Effect of Irregular Road on Dynamic Response of Car Seat Suspended by a Magneto-Rheological (MR) Damper

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Abstract

Ride with comfort is the most important demand in vehicle industry as it affects directly seats mounting. For that reason a lot of effort has been made to improve ride comfort by isolating vehicle from road irregularities as possible. That can be done by improving vehicle suspension ability to reduce the vibrations induced from road irregularities during vehicle maneuver before it reaches the vehicle seat. This work investigates the effect of active damping on vehicle vibrations by presenting mathematical model of full active vehicle suspensions. That model introduced a new active damper tested recently and proved to be efficient. These dampers depend on kind of Magneto-Rheological fluid (MR). Also a mathematical model is derived for full vehicle its seat is suspended by a classical damper or Magneto-Rheological (MR) damper. Many types of roads are tested for vibration effect on the driver and suspension of the vehicle. These roads are of types repeated pulses, repeated trapezoid bumped, repeated half sine, repeated cosine and repeated simple harmonic. Also another three different roads are taken into consideration (half sine, ramp, and sudden step type). Computer programs are constructed which include a classical damper or Magneto-Rheological (MR) damper and can be implemented on a full vehicle suspension. Also a simulation program (MATLAB-SIMULINK) from which any response to any changes in any parameter can be obtained. Time response curves for all types of roads are plotted for both classical and (MR) damper. A comparison between the types of roads as effect for seat (heave), and axle suspension was studied. Also the effect of voltage input to (MR) damper on vibration of the vehicle seat was investigated.

Keyword: Magneto -Rheological fluid, Vehicle, Seat, Damper, Bumped, Road, Modeling

1. Introduction

Passive seat suspensions have long demonstrated that they are suboptimal single degree of freedom (SDOF) suspension systems. Once the spring rate has been chosen, thus setting the natural frequency, the other parameter that needs to be chosen is the system damping. Insufficient damping provides poor resonance control, but good isolation at high frequencies. Conversely, large damping results in good resonance control while sacrificing high frequency isolation.

Lord Corporation has been working on semi active suspensions for the past two decades. They have recently begun to apply the technology of MR fluids to semi active dampers. One potential application of MR semi active dampers is the seat suspension commonly found on heavy vehicles. This market has been fueled by the ever-tightening standards aimed at protecting the health and well-being of the vehicle operators. Lord has developed several control policies for controlling the damper in MR seat suspensions. Lord engineers also noticed that when comparing output acceleration frequency spectra to input spectra, the seat seemed to be responding at frequencies that did not exist in the input.

(Lieh 1993) explored the use of semi active suspensions to control the dynamics of a full car model. He concluded that the use of the skyhook control policy reduces the root mean square (RMS) acceleration of the car body while increasing the RMS tire forces. Two studies, one by (Yi and Hedrick 1993) and the other by (Valesek et al. 1997) were found to be in the area of dynamic tire loading. Both studies were interested in methods to reduce the dynamic tire loading of a vehicle in order to reduce the amount of road damage that it causes. Three other studies focused on control methods that are able to teach themselves how to control the semi active suspension system.

(Cheok and Huang 1989) and (Yoshimura et al.1997) used fuzzy logic and neural network methods to teach the controller, while (Frost et al.1996) used a moderated reinforcement learning technique. All three studies show that there are benefits associated with these types of control approaches. The next study by (Margolis 1982) examined the effects of using realistic feedback signals when controlling active and semi active suspension systems. This is an analytical study that suggests several feedback strategies for the semi active suspension system so that the performance can approach the fully active suspension performance.

In another study by (Margolis1983) a procedure was outlined to examine the feasibility of using semi active or active vibration isolation instead of purely passive approaches. (Hwang et al. 1997) presented an interesting method for testing the semi active damper hardware without using a complete vehicle. They explored the test method known as hardware in the loop simulation. Essentially, the dynamic model of the system is coded for simulation in a computer. The piece of hardware under test (i.e., the semi active damper) is excited according to the computer numerical simulation, and the response of the hardware is measured and fed back into the computer to complete the simulation. (Jezequel and Roberti 1996) investigated an optimal preview of the suspension system which is useful for trains: once the lead train passes over a point, the rest of the train gets knowledge of the track conditions.

(Miller 1988) explored the effects of the levels of both on-state and off-state damping on the performance of the quarter car semi active suspension system. (Bellizzi and Bouc 1995) and (Hrovat, et al.1988) studied optimal control techniques for the semi active suspension, while (Tibaldi and Zattoni 1996) studied the robustness of linear quadratic Gaussian control techniques.(Lazareva and Shitik 1997) studied the properties of MR fluids that are based on barium and strontium ferrites and iron oxides. The fluids were prepared using various combinations of the materials, and their properties, such as the MR effect, were studied.

(Ashour et al) studied the effects of components of the MR on sedimentation of the magnetic particles and initial viscosity. An attempt was made to optimize the composition of the fluid such that the fluid had the desired properties. In another study, (Ashour et al.1996) studied the general composition of MR fluid along with the methods that are used to evaluate the performance of the fluids. There is also an introduction to the fundamental MR devices to exploit the MR effect. The next three studies explored the design of MR fluid devices. (Carlson et al) studied the advantages of MR over ER fluid devices in areas such as the yield strength, the required working volume of fluid, and the required power. The operational modes of the MR fluid are presented along with the linear fluid damper, the rotary brake, and the vibration damper. (Kordonsky 1996) developed the concept of the MR converter (or valve) and applied the MR converter to create devices such as the MR linear damper, the MR actuator, and the MR seal. Finally (Bolter and Janocha 1997) examined the rules that should be applied when designing the magnetic circuit for MR devices that are working in the different modes of the MR fluid. Bolter also examined the use of permanent magnets in the design of the magnetic circuit to change the operational point of the MR device.

The final Magneto-rheological article by (Jolly et al.1996) presents a model based on dipolar interaction of particles, and is used to predict the behaviour of both MR fluids and MR elastomers. The model is compared to experimental data and it is shown that the model is semi-empirical in that it must be fit to the experimental data by adjusting a parameter that accounts for a modeled magnetic interactions. (Ballo 1995) analytically derived the required power usage for active seat suspension systems. The first system is actually a combination of a hydraulic actuator and passive suspension in series, while the second system is an active electro-pneumatic system which controls vibrations by controlling the air flow in and out of a pneumatic spring.

(Grimm et al.1974) designed a fully active seat suspension for farm vehicles using a hydraulic actuator. They used a simple compensator to control the actuator using the relative displacement of the seat and either of the base or seat acceleration. (Nevala et al. 1996) considered the design of the seat suspension even further.

They designed and mathematically modeled a 2DOF seat suspension that not only allows for vertical translation of the seat, but also allows the seat to rotate fore and aft. Even though the seat is completely modeled, there is no mention of a control strategy for the novel suspension. The seat suspension was studied by (Wu and Griffin 1997) .They focused on reducing end stop impacts on a seat suspension. They accomplished this by using a two- state damper. They have experimented with changing the size of the bands and observing the effects on the number of end stop hits.

This work investigates the effect of active damping on vehicle vibrations by presenting a mathematical model of full active vehicle suspensions. Mathematical model is derived for full vehicle suspended by a classical damper and Magneto-Rheological (MR) damper. Many types of roads are tested for vibration effect on the driver and suspension of the vehicle. These roads are (repeated pulses road, repeated trapezoid bumped road, repeated half sine road , repeated cosine road and repeated simple harmonic road. Also another three different roads are taken into consideration (half sine road, ramp and sudden step type). Computer programs are constructed which include a classical damper or Magneto-Rheological (MR) damper and can be implemented on a full vehicle suspension.

Also a simulation program (MATLAB-SIMULINK) from which any response to any changes in any parameter can be obtained. Time response curves for all types of roads are plotted for both classical and (MR) damper. A comparison between the types of roads as effects of seat heave, and axle suspension was studied. Also the effect of voltage input to (MR) damper on vibrations of the vehicle was investigated.

2. Spencer Model

The seat is assumed to be suspended by an MR damper .The damper model is chosen as (Spencer et al. 1997). The structure of this model is shown in Fig. 1

Damping force in Spencer's model can be expressed as

$$F = \alpha z + c_o(\dot{x} - \dot{y}) + k_o(x - y) + k_1(x - x_o) \tag{1}$$

Or can be also written as

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

The displacements *z* and *y* are respectively defined as

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

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

The adjustment of hysteresis parameters α, β, and A determines the linearity in the unloading region as well as the transition smoothness from the pre-yield to the post-yield region. The parameters γ, β, A, n and k_1 are considered fixed and the parameters c_o, c_1 and α are assumed to be functions of the applied voltage *u*.

$$\alpha = \alpha_o + \alpha_b u \tag{5}$$

$$c_1 = c_{1a} + c_{1b} u \tag{6}$$

$$c_o = c_{oa} + c_{ob} u \tag{7}$$

$$\dot{u} = \eta(u - v) \tag{8}$$

c_1 is the viscous damping at higher velocities , k_1 is the stiffness representing the accumulator, k_0 is the stiffness at higher velocities , x_o is the initial deflection of the accumulator gas spring , η is the time constant , α is the Bouc- Wen parameter describing the MR fluid yield stress and υ is the command voltage sent to the current driver. Table 1 show the parameters values of the model

Table 1: Parameters of Spenser model

Parameter	Value	Units
c_{oa}	785	N.s/m
c_{ob}	1803	N.s/Vm
k_o	3610	N/m
c_{1a}	14649	N.s/m
c_{1b}	34622	N.s/Vm
k_1	840	N/m
x_o	0.0245	m
α_a	12441	N/m
α_b	38430	N/Vm
γ	136320	m^{-2}
β	2059020	m^{-2}
A	58	
n	2	
η	190	s^{-1}

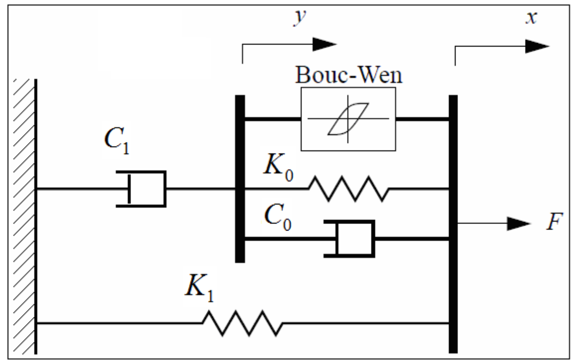

Fig. 1 Spencer MR damper model

3. Vehicle Model Suspension System

The non-linear full car model used in this study is shown in Fig. 2. This full car model has eight degrees of freedom; its variables are z_{u1}, z_{u2}, z_{u3}, z_{u4}, z_{u5}, z, θ and Φ.

These are the motion of the right front axle, the motion of the left front axle, the motion of the right rear axle, the motion of the left rear axle, the bounce motion of the passenger seat, the bounce motion of the vehicle body, the pitch motion of the vehicle body and the roll motion of the vehicle body, respectively.

The aim is to improve the ride comfort of the passengers. The common application in modeling the vehicle with a passenger seat is to add only one passenger seat preferably in the driver seat position though considering only one suspended seat implies that other seats are assumed to be fixed rigidly to the chassis.

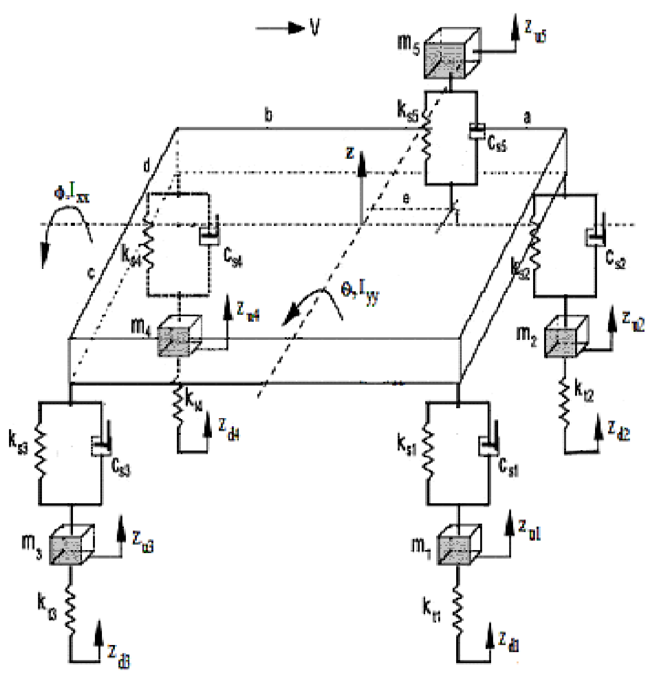

Fig.2: The Non-Linear Full car Model with a Passenger seat (Baumal et al 1998)

Equation of motion for heave at center of gravity of vehicle can be written as:

$$M\ddot{z} = k_{s1}z_{u1} + k_{s2}z_{u2} + k_{s3}z_{u3} + k_{s4}z_{u4} + k_{s5}z_{u5} - [a(k_{s1} + k_{s2}) - b(k_{s2} + k_{s4}) + ek_{s5}]\theta$$
$$- [d(k_{s2} + k_{s4}) - c(k_{s1} + k_{s3}) + fk_{s5}]\phi + c_{s1}\dot{z}_{u1} + c_{s2}\dot{z}_{u2} + c_{s3}\dot{z}_{u3} + c_{s4}\dot{z}_{u4}$$
$$- [c_{s1} + c_{s2} + c_{s3} + c_{s4}]\dot{z} - [a(c_{s1}+c_{s2}) - b(c_{s3} + c_{s4})]\dot{\theta} - [d(c_{s2}+c_{s4}) - c(c_{s1}+c_{s3})]\dot{\phi} + F_{mr} \quad (9)$$

Equation of motion for pitch at center of gravity around Y-axis clockwise can be written as:

$$I_{yy}\ddot{\theta} = ak_{s1}z_{u1} + ak_{s2}z_{u2} - bk_{s3}z_{u3} - bk_{s4}z_{u4} + ek_{s5}z_{u5} - [a(k_{s1} + k_{s2}) - b(k_{s3} + k_{s4}) + ek_{s5}]z$$
$$- [a^2(k_{s1} + k_{s2}) + b^2(k_{s3} + k_{s4}) + e^2K_{s5}]\theta - [d(ak_{s2} - bk_{s4}) - c(ak_{s1} - bk_{s3}) + efk_{s5}]\phi$$
$$- [a(c_{s1} + c_{s2}) - b(c_{s3} + c_{s4})]\dot{z} - [a^2(c_{s1} + c_{s2}) + b^2(c_{s3} + c_{s4})]\dot{\theta}$$
$$- [d(ac_{s2} - bc_{s4}) - c(ac_{s1} - bc_{s3})]\dot{\phi} + ac_{s1}\dot{z}_{u1} + ac_{s2}\dot{z}_{u2} - bc_{s3}\dot{z}_{u3} - bc_{s4}\dot{z}_{u4} + eF_{mr} \quad (10)$$

Equation of motion for roll at center of gravity around X-axis clockwise can be written as:

$$I_{xx}\ddot{\phi} = -ck_{s1}z_{u1} + dk_{s2}z_{u2} - ck_{s3}z_{u3} + dk_{s4}z_{u4} + fk_{s5}z_{u5} - [d(k_{s2} + k_{s4}) - c(k_{s1} + k_{s3}) + fk_{s5}]z$$
$$- [d^2(k_{s1} + k_{s2}) + c^2(k_{s3} + k_{s4}) + f^2K_{s5}]\phi - [d(ak_{s2} - bk_{s4}) - c(ak_{s1} - bk_{s3}) + efk_{s5}]\theta$$
$$- [d(c_{s2} + c_{s4}) - b(c_{s1} + c_{s3})]\dot{z} - [d^2(c_{s2} + c_{s4}) + c^2(c_{s1} + c_{s3})]\dot{\phi}$$
$$- [d(ac_{s2} - bc_{s4}) - c(ac_{s1} - bc_{s3})]\dot{\theta} - cc_{s1}\dot{z}_{u1} + dc_{s2}\dot{z}_{u2} - cc_{s3}\dot{z}_{u3} + dc_{s4}\dot{z}_{u4} + fF_{mr} \quad (11)$$

Equation of motion for (front – right) unsprung mass can be written as:

$$m_1\ddot{z}_{u1} = -(k_{s1} + k_{t1})z_{u1} + k_{s1}z + ak_{s1}\theta - ck_{s1}\phi - c_{s1}\dot{z}_{u1} + c_{s1}\dot{z} + ac_{s1}\dot{\theta} - cc_{s1}\dot{\phi} + k_{t1}z_{d1} \quad (12)$$

Equation of motion for (front – left) unsprung mass can be written as:

$$m_2\ddot{z}_{u2} = -(k_{s2} + k_{t2})z_{u2} + k_{s2}z + bk_{s1}\theta + dk_{s1}\phi - c_{s2}\dot{z}_{u2} + c_{s2}\dot{z} + bc_{s1}\dot{\theta} + dc_{s1}\dot{\phi} + k_{t2}z_{d2} \quad (13)$$

Equation of motion for (rear – right) unsprung mass can be written as:

$$m_3\ddot{z}_{u3} = -(k_{s3} + k_{t3})z_{u1} + k_{s3}z - bk_{s3}\theta - ck_{s3}\phi - c_{s3}\dot{z}_{u3} + c_{s3}\dot{z} - bc_{s3}\dot{\theta} - cc_{s3}\dot{\phi} + k_{t3}z_{d3} \quad (14)$$

Equation of motion for (rear – left) unsprung mass can be written as

$$m_4\ddot{z}_{u4} = -(k_{s4} + k_{t4})z_{u4} + k_{s4}z - bk_{s4}\theta + dk_{s4}\phi - c_{s4}\dot{z}_{u2} - c_{s4}\dot{z} - bc_{s4}\dot{\theta} + dc_{s4}\dot{\phi} + k_{t4}z_{d4} \quad (15)$$

Equation of motion for the seat mass can be written as:

$$m_5\ddot{z}_{u5} = -k_{s5}z_{u5} + k_{s5}z + ek_{s5}\theta + fk_{s5}\phi - F_{mr} \tag{16}$$

For conventional damper the damping force is as

$$F_{mr} = c_{s5}\dot{z}_{u5} - c_{s5}\dot{z} - ec_{s5}\dot{\theta} - fc_{s5}\dot{\phi} \tag{17}$$

While the damping force of MR damper can be obtained from equation 1

Where a, b are the distances of front and rear axle to the center of gravity of the vehicle body ,c, d are the distances of unsprung masses to the center of gravity of the vehicle body ,e, f are the distances of passenger seat to the center of gravity of the vehicle body , c_{s1} and c_{s2} are the damping coefficient of suspension at front wheel,c_{s3} and c_{s4} are the damping coefficient of suspension at rear wheel ,c_{s5} is the damping coefficient of passenger seat.

Also ,k_{s1} ,k_{s2} are the spring constant of suspension for front wheel , k_{s3} ,k_{s4} are the spring constant of suspension for rear wheel , k_{s5} is the spring constant of passenger seat,k_{t1} , k_{t2} are the stiffness coefficient of tire, m_1, m_2 are the mass of axle at front m_3, m_4 are the mass of axle at rear, m_5 is the mass of the passenger,z_{d1},z_{d2} are the road excitation under front wheel , z_{d3} ,z_{d4} are the road excitation under rear wheel , I_{yy} is the mass moment of inertia of the vehicle body for pitch motion, I_{xx} is the mass moment of inertia of the vehicle body for roll motion , M is the mass of the vehicle body.

Table 2: The Different Parameters of the Vehicle

Parameter	Value	Units
M	1100	kg
I_{yy}	1848	$kg.m^2$
I_{xx}	550	$kg.m^2$
m_1,m_2,m_3,m_4	45	kg
m_5	90	kg
k_{s1},k_{s2}	15000	N/m
k_{s3},k_{s4}	17000	N/m
k_{s5}	15000	N/m
c_{s1},$c_{s2}c_{s3}$,c_{s4}	2500	N.s/m
c_{s5}	150	N.s/m
$k_{t1=}k_{t2}=k_{t3}=k_{t4}$	250000	N/m
a	1.2	m
b	1.4	m
c	0.5	m
d	1	m
e	0.3	m
f	0.25	m

4. Results and Discussions

4.1 Road Excitation

Five different road types were considered in this work. The distance of bumped road is assumed to be 1.7 m and its height is 0.05 m .These types are (repeated pulses road, repeated trapezoid bumped road, repeated half sine road , repeated cosine road and repeated simple harmonic road. Figures 3, 4, 5, 6 and 7 shows the different shapes of these of roads .In this section theoretical results are represented using simulation programs to investigate the effect of active damping on car ride comfort (seat motion).

The values of car parameters are extracted from a (DUGAN) or (SHAHIN) car type. For comparison purposes, the (Simulink) program is running in two cases the first one, when the car seat is suspended by conventional damper and the second case when the car seat is suspended by (MR) damper.

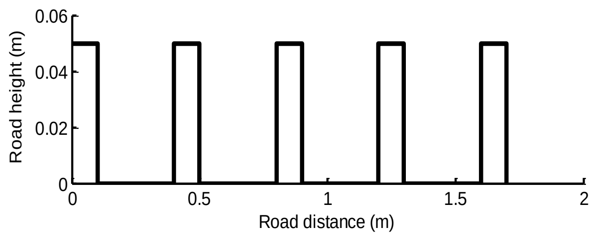

Fig. 3: Repeated pulses road

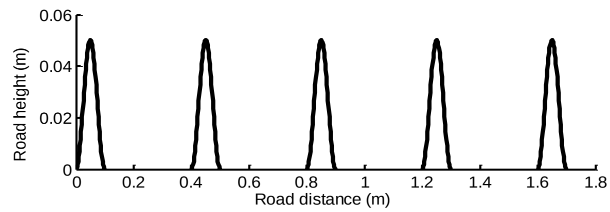

Fig.6 Repeated cosine road

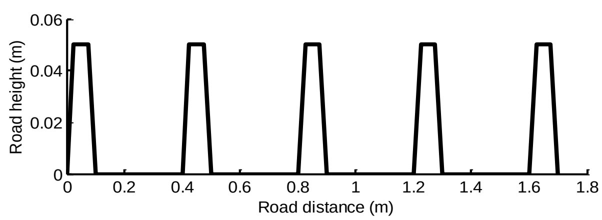

Fig. 4 Trapezoid bumped road

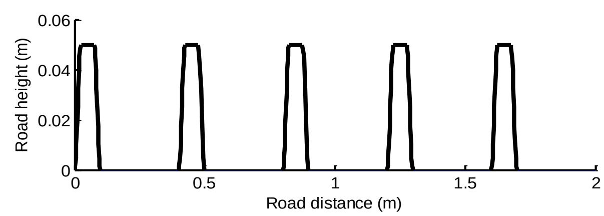

Fig. 7 Repeated simple harmonic road

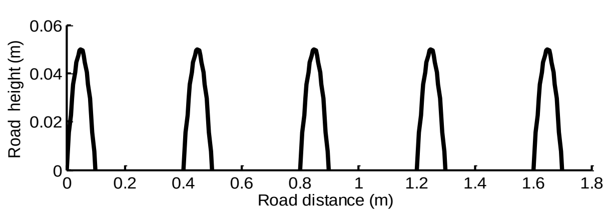

Fig. 5 Repeated half sine road

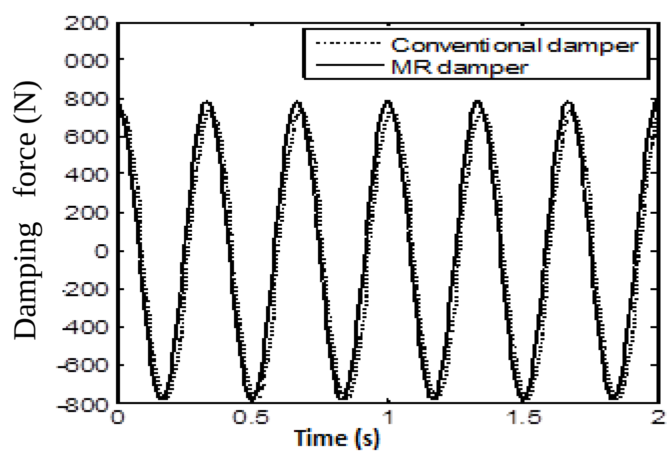

Fig. 8 Damping force vs. time for (MR) and conventional damper.

In order to compare the damping performance of (MR) damper with that of a conventional viscous damper (classical type), an equivalent damping coefficient for classic damper is determined by equating the energy dissipated during full cycle for conventional and (MR) damper.

The equivalent damping coefficient is chosen to be,c_{s5} = 830 N.s/m.

Figure 8 presents the variation of damping force during a period of 2 s for both (MR) and conventional damper.

Figures 9, 10, 11, 12 and 13 shows the time history of (car seat) displacement and comparison between (MR) and conventional damper along three seconds when the car moves on different types of road with a velocity of 50 km/hr.

As mentioned before five types of roads are taken into consideration (repeated pulses road, repeated trapezoid bumped road, repeated half sine road, repeated cosine road and repeated simple harmonic road. It is shown from these figures that, the unsteady vibrations for the car appear during a period of time its value changes according to the road type.

Also it is clear from these figures that, cosine road gives smallest amplitude of vibrations comparing to the other roads.

Also, it is be noticed that, using MR damper realistic motion is achieved because the displacement of vibrations changes gradually during a period of unsteady motion and before it reaches the steady state case, but the vibrations amplitude changes suddenly when using conventional damper.

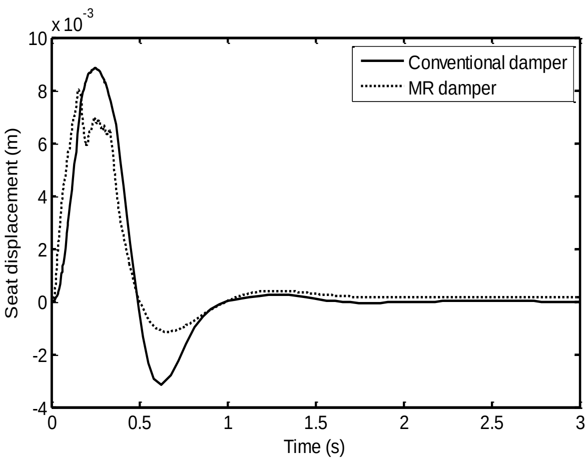

Fig.9 Time history of seat displacement for repeated pulses road, v=50 km/hr, E=2 V, s=1.7m.

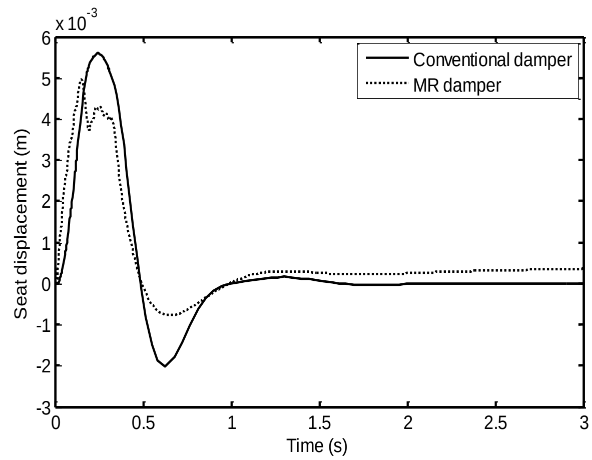

Fig.11 Time history of seat displacement for repeated half sine road, v=50 km/hr , E=2 V, s=1.7m.

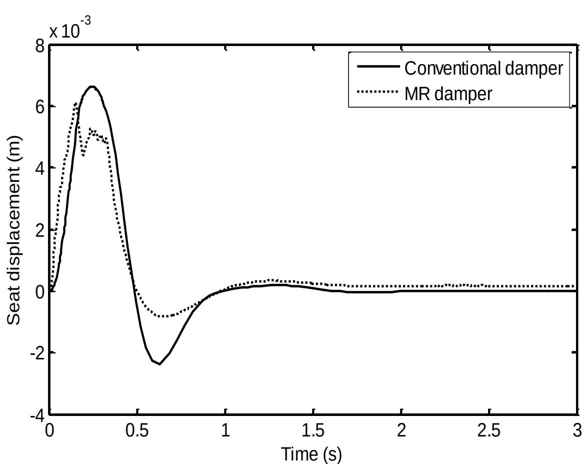

Fig.10 Time history of seat displacement for repeated trapezoid road, v=50 km/hr , E=2 V, s=1.7m.

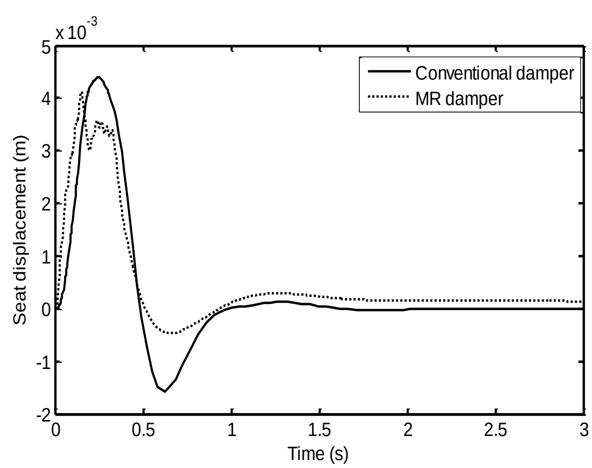

Fig.12: Time history of seat amplitude for repeated cosine road, v= 50 km/hr, E= 2 V, s= 1.7 m.

Figures 14, 15, 16, 17 and 18 show the time history of seat acceleration along one second using (MR) or conventional damper for car suspension when the car is assumed to move on different types of road with a velocity of 50 km/hr.

It is shown from this figure that, road type is a main factor for changing the amount of seat acceleration and the unsteady time during the car vibrations. Also it is clear from these figures that, cosine road type gives a smallest accelerations comparing to the other roads.
This means making a bumped road its shape as repeated cosine type is better for the car driver because the transmitted acceleration for the seat is small.

Also, it is noticed that, using conventional damper achieves the acceleration to be less than MR damper during unsteady period time, and in addition the unsteady time is the same for both conventional and MR damper.

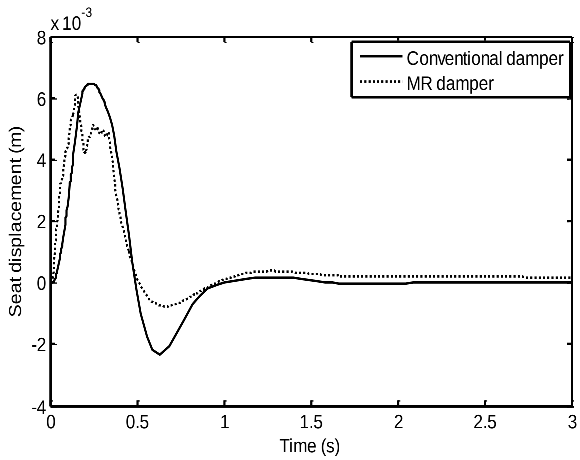

Fig.13 Time history of seat displacement for repeated simple harmonic road, v= 50 km/hr, E= 2 V, s= 1.7 m.

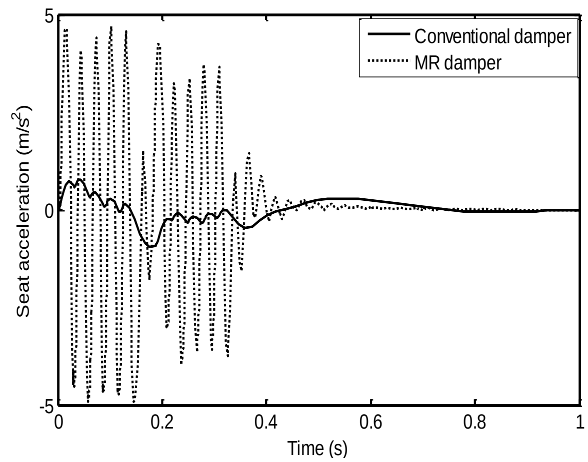

Fig.15 Time history of seat acceleration for repeated trapezoid bumped road, v = 50 km/hr, E=2V, s= 1.7 m.

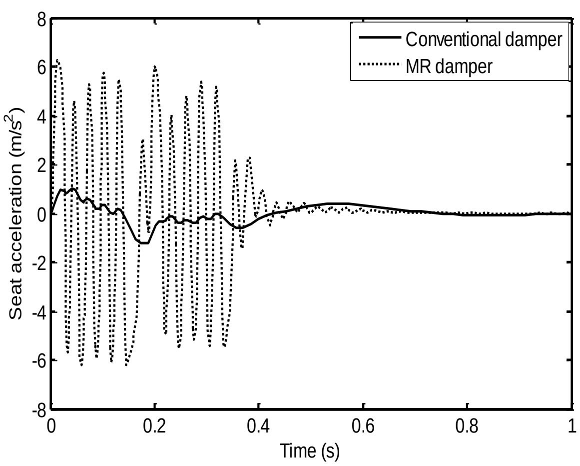

Fig. 14 Time history of seat acceleration for repeated pulses road=50 km/hr, E=2 V, s=1.7m.

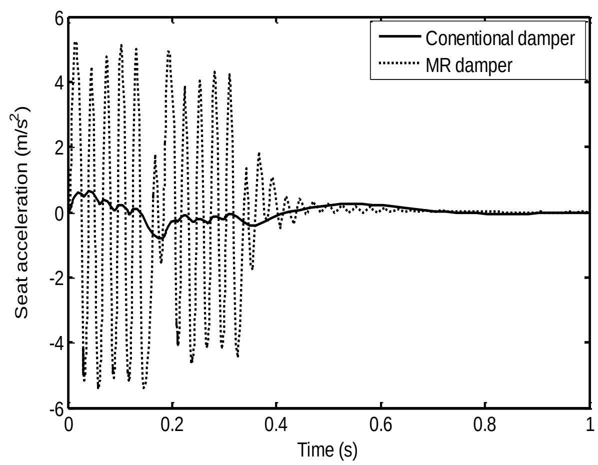

Fig.16 Time history of seat acceleration for repeated half sine road, v = 50 km/hr, E=2 Vs= 1.7 m.

The effect of input voltage to MR damper on the displacement of driver seat is clear from Fig.19 .Three values of voltage are taken into consideration 0, 2 and 4 volt.

This figure illustrates that, as the voltage increases the displacement decreases especially after beginning of motion with time of 0.1s. Also Fig. 20 shows the time history of seat acceleration for different values of voltage. It is found from this relation that, zero volts gives smallest acceleration but as the voltage increases from 2 to 4 volt the acceleration decreases from maximum value of 6 m/s^2 to be 4.5 m/s^2.

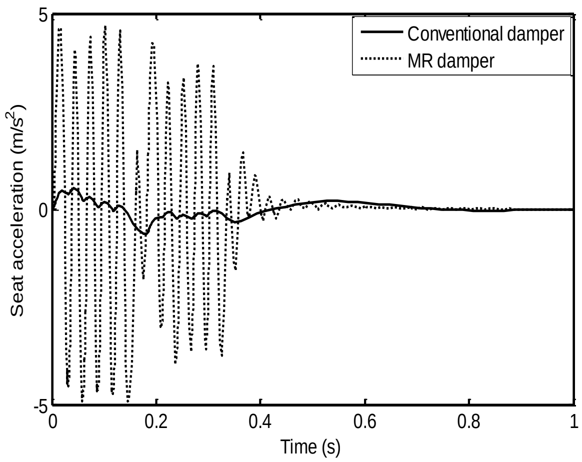

Fig.17 Time history of seat acceleration for repeated cosine road, v= 50 km/hr, E=2V, s= 1.7 m

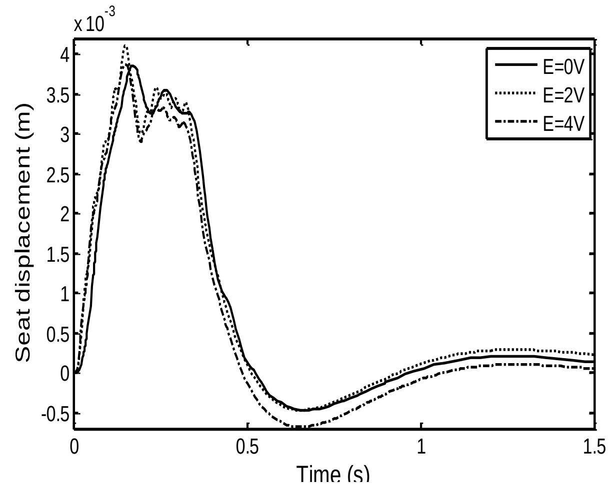

Fig.19 Effect of voltage on time history for seat displacement for Repeated cosine bumped road, v= 50 km/hr.

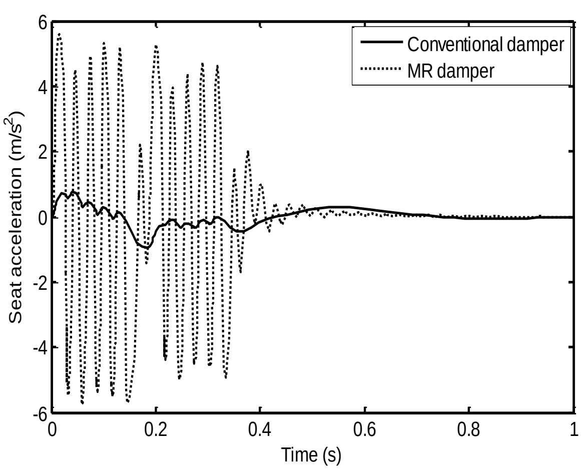

Fig.18 Time history of seat acceleration for repeated simple harmonic road, v= 50 km/hr, E=2 V, s= 1.7 m

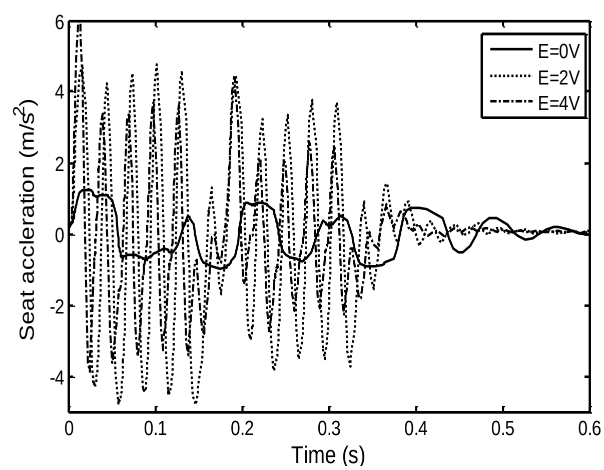

Fig.20 Voltage Effect on time history of seat acceleration, repeated cosine bumped road, v= 50 km/hr.

In addition to the mentioned comparison it is clear that, high voltage realizes small unsteady period of time. The value of pitch angle for seat changes according to the road type.The car is subjected to pitching motion when it moves sunddly on bumped road.

Figure 21 shows the time history of pitch angle for different roads .Four roads are chosen for comparison.

The car seat is assumed to be suspended by MR damper and its input voltage is 4 volt and the car velocity is 50 km/hr. It is clear from this figure that, cosine road gives the smallest value of pitching angle by comparison to other roads, so it is important for the engineer to select and design the shape of bumped road to achieves less trouble for car driver during the motion on these type of roads. Also rolling motion of car seat during the vibrations depends on the nature of bumped road .The above four roads are taken into account. Figure 22 shows the roll angle for different type of roads.

The car seat is assumed to be suspended by MR damper and its input voltage is 4 volt .This figure gives the same previous results which shows that, repeated cosine bumped roads realizes smallest displacement, pitch angle, and roll angle as compared to other roads, so practically the engineer builds this road to achieve less trouble for car driver when the car moves on this type of bumped road. To make analysis for the signal obtained from the vibration of the car a Fourier series is used to convert these signals to frequency domain (the relation between frequency and amplitude).

Figures 23 and 24 show Fast Fourier Transformation (FFT) for heave response when the velocity of car is 50 km/hr and the seat is suspended to conventional damper or MR damper.

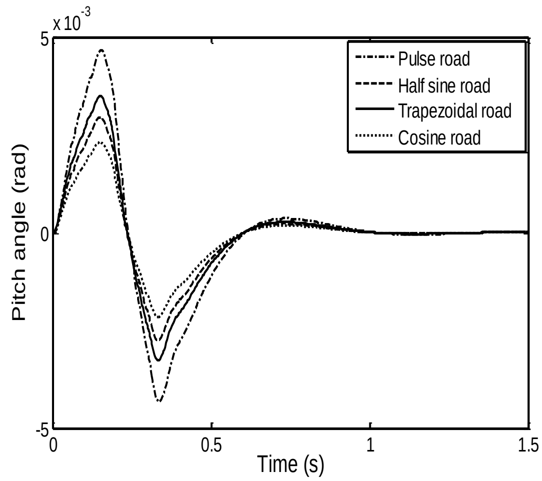

Fig.21 Effect of road type on pitch angle for vehicle, E=4V, v= 50 km/hr.

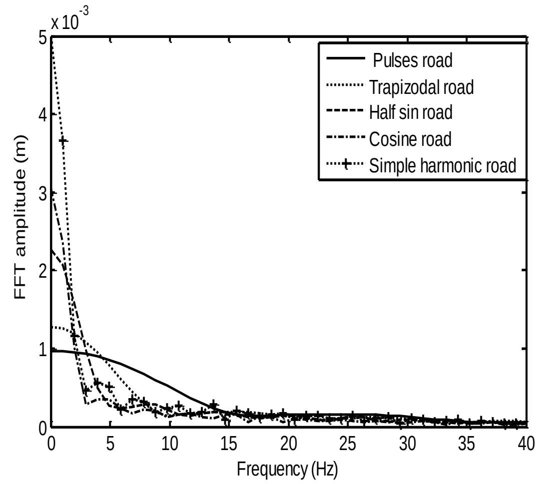

Fig.23 Fast Fourier transformation for seat displacement, conventional damper, v =50 km/hr

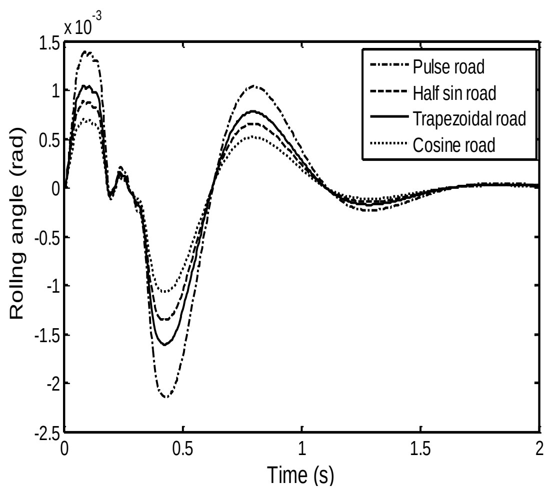

Fig.22 Effect of road type on roll angle for vehicle, E=4V, v= 50 km/hr.

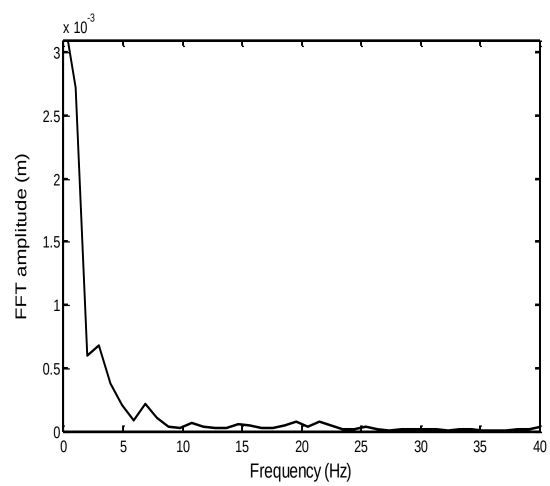

Fig.24 Fast Fourier transformation for seat displacement, MR damper E=2V, v=50 km/hr

Also in this case four roads are taken into consideration and its disturbance height and bumped length are 0.05 m and 1.7 m. It is shown from this figure that, the road type is a main factor for changing the maximum heave amplitude of vibration, since the amplitude of vibrations is high at low frequency (velocity) and low at high frequency. It should be noted that the change of car velocity means the variation in the forced frequency. So increasing the frequency achieves small vibration amplitudes.

The different types of roads are tested in this work (repeated pulses road, repeated trapezoid bumped road, repeated half sine road, repeated cosine road and repeated simple harmonic road. These roads can be classified into (five classes) which are numbered from (1→5).

The relation between road class and maximum vibration amplitude is shown by Fig. 25. This figure shows how the road type affects the value of heave amplitude of vibration for a car its velocity is 50 km/hr, and is suspended by classical damper or MR damper.

It clear from this relation that, repeated cosine road (class 4) gives minimum value of vibration amplitude. Also as illustrated before using MR damper for car suspension realizes more comfort for the driver because sudden vibration amplitude is smaller than that for classical damper.

The seat acceleration for different types of road as the car moves with a velocity of (50) km/hr, is illustrated in Fig. 26. It is clear from this figure that, seat acceleration has a smallest value when the vehicle moves on a repeated cosine road (class 4). Also this figure shows that, using conventional damper gives acceleration for the driver less than that for MR damper.

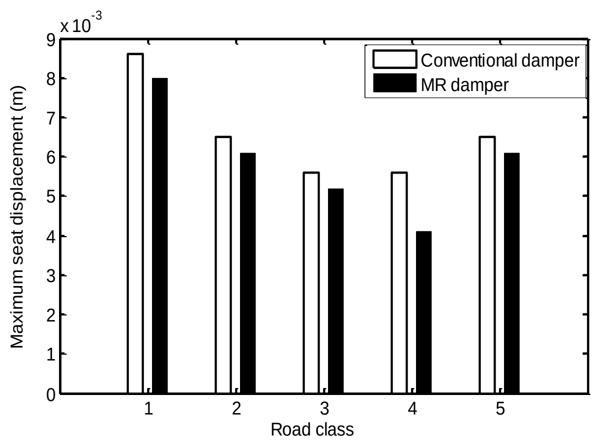

Fig.25 Road class vs. maximum amplitude v= 50 km/hr.

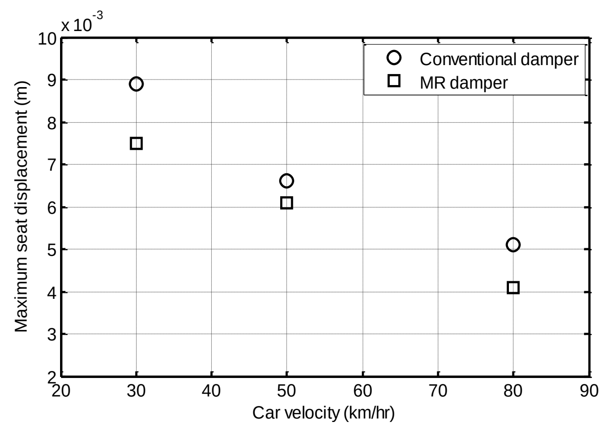

Fig.27 Velocity class vs. maximum amplitude on repeated trapezoid road , v=50 km/hr

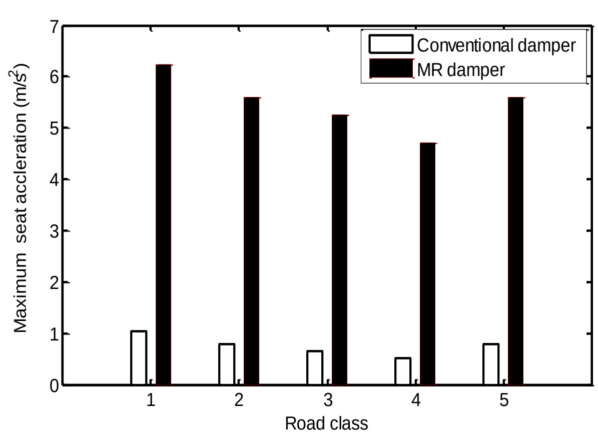

Fig.26 Road class vs. maximum acceleration v= 50 km/hr.

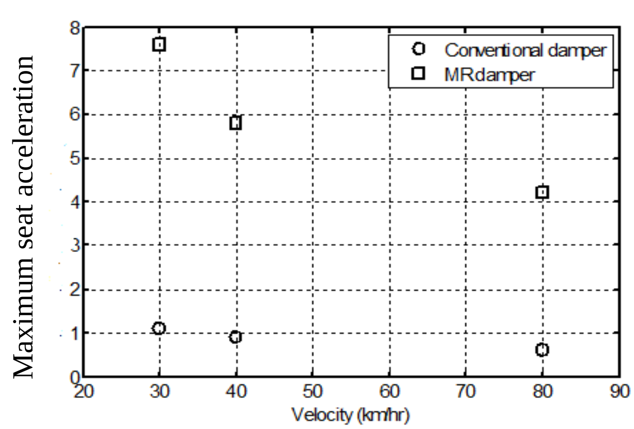

Fig.28 Velocity class vs. maximum acceleration repeated trapezoid road, E=2V, v= 50 km/hr.

Figure 27 shows the effect of car velocity on the vibration amplitude for car seat .Three different velocities (30, 50, and 80) km/hr, are taken into consideration. It is clear from this figure that, the amplitude of vibrations is high at low velocity and low at high velocities. The seat acceleration for different velocities (30, 50, and 80) km/hr, as the car moves on (repeated trapezoid bumped) road is shown in Fig.28.

As mentioned before the acceleration amplitude decreases as the velocity of vehicle increases .It is clear from this figure that, the seat acceleration reaches to the driver of the car is high at low velocity and low at high velocities. Sometimes the car is subjected to a sudden road variation .Many shapes of these roads can be studied, but in this work, three different roads were considered as shown in Figs. 29, 30 and 31 .The shapes of these roads are (single half sine, ramp, and sudden step type).

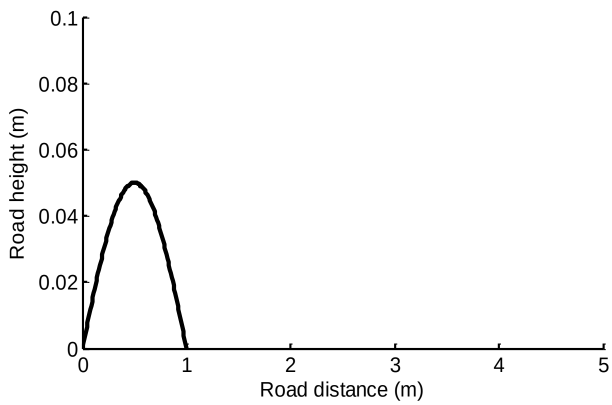

Fig.29 Half sine road

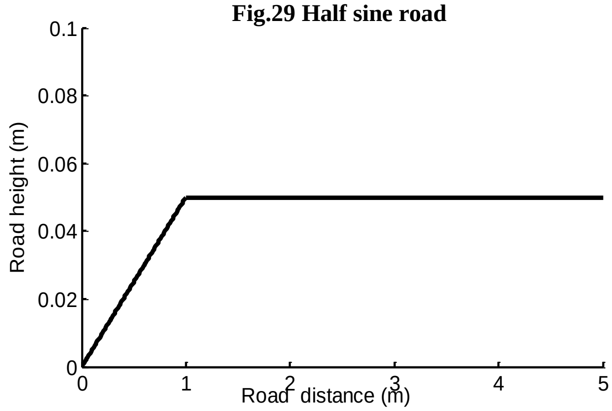

Fig.30 Ramp road

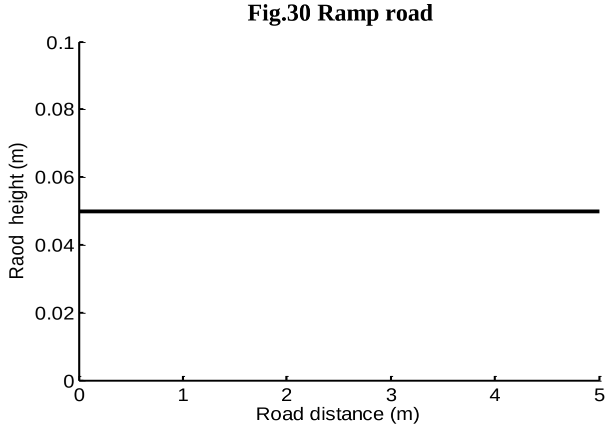

Fig.31 Step road

Figures 32, 33, and 34 show the time history of heave displacement along three seconds when the car moves on different types of roads. The seat is suspended by conventional or MR damper.

As mentioned before, three types of roads are taken into consideration (single half sine, ramp road, and step type). It is shown from these figures that, the unsteady vibrations for the car seat appears during a period of time its value changes according to the road type. Also it is found from these figures that, half sine road gives smallest amplitude of vibrations comparing to the other types of roads. Figs. 35, 36, and 37 shows the change in time history of seat acceleration for previous three roads.

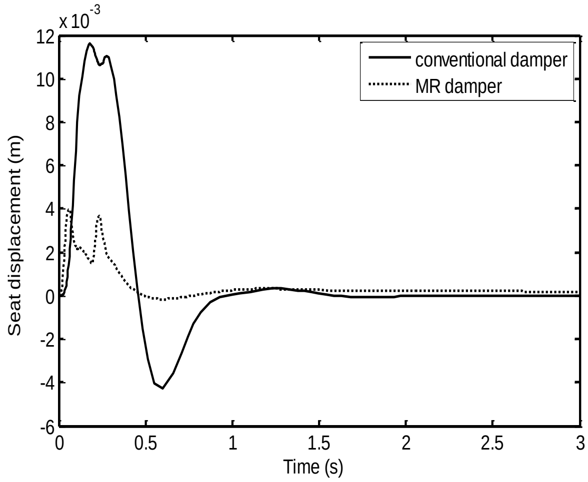

Fig. 32 Time history of seat displacement for half sine road v=50 km/hr, E= 2 V, s = 1m

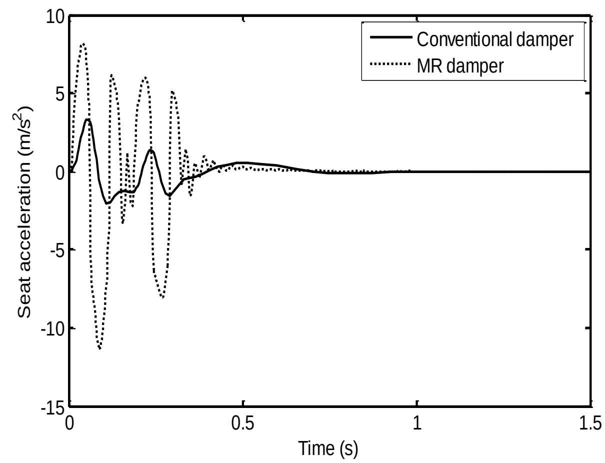

Fig. 35 Time history of seat acceleration for half sine road v=50 km/hr, E= 2V, s=1m

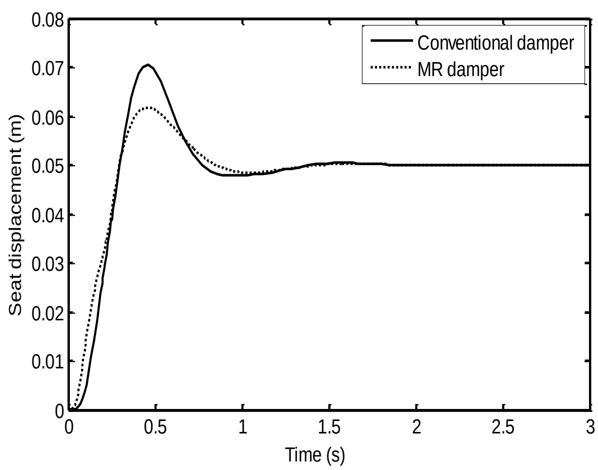

Fig.33 Time history of seat displacement for ramp road v=50 km/hr, E= 2V, s=1 m

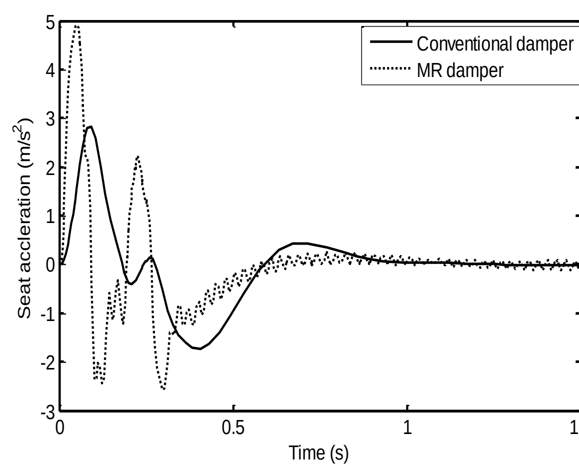

Fig.36 Time history of seat acceleration for ramp road v=50 km/hr, E= 2V, s= 1m

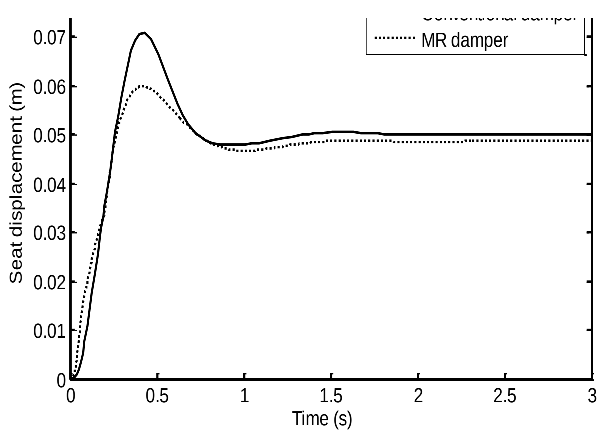

Fig.34 Time history of seat displacement for step road v=50 km/hr, E= 2V

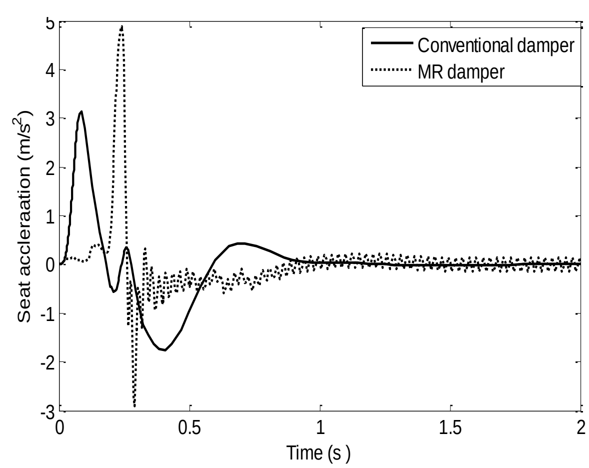

Fig. 37 Time history of seat acceleration for step road, v=50 km/hr, E=2V

5. Conclusions

According to this work, the following conclusions are summarized

* Magneto-Rheological (MR) damper is more efficient to reduce amplitude of vibrations comparing with, that for classical type.

*In general, the use of Magneto- Rheological (MR) dampers increases the performance of vehicle suspension and reduces the effect of road irregularities on passengers.

* MR damper is more realistic and achieves comfort for vehicle driver.

* MR damper makes the unsteady state time during vibrations less than, that for conventional type.

* The amplitude of vibrations decreases as the input voltage to MR damper increases.

*The amplitude of vibrations decreases as the velocity of vehicle increases for all types of roads if MR damper or conventional damper is used.

* Repeated cosine road gives minimum amplitude of vibrations for both conventional and (MR) damper.

* Single half sine road gives minimum amplitude of vibrations for both conventional and (MR) damper.

* Using conventional damper realizes small acceleration of vibrations comparing with, that for (MR) damper.

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