2} + c\,\frac{d\,y(t)}{dt} + k\,y\,(t\,) = 0$$

This is a second order homogeneous equation for the instantaneous position of the vibrating mass. Which is the same as:

$$\frac{d^2\ y\,(t)}{dt^2} + \frac{c}{m}\,\frac{d\,y(t)}{dt} + \frac{k}{m}\,y\,(t\,) = 0 \qquad (2)$$

We can compare it with

$$\frac{d^2\ y\,(t)}{dt^2} + v(t)\,\frac{d\,y(t)}{dt} + u(t)\,y\,(t\,) = g(t) \qquad (3)$$

where $v(t) = c/m$, $u(t) = k/m$ and $g(t) = 0$ which are all constant functions,hence continuous functions [8].

Thinking the vehicle model to be with high accuracy in analyzing the suspension dynamics, it is employed to model the suspension in this paper. Fig. 5 presented a three DOF model of vehicle suspension system containing the seat suspension with a damper. In this figure the vehicle body, seat and human body are included as the sprung masses: m_v and m_s, and the vertical dynamics of tire and axle is often considered by introducing an unsprung mass m_t and spring k_t. Damper is placed between the seat and the car body to form the seat suspension together with a spring and a damper [9]. Based on Newton second law, the dynamic equations of seat suspension system is:

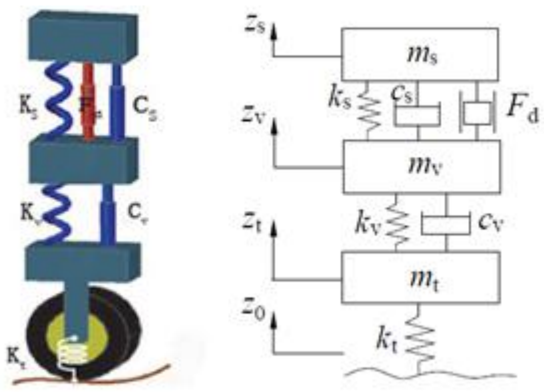

Figure 5: Model of suspension [9]

$$\begin{cases} m_s\ddot{z}_s = -c_s(\dot{z}_s - \dot{z}_v) - k_s(z_s - z_v) - F_d, \\ m_v\ddot{z}_v = c_s(\dot{z}_s - \dot{z}_v) + k_s(z_s - z_v) + F_d - c_v(\dot{z}_v - \dot{z}_t) - k_v(z_v - z_t), \\ m_t\ddot{z}_t = c_v(\dot{z}_v - \dot{z}_t) + k_v(z_v - z_t) - k_t(z_t - z_0), \end{cases} \qquad (4)$$

where m_t, m_v and m_s are unsprung mass, quarter car body mass and seat (plus human body) mass respectively. k_t, k_v and k_s are the stiffness coefficients of the tire, quarter car suspension and seat suspension respectively. c_v and c_s are the damping coefficients of vehicle suspension and seat suspension respectively. *Fd* is the damping force created by the damper. z_o, z_t, z_v and z_s are the road excitation, vertical displacements of car axle, body and seat, respectively.Based on Eq. (5), the state equation of the system is[10]:

$$\begin{cases} \dot{X} = AX + BU, \\ Y = CX + DU. \end{cases} \qquad (5)$$

Where

$$X = [x_1\ \ x_2\ \ x_3\ \ x_4\ \ x_5\ \ x_6]^T = [z_s\ \ \dot{z}_s\ \ z_v\ \ \dot{z}_v\ \ z_t\ \ \dot{z}_t]^T,$$
$$Y = [y_1\ \ y_2\ \ y_3\ \ y_4\ \ y_5\ \ y_6]^T = [z_s\ \ \dot{z}_s\ \ z_v\ \ \dot{z}_v\ \ \ddot{z}_s\ \ z_s - z_v]^T,$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ \frac{-k_s}{m_s} & \frac{-c_s}{m_s} & \frac{k_s}{m_s} & \frac{c_s}{m_s} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_s}{m_v} & \frac{c_s}{m_v} & \frac{-(k_s+k_v)}{m_v} & \frac{-(c_s+c_v)}{m_v} & \frac{k_v}{m_v} & \frac{c_v}{m_v} \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \frac{k_v}{m_t} & \frac{c_v}{m_t} & \frac{-(k_t+k_v)}{m_t} & \frac{-c_v}{m_t} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & 0 \\ \frac{-1}{m_s} & 0 \\ 0 & 0 \\ \frac{1}{m_v} & 0 \\ 0 & 0 \\ 0 & \frac{k_t}{m_t} \end{bmatrix}$$

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{-k_s}{m_s} & \frac{-c_s}{m_s} & \frac{k_s}{m_s} & \frac{c_s}{m_s} & 0 & 0 \\ 1 & 0 & -1 & 0 & 0 & 0 \end{bmatrix}$$

$$D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ \frac{-1}{m_s} & 0 \\ 0 & 0 \end{bmatrix}$$

$$U = \begin{bmatrix} F_d \\ z_0 \end{bmatrix}$$

2.2 Transmissibility

Transmissibility in the state of force-drived system is defined as the ratio of the force transmitted to the base to that of impression force on the system. For analysis it is customary to idealize structures, objects and isolation systems as simple mass-spring-damper systems as shown in Fig.6. The mass m is infinitely rigid. The spring is weightless and its stiffness is K. The damper, or dashpot, is weightless and its damping coefficient is C. Where F_T (t), Transmissibility force, m: mass, x(t), displacement of mass, C: damping coefficient K: stifness coefficient respectively as shown in Fig.6.

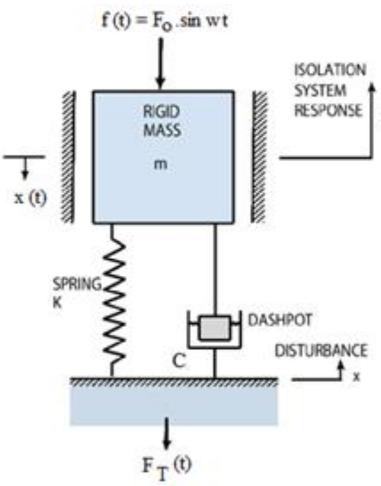

Figure 6: Simple typical isolatör system[11]

All objects vibrate when subjected to impact, noise or vibration. When the stimulation is removed, the object will experience periodic sinusoidal oscillations or free vibration at a frequency which is called its natural frequency w_n. With little or no damping, the natural frequency of a simple system such as Fig.6. is defined. If the simple system in Fig.6. is subjected to forced vibration at frequency w , and sinusoidal foundation motion at amplitude x , the absolute value of the mass response amplitude y expressed as a ratio (y/x) , also known as Transmissibility T. Transmissibility T is described rate of F_o and F_T this term shows in equation (6).

$$T = \frac{F_T}{F_o} = \frac{y}{x} = \frac{\sqrt{1+\left[2\xi\frac{w}{w_n}\right]^2}}{\sqrt{\left[1-\left(\frac{w}{w_n}\right)^2\right]^2+\left[2\xi\frac{w}{w_n}\right]^2}} \qquad (6)$$

Where F_T: supporting structure force, F_o : effect force to mass, W_n : undamped natural frequency, W : forcing frequency, ξ : damping factor, $\frac{w}{wn}$:frequency ratio, $\frac{F_T}{F_o}$: and y/x: transmissibility ratio.

Transmissibility - damping factor - frequency ratio curves shows in Fig. 7. Here , if $\frac{w}{wn} = 1$ and $\frac{w}{wn} < \sqrt{2}$, stiffness is decrease, damping is increase. If $\frac{w}{wn} < 1$ and $\frac{w}{wn} > \sqrt{2}$ stiffness is decrease sufficiently. If $\frac{w}{wn} > \sqrt{2}$ and $\frac{w}{wn} <$ 0.3, decrease both stiffness-damping and increase mass if feasible like increasing the clamping force of connection. If T< 0.3, no thing needed, but if T > 0.3, calculate the frequency ratio $\frac{w}{wn}$, for this value of transmissibility[12].

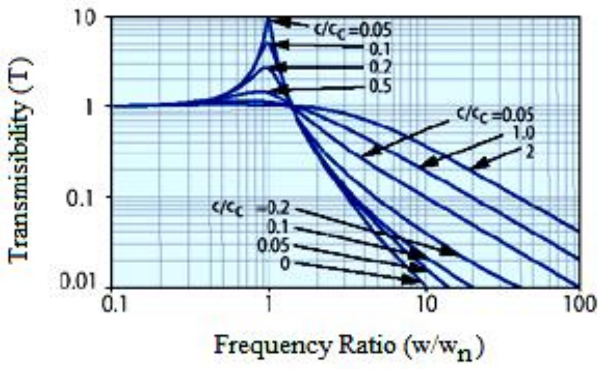

Figure 7: Transmissibility diagram

In a simple system, Damping serves as an energy dissipation "dashpot" to limit the magnification of system response. Actual damping c is conveniently referenced to critical damping c_c which is the value of damping at which a system will not oscillate when disturbed from equilibrium. critical damping is related to the system mass and natural frequency. Fig.7. is shown as a function of forcing frequency ratio w/w_n and critical damping ratio c/c $_c$. If external vibration is applied at a frequency which coincides with the Natural Frequency; i.e., $(w/w_n) = 1$, a condition of Resonance occurs. At resonance the system will experience very large potentially damaging magnification of the disturbing forces [12]. Isolation Efficiency E in percent transmission is related to Transmissibility as equation (7) :

$$E = 100\,(1-T)\% \qquad (7)$$

In the vibration of a system, the natural frequency, which causes free vibration, initially rises and peaks. Then it falls with the effect of damping. Transmissibility levels of this increase and decrease trend is called isolation efficieny. This meaning in Equation (7) is shown in Fig.8.

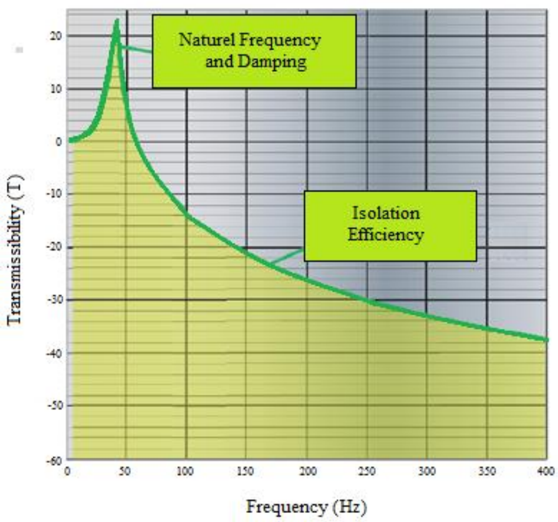

Figure 8: Isolation efficiency diagram

3. Fuzzy Logic Controller

In this study a fuzzy logic controller is developed for the 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. Equidistant domain partitioning method is used in fuzzy control generally. When the error is large, the system has sufficient error resolution, and it is shown as the "big error" dotted line in Fig. 9. When the error is small, the system response only changes around "ZO" corresponding to the original fuzzy partition, and other fuzzy subsets obviously do not work. Ideally, when the error is reduced, the domain of the fuzzy controller should be able to make self-adaptation adjustment. The accuracy of the fuzzy controller is related to the number of output variables and fuzzy rules.

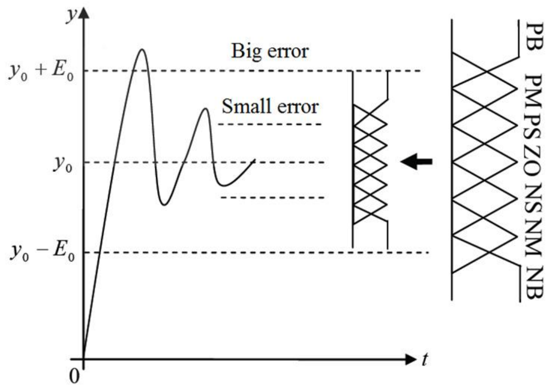

Figure 9: Adjustment of the domain

To evaluate the effectiveness of the proposed vehicle, A Simulink model is completed according to a certain model of vehicle parameters which are shown in Table 1.

Table 1: Simulation parameters of a model of vehicle

Parameter	Value	Unit
m_s	80	kg
m_v	400	kg
m_t	40	kg
k_s	8000	N/m
c_s	250	$N/(m \cdot s^{-1})$
c_{s1}	700	$N/(m \cdot s^{-1})$
c_{sh}	2000	$N/(m \cdot s^{-1})$
k_v	18500	N/m
c_v	1500	$N/(m \cdot s^{-1})$
k_t	185000	N/m
ε_{fuzzy}	2-5	–

4. Simulation Results

The result of sprung mass displacement for the step input is shown in the Fig. 10 and Fig. 11. shows a comparison between linear and non-linear vehicle suspension system and the result is tabulated in Table 1. It is obvious that graph of the linear system deviates from the non-linear system.

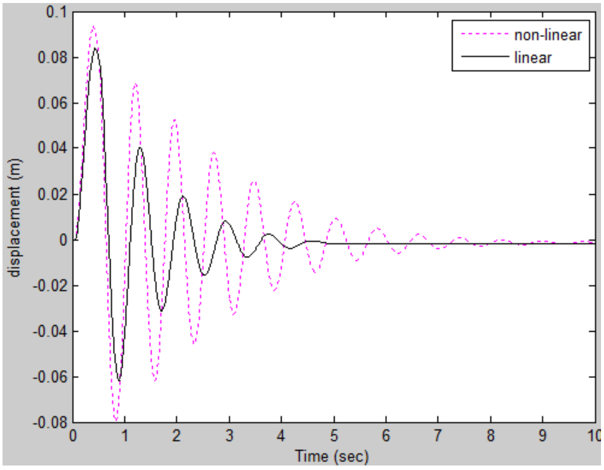

Figure 10: Linear vs nonlinear response

Fig. 11 shows comparison of the graphical results of uncontrolled and fuzzy controlled vehicle 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.

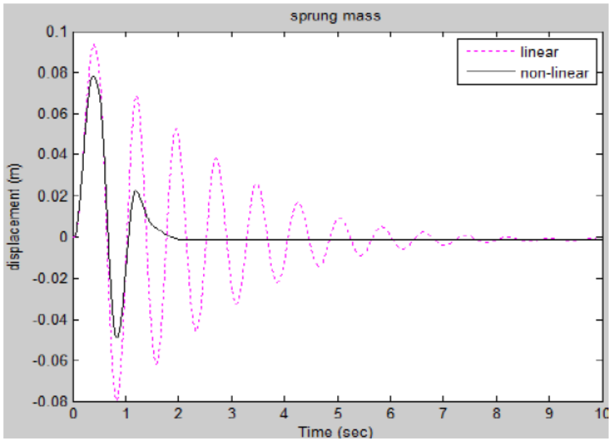

Figure 11: Fuzzy controlled vs uncontrolled response

5. Conclusion

Main objective in this paper is to study the points where the vibration in seat suspension system is occurring and how it gets damping. This paper dealt with the vibration transfers and damping. FL controllers for an active seat suspension have been developed, which use inexpensive and available preview information from the vehicle suspensions while satisfying the physical system constraints at a range of different operational conditions. The vibrations absorption capability of seat suspension system was enhanced using fuzzy logic controller for continuous monitoring of the performance of the damper. The vibrations absorption capability of a vehicle 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 vehicle suspension system by reducing peak displacement and transient response of the vehicle body's mass.

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