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Numerical analysis using state space method for vibration control of car seat by employing passive and semi active dampers

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Abstract

In passenger cars the vibrations developed at the ground are transmitted to the passengers through seats. Due to vibrations discomfort is experienced by the passengers. Dampers are being successfully utilized to reduce the vibrations in civil engineering structures. Few dampers are used in passenger cars as well. In the present study viscous dampers and semi-active variable friction dampers are used in passenger cars. The car and car seat are modeled together as two degrees of freedom system. The study is carried out for two cases namely car moving on sagged bridges and car moving on road humps. The paper also presents the comparison of performance of both the dampers for the two cases. State space method has been employed for the numerical analysis of the study. It is found that the amplitude of displacements is reduced considerably by the employment of dampers and resonance is avoided. Performance wise viscous damper is found to be slightly better than semi-active variable friction damper.

Keywords: Car seat, viscous damper, semi-active variable friction damper, two degrees of freedom, state space method.

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1. Introduction

Ride quality is an important aspect in determining the performance of passenger cars. Due to unavoidable humps (rumble strips) the discomfort is experienced by passengers. The discomfort is also experienced due to deflection (sagging) of roadways over the bridges. Various types of dampers are being successfully utilized to reduce the vibrations in civil engineering due to natural dynamic loads such as wind and earthquake loads. Ahmadizadeh (2007) has studied the capabilities of passive control of structures using hydraulic fluid dampers as compared to those of semi-active control. Sinha (2004) has studied the performance of fluid viscous damper in controlling the vibrations of steel moment resisting frame in reinforced cement concrete buildings. Large-size energy dissipation dampers have also been developed and applied to various structures for vibration control and have achieved successful results (Spencer and Soong, 1999). Suspension is the equipment that serves the basic function of isolating passengers and chassis from the vibrations due to roughness of the road (G*ü*cl*ü*, 2001). Unsal *et al*. (2004) have mentioned that, Ferri and Heck where the first to come up with the idea of varying the normal force in a frictional joint to enhance energy dissipation from a vibrating structure. Dampers can be used in passenger cars as well. Piezoelectric friction damper and magnetorheological damper were also used for vibration isolation of a single degree of freedom system.

A methodology for optimizing the suspension parameters for quarter car models was presented by Verros *et al.* (2005). Rao *et al.* (2010) have analyzed the passive and active controlled suspension systems in an omnibus passing over a speed bump. A low frequency analysis on passenger car seats was made by Joseph *et al.* (2013) and they concluded that the transmissibility will vary with terrain. Subject weight does not have a strong impact on seat transmissibility. The principal frequency in the seat transmissibility reduces with higher vibration magnitudes was reported by Toward and Griffin (2010). Ashish *et al.* (2014) have studied the variations in the car driver's seat suspension system with air bellow and viscous damper. Bhiwapurkar *et al.* (2010) have studied sketching activity in a mock-up of passenger's compartment. The authors investigated the performance measure for sketching activity in multi axis random vibrations using subjective evaluation and two percentage distortion methods. The sensitive parameters influencing vertical ride are discussed using parametric analysis (Sharma,2011).

Thus employment of dampers can reduce the vibrations to certain extent. There are four kinds of dampers namely active, passive, semi-active and hybrid dampers. Active dampers are the ones that make use of sensors, continuous monitoring systems and actuators. Active dampers are meant to provide additional energy to the controlled structure. They compensate to that energy delivered by the dynamic loading. Passive dampers are devices which develop forces at the location of the device itself by utilizing the movement of the structure. A Semi-active damper is a combination of passive and active dampers. In semi-active dampers the properties of passive dampers are modified according to a control algorithm. Hybrid damper is a combined system of passive or active or semi-active dampers rolled into one (Spencer Jr. and Soong, 1999).

Employment of active dampers have advantages such as wider range of force, no force-velocity constraint and can achieve better performance (vehicle dynamics). But it has certain disadvantages such as high power consumption, higher weight to power ratio, high cost and major modifications should be made before installing active system into the existing vehicle. So, passive and semi-active dampers are preferred over the active ones for the sake of simplicity and their advantages. Passive dampers have advantages such as low cost, no external energy consumption and are inherently stable. Semi-active dampers have advantages such as low implementation cost, low power consumption, ease of control, simple design and ease of installation (Housner *et al.*, 1997; Fujino *et al.*, 1996). Senthilkumar *et al.* (2011) have investigated the efficiency of particle damping (passive damping) in control of vibrations in a boring bar using copper and lead particles of various sizes. It was found that considerable reduction in vibration was achieved with the particle damping.

Due to the above mentioned reasons, an attempt has been made to study the performance of viscous damper and semi-active variable friction damper on the car seat. Fluid viscous dampers consist of a cylinder which encloses a piston head with orifices, and is filled with a highly viscous fluid which is usually a compound of silicone or similar oil. When piston head moves through the fluid, energy is dissipated in the damper by fluid orificing. When the damper is subjected to compressive force, the volume of nearly incompressible fluid inside the cylinder decreases. This results in a restoring force and is undesirable. It is usually prevented by using a run-through rod that enters the damper. It is connected to the piston head, and then passes out the other end of the damper. Thus viscous fluid damping, which is a passive damping method utilized in this study. It works by transferring the system's kinetic energy into heat through fluid orificing (Housner *et al.*, 1997). In mechanical engineering, dry friction has long been used for reducing the motion of moving or rotating objects because it is an effective as well as economic method. Friction dampers are preferred over other types of energy dissipating techniques because of the advantages such as: materials are less affected by the degradation due to aging; materials are insensitive to the change in ambient temperature; and no material yielding and replacement problems; and no fluid leaking problems. Predictive control algorithm is used for the semi-active variable friction damper (Lu, 2004). Predictive control algorithm is capable of keeping the friction force of a semi-active friction damper slightly lower than the critical friction force. So the damper is kept in continuously slipping to avoid the unwanted high-frequency structural response, which would have been produced in passive friction dampers (Patil and Jangid, 2009).

Due to requirements of complex tasks and good accuracy, the modern trend in engineering systems is towards complexity. Complex systems may have multiple inputs and outputs and may be time varying. In order to meet the stringent requirements on the performance of control systems, the increase in system complexity and availability of large scale computers modern control theory is developed since around 1960. This new approach is based on the concept of state, which has been in existence for a long time in the field of classical dynamics and other fields. Thus the state space method which is a modern control theory can deal with multiple input and output systems as well as time variant systems (Ogata, 2010). The state space method has been employed for numerical simulation of seismic structures (Lu and Chung, 2001). For the modal control of the structures, the numerical formulation is done using the state space method (Lu, 2003). Chen and Lee (2004) have investigated the bending and free vibration problems of simply supported angle-ply laminates with interfacial damage in cylindrical bending by employing the state-space method.

As per the best knowledge of authors, very few attempts of using viscous damping for seat isolation are made in cars. In the paper, an attempt is made to study the performance of the car moving on sinusoidal road profile (like sagged bridges and road humps) by employing state space method for the numerical analysis. Predictive control theory used in civil structures for earthquake control has been employed in the isolation of car seat during the semi-active control. The objectives of the study are,

- To study the performance of viscous dampers and semi-active variable friction dampers when employed between the chassis and car seat.
- To carry out a parametric study by varying the velocity of car moving over a sagged bridge and hump.
- To carry out a parametric study by varying the amplitude in case of sagged bridges and humps.
- To perform a comparative study of viscous damper and semi-active variable friction dampers.

2. Modelling

2.1. Model of Car

Car and car seat are modeled as two degrees of freedom (TDOF) system with inherent damping and stiffness and is shown in Figure 1. An additional damper for vibration control is employed between car seat and chassis. The mass of the car (m_1) considered for the study is 1396 kg and the considered inherent stiffness of the car (k_1) for the study is 827248 N/m (Agharkakli *et al.,* 2012). The mass of the seat (m_2) considered for the study is 80 kg including the passenger (Singh *et al.,* 2013). The considered seat stiffness (k_2) for the study is 20800 N/m (Prasad, 2010). The inherent damping ratio of car considering the tire and suspension is 40% (Chopra, 2005). The inherent damping of seat is 3.49% (Ashish *et al.,* 2014).

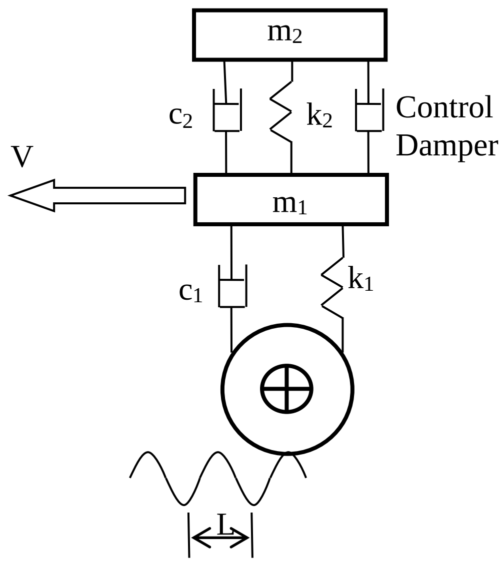

Figure 1: TDOF model of car and car seat

2.2. Governing Equation of Motion

The dampers are installed below the seat as shown in Figure 1. In general, equations of motion of the TDOF model of the car installed with dampers can be written as,

$$M\ddot{x} + C\dot{x} + Kx + \Lambda u = F \tag{1}$$

where, M is the mass matrix, K, stiffness matrix, and C is damping matrix each of the order of (2×2) are constructed for the TDOF model of the car and car seat; x is the system displacement vector; $\dot{x}$ and $\ddot{x}$ are the system velocity and acceleration vectors, respectively; $u=[f_{d1}]$ is the vector of damper forces and $F=[f_1, f_2]^T$ is the exciting force vector; Λ is a matrix of zeros and ones, in which 'one' will indicate where the damper force is being applied. The profile of the road considered is sinusoidal and is given by,

$$p = P\sin(\omega t) \tag{2}$$

where, P - amplitude of the sinusoidal road profile in meter; t -time in seconds; ω - forcing frequency in rad/sec,

$$\omega = \frac{2\pi L}{V} \tag{3}$$

L - wavelength of the sinusoidal road in m; and V - velocity of the car in m/s.

The equations of motion (1) can be written in the state-space form as,

$$\dot{z} = Az + Bu + EF \tag{4}$$

where, z is the state vector of structure, and contains displacement and velocity of each mass; A denotes the system matrix composed of structural mass, damping and stiffness matrices; B represents the distributing matrices of the control forces; and E represents the distributing matrices for wind load excitation.

The Equation (4) is discretized in the time domain and excitation force is assumed to be constant at any time within the interval and can be written into a discrete-time form (Lu, 2004).

$$z[k+1] = A_d z[k] + B_d u[k] + E_d F[k] \tag{5}$$

where, $A_d = e^{A\Delta t}$ represents the discrete time system matrix with Δt as the time interval and

$$B_d = A^{-1}(A_d - 1)B \tag{6}$$

$$E_d = A^{-1}(A_d - 1)E \tag{7}$$

$$B = \left[\frac{0}{M^{-1}\Lambda}\right];\ E = \left[\frac{0}{M^{-1}}\right] \tag{8}$$

$$A = \left[\frac{0 \qquad\qquad I}{-M^{-1}K \qquad -M^{-1}C}\right] \tag{9}$$

For the validation of the above mentioned state space method, a numerical analysis on a single degree of freedom (SDOF) system has been carried out. The response of the system is plotted using the state space method (numerical) as well as the analytical method. The analytical transmissibility equation is given by,

$$\boldsymbol{T_d} = \frac{\boldsymbol{X}}{\boldsymbol{P}} = \sqrt{\frac{\mathbf{1}+(\mathbf{2}\zeta r)^{\mathbf{2}}}{(\mathbf{1}-r^{\mathbf{2}})^{\mathbf{2}}+(\mathbf{2}\zeta r)^{\mathbf{2}}}} \tag{10}$$

where,

X - displacement of car P- displacement of ground car

ζ - Damping ratio $r = \omega/\omega_n$, where ω - forcing frequency and ω_n – natural frequency of the system (Rao, 2011)

The time history of the displacement of the car moving over a sinusoidal road profile is presented in Figure 2. From the figure it is found that there is reasonable match between state space and analytical methods.

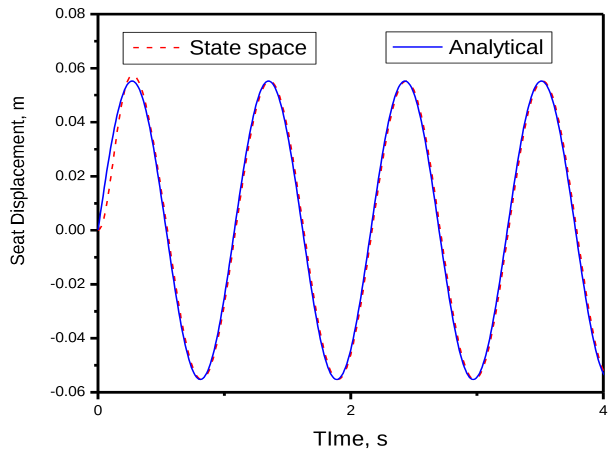

Figure 2: Displacement of the car moving over a sinusoidal road profile

2.3. Viscous Damper

In the work of Patil and Jangid, (2009) it is clearly mentioned that, viscous damper consists of viscous material in the form of either liquid (silicone gel) or solid (special rubbers or acrylics). There are two categories of viscous dampers based on their functioning; they are the ones in which (a) energy dissipation is achieved through the deformation of viscous fluid or special solid material (i.e. through fluid viscosity) and (b) energy dissipation is achieved by the principle of flow through orifice. In a fluid viscous damper, the difference of the pressure on each side of the piston head results in the damping force, and the damping constant of the damper which can be determined by adjusting the configuration of the orifice of the piston head. When it comes to pure viscous behavior, damper force and the velocity should remain in phase. However, for a damper setup shown in Figure 3(a), the volume for storing the fluid will change while the piston begins to move. Thus a restoring force, which is in phase with displacement rather the velocity, will be developed. Configuration of an accumulator is used to solve the problem.

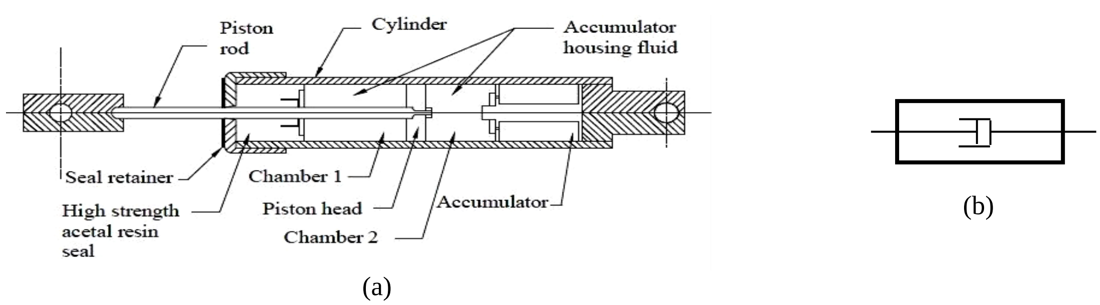

Figure 3: (a) Construction and (b) Mathematical Model of viscous damper (Symans and Constantinou, 1998)

The ideal force out for a viscous damper is given by,

$$f_{di} = C_{md}|\dot{x}_{i2} - \dot{x}_{i1}|^{\varepsilon} sgn(\dot{x}_{i2} - \dot{x}_{i1}) \tag{11}$$

Where, C_{md} is coefficient of damper, $(\dot{x}_{i2} - \dot{x}_{i1})$ is relative velocity between the ends of i^{th} damper and ε is exponent having value between 0 and 1. The damper with $\varepsilon = 1$ is called a linear viscous damper (LVD). The damper with ε larger than 1 have not been seen often in practical applications. The damper with ε smaller than 1 is called a nonlinear viscous damper which is effective in minimizing high velocity shocks.

2.4. Semi-active Friction Damper

A typical friction damper usually consists of a frictional sliding interface and a clamping mechanism that produces normal contact force on the interface. For a passive friction damper, the clamping force of the damper is a pre-determined fixed value, as is the slip force applied on the frictional interface. A passive damper will be activated and start to slip once the externally induced damper force exceeds the pre-determined slip load. A friction damper is able to dissipate energy only if the damper is in its slip state; otherwise, the damper is no different from a regular bracing member.

In order to improve the performance of friction dampers, the concept of semi-active control is introduced in the dampers. Construction and Mathematical Model of semi active variable friction damper is shown in Figure 4. A semi-active friction damper is able to adjust its slip force by controlling its clamping force in real-time in response to car's motion during the road disturbance. Because of this adaptive nature, a semi-active friction damper is expected to be more effective than a passive damper. On the other hand, just like in an active structural control, the control of semi-active friction dampers requires a feedback control algorithm and online measurement of structural response in order to determine the appropriate level of adjustable clamping forces of the dampers. Nevertheless, a semi-active control device generally has the advantages over an active control viz., (i) because the control action is carried out by adjusting the internal mechanism (i.e., the clamping force for a semi-active friction damper), the required control stroke and energy can be very small; and (ii) because it does not pump energy into the controlled structure, control instability can be prevented. As already mentioned, the control of semi-active dampers requires a control algorithm. Needless to say, the control performance of the semi-active dampers significantly relies on the control algorithm applied.

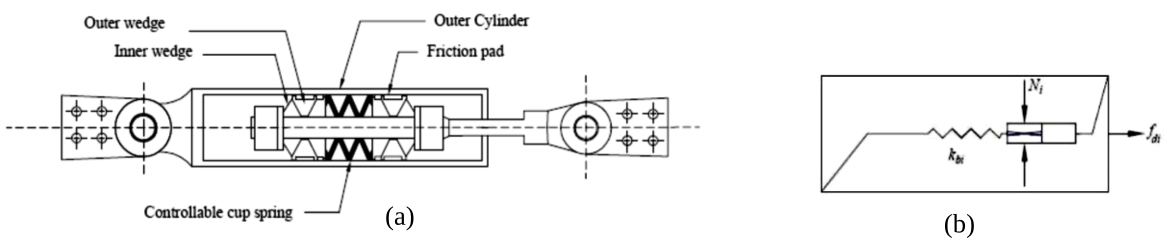

Figure 4:(a) Construction and (b) Mathematical Model of semi active variable friction damper (Symans and Constantinou, 1998)

One of the recent control laws "The predictive control law" (Lu, 2004) which determines the vector of critical friction forces of dampers for the next, time step is given by;

$$\tilde{u}[k] = G_z z[k-1] + G_u u[k-1] + G_w F[k-1] \tag{12}$$

The control force vector actually maintained so that all the dampers will be in slip state is given by;

$$u[\mathrm{k}] = R_f \tilde{u}[k] \tag{13}$$

where, R_f is a gain multiplier defined as the ratio of damper force to critical damper control force and plays an important role in the present control law and,

$$G_z = K_b D(A_d - I) \tag{14}$$

$$G_u = K_b D B_d + I \tag{15}$$

$$G_w = K_b D E_d \tag{16}$$

Let y be a vector listing all damper elongations (deformations) that are equal to the drifts of the masses between which the dampers are installed. At any given point of time, the relation between y and the state of the system, z may be written as,

$$y[k] = Dz[k] \tag{17}$$

It can be observed that when the damper is connected between the car and car seat the matrix D will be modified so as to give the drifts of the alternative masses which would be generally more than those connected between the successive masses. This modification of D would modify all the three terms G_z, G_u and G_w to give enhanced friction force which would result in better energy dissipation.

3. Numerical Study

3.1. Viscous Damper

Initially the variation of the displacement quantities of the car seat by employing LVD is studied. The parameters such as car velocity and amplitude of the road profile are varied. The study is carried out for two cases. In the first case, the performance of the damper is studied when the car moves over sagged bridges. In the second case the performance is studied when the car moves over road humps. The natural frequencies of the first and second mode of the system are calculated from the parameters considered in section 2.1 and are 15.785 rad/s and 24.865 rad/s respectively.

a) Car Moving on Sagged Bridges:

For studying the effect of car velocity on the seat displacement quantity the car velocity is varied from 10 to 100 kmph with an interval of 10 kmph. The wavelength of the sagged bridge considered in the present study is 6 m. The amplitude of sagged bridge is 0.05 m. Beyond the co-efficient of viscous damper of 3400 Ns/m the damper is becoming unstable. Hence the co-efficient of viscous damper considered is 2000 Ns/m to be well within the range. Using equation (5), the displacement quantities of the car seat are obtained using MATLAB programs and the amplitude of the sinusoidal vibrations is taken into consideration. The same procedure is followed in all the remaining cases of the numerical study. The variation of peak displacement quantity of car seat with the car velocity is presented in Figure 5.

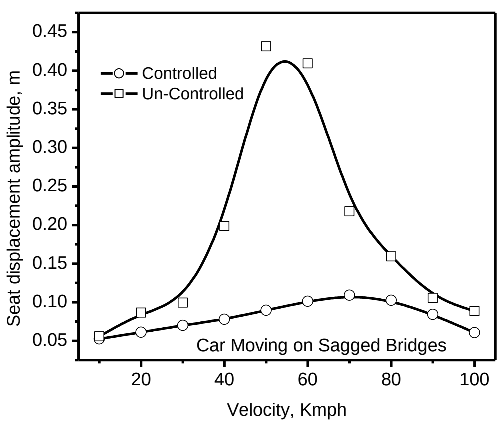

Figure 5: Variation of peak seat displacement with car velocity when employing LVD

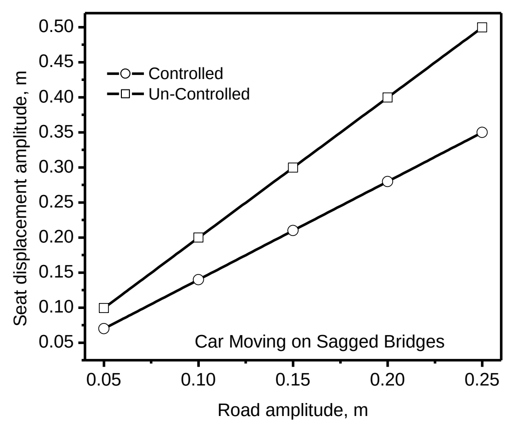

Figure 6: Variation of peak seat displacement with road amplitude when employing LVD

From the figure, the uncontrolled seat displacement amplitudes attain a maximum value at the car velocity of 54.264 kmph. This is because at that velocity the forcing frequency becomes equal to the natural frequency of first mode of vibration of the system which is 15.785 rad/s. The transmissibility of the seat increases around this frequency. It can be seen that, when LVD is employed the amplitude of displacements reduces considerably. They follow an increasing pattern till 60 kmph. They reach a maximum at the car velocity of 70 kmph, where the forcing frequency becomes equal to the natural frequency of second mode of vibration which is 24.865 rad/s. It can be inferred that the car velocity between 65 and 75kmph should be avoided in order to experience less disturbance when car seat is employed with LVD.

In order to study the effect of road amplitudes on the seat displacement quantity, the road amplitude is varied from 0.05 to 0.25m with an interval of 0.05 m. From the Figure 6, the velocity of car is taken as 30 kmph to avoid large amplitudes. The co-efficient of LVD considered is 2000 Ns/m. The variation of peak seat displacement quantity of the car seat with amplitude of the road profile is shown in Figure 6. From the figure, it can be seen that the amplitude of displacement of the car seat follows a linearly increasing pattern against the variation in road amplitude. The controlled seat displacement amplitudes are lesser than that of the uncontrolled system. As the amplitude of the road profile increases, the performance of the damper is observed to be more effective.

b) Car Moving over humps:

In order to study the effect of car velocity on the amplitude of seat displacement, the car velocity is varied from 10 to 100 kmph with an interval of 10 kmph. The wavelength and amplitude of the sinusoidal humps in the present study are considered as 0.2 m. The co-efficient of LVD considered is 2000 Ns/m. Since the wavelength of the road disturbance is small the displacement will be maximum when the car hits the first hump. The variation of peak displacement quantity of car seat with the car velocity is presented in Figure7.

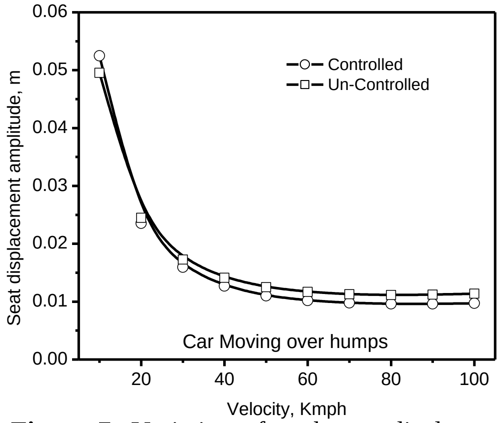

Figure 7: Variation of peak seat displacement with car velocity when employing LVD

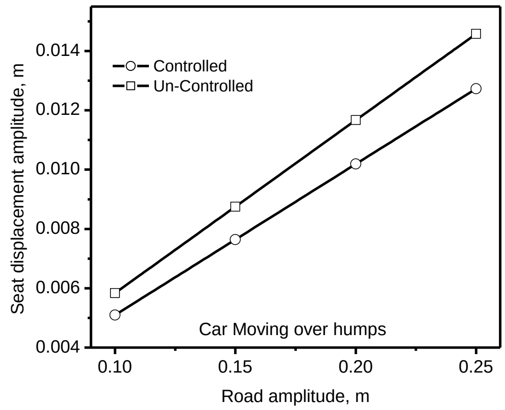

Figure 8: Variation of peak seat displacement with road amplitude when employing LVD

Figure 7 shows that, the amplitude is maximum at speed of 10 kmph and goes on reducing with increase on the car velocity for both controlled and uncontrolled. At 10 kmph the controlled seat amplitude is more than the uncontrolled one. The reason may be the lesser relative velocity between car and car seat. As car velocity is increased the relative velocity increases between chassis and seat. As the value of r [$r = \omega/\omega_n$] is quite high due to smaller wavelength, increasing the damping will lead to higher amplitudes at lower velocities. Raise in the relative velocity increase the magnitude of the damper force. The transmissibility of the seat varies with the terrain and similar conclusions were dawn by Joseph *et al.* (2013).

Further the study is carried out to find the effect of road amplitudes on the seat displacement the road amplitude is varied from 0.1 to 0.25 m with an interval of 0.05 m. From the Figure 7 the velocity of car is taken 60 kmph so as to avoid large amplitudes. The wavelength of the hump profile considered is 0.2 m. The co-efficient of LVD considered is 2000 Ns/m. The variation of peak displacement quantity of car seat with the amplitude of road profile is presented in Figure 8. From the figure, it can be seen that the amplitude of displacement of the car seat follows a linearly increasing pattern against the variation in road amplitude. The controlled seat displacement amplitudes are lesser than that of the uncontrolled system. As the amplitude of the road profile increases, the performance of the damper is observed to be more effective.

3.2. Semi-active Variable Friction Damper

The response of the car seat by employing predictive control theory for semi-active variable friction damper (SAVFD) is studied. The two cases when car moves on sagged bridges and humps are presented below. In the following studies the parameters are compared for vibration control with and without using SAVFD.

a) Car Moving on Sagged Bridges:

For studying the effect of car velocity on the seat displacement quantity the car velocity is varied from 10 to 100 kmph with an interval of 10 kmph. The wavelength of the sagged bridge considered is 6 m. The amplitude of sagged bridge is 0.05 m. The stiffness of the friction damper considered is five folds the actual stiffness of the car. The optimum value of five times is being taken after numerical analysis using MATLAB code. The variation of peak displacement quantity of car seat with the car velocity is presented in Figure 9.

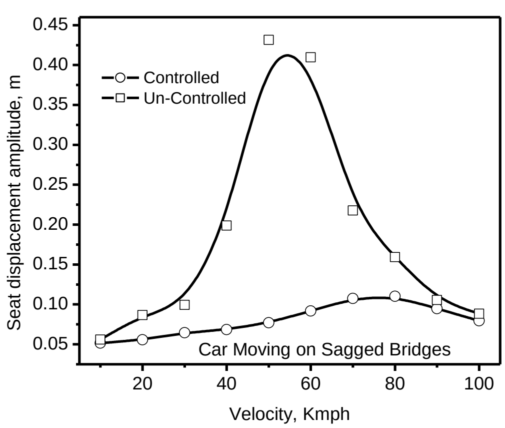

Figure 9: Variation of peak seat displacement with car velocity when employing SAVFD

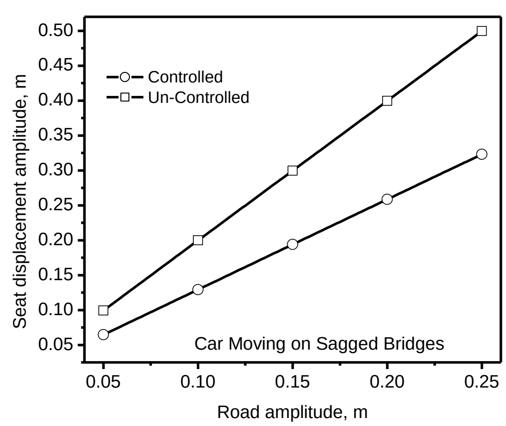

Figure 10: Variation of peak seat displacement with road amplitude when employing SAVFD

From the figure, the uncontrolled seat displacement amplitudes attain a maximum value at the car velocity of 54.264 kmph. This is because at that velocity the forcing frequency becomes equal to the natural frequency of first mode of vibration of the system which is 15.785 rad/s. The transmissibility of the seat increases around this frequency. It can be seen that, when SAVFD is employed the amplitude of displacements reduces considerably. They follow an increasing pattern until 60 kmph. They reach a maximum at car velocity of 80 kmph, where the forcing frequency becomes equal to the natural frequency of second mode of vibration (24.865 rad/s). It is clear from the figure that, the car velocity between 70 and 80 kmph should be avoided in order to minimize disturbance when car seat is employed with SAVFD.

For studying the effect of road amplitudes on the seat displacement quantity, the road amplitude is varied from 0.05 to 0.25 m with an interval of 0.05 m. The velocity of car is maintained as 30 kmph as it is clear from Figure 8 where the amplitude is quite small. The stiffness of the SAVFD considered is five folds of the stiffness of car. The variation of peak seat displacement quantity of the car seat with amplitude of the road profile is presented in Figure 10. From the figure it is clear that the amplitude of displacement of the car seat follows a linearly increasing pattern against the variation in road amplitude. The controlled seat displacement amplitudes are lesser than that of the uncontrolled system. As the amplitude of the road profile increases, the performance of the damper is observed to be more effective.

b) Car Moving over Humps:

For studying the effect of car velocity on the seat displacement quantity, the car velocity is varied from 10 to 100 kmph with an interval of 10 kmph. The wavelength and amplitude of the sinusoidal humps considered are 0.2 m. The stiffness of the SAVFD considered is five folds of the stiffness of car. As the wavelength of the road profile is small the displacement will be maximum when the car hits the first hump. The variation of peak displacement quantity of car seat with the car velocity is presented in Figure 11.

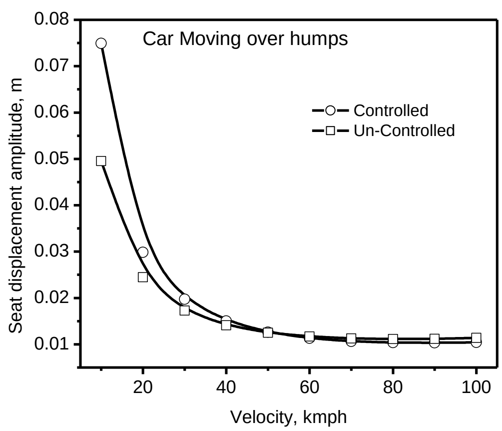

Figure 11: Variation of peak seat displacement with car velocity when employing SAVFD

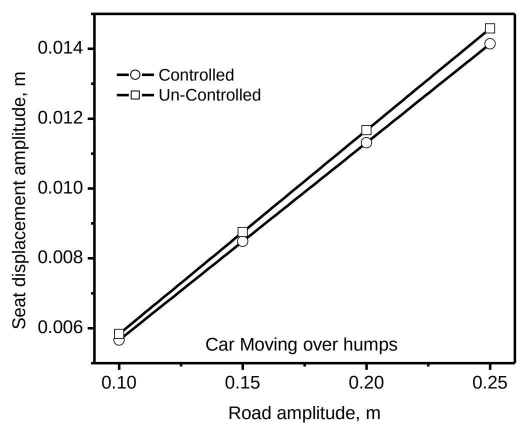

Figure 12: Variation of peak seat displacement with road amplitude when employing SAVFD

Figure 11 shows that the amplitude is maximum at speed of 10 kmph and goes on reducing with increase on the car velocity. In the range of car velocity 10 to 30 kmph the controlled amplitude is more than the uncontrolled amplitude. The reason is lesser relative displacement between car and car seat until car velocity of 30 kmph is reached. As the value of r is quite high due to smaller wavelength of the road profile, increasing the damping will lead to higher amplitudes at lower velocities. As car velocity is increased the relative displacement increases between chassis and seat. Rise in the relative displacement increases the magnitude of the damper force and thus, the controlled displacement amplitudes are slightly lesser than the uncontrolled ones at higher speeds. The transmissibility of the seat varies with the terrain and similar conclusions were dawn by Joseph *et al.* (2013).

For studying the effect of road amplitudes on the seat displacement quantity, the road amplitude is varied from 0.1 to 0.25 m with an interval of 0.05 m. From Figure 12, the velocity of car maintained as 60 kmph so as to avoid large amplitudes. The stiffness of the SAVFD considered is five folds of the stiffness of car. The variation of peak seat displacement quantity of the car seat with amplitude of the road profile is presented in Figure 12. From the figure, the amplitude of displacement of the car seat follows a linear pattern against the variation in road amplitude. The controlled displacement amplitudes are almost equal to that of the uncontrolled system. As the amplitude of the road profile the performance of the damper is observed to be more effective.

4. Comparative Study

The performance of both the dampers is studied by keeping the maximum damper force as 2210 N. The displacements of car seat are compared for both the dampers. The hysterisis curves for both the dampers are plotted and presented in Figure 13.

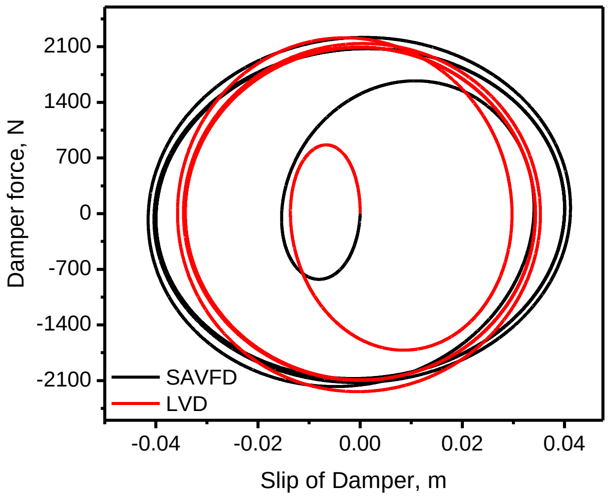

Figure 13: Hysterisis curves for SAVFD and LVD

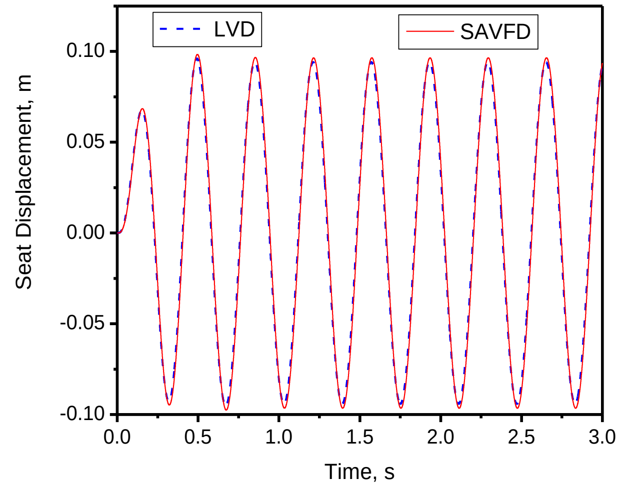

Figure 14: Comparison of seat displacements of SAVFD and LVD.

The maximum slip of the SAVFD is 0.0414 m whereas that of LVD is about 0.0356 m. The SAVFD thus shows relatively more displacement (slip) as compared to LVD for the same maximum damper force. The variation of displacement quantities with the two dampers are presented in Figure 14. It shows that, the LVD has slightly lesser amplitude than the viscous damper. The figure also reveals that the slip of the LVD is lesser than that of SAVFD. Thus it can be concluded that LVD is slightly better than the SAVFD in controlling the vibration amplitudes of car seat.

5. Conclusion

A passenger car is modeled as a TDOF system and numerical study of its vibration control is carried out. The variation of displacement quantities car seat is studied by employing LVD and SAVFD. Initially, the vibrations of a passenger car are studied in its uncontrolled state, when it moves over sagged bridges. The peak value of car seat amplitude is observed to be at the car velocity of 54.264 kmph. For the same case, in the controlled state the peak value is observed to be in the velocity range of 70-80 kmph, when LVD is employed. Whereas, the peak value of car seat amplitudes is in the range of 65-80 kmph, when SAVFD is employed. Both the dampers are quite effective in reducing the displacement quantities.

In the latter case, the vibrations of passenger car are studied while it moves over road humps. The effect of dampers is not found to be much significant in this case.

A parametric study of the variation of displacement quantities of the car seat with amplitude of road profile is also carried out. The seat amplitudes increase linearly with the amplitude of the road profile.

Nomenclature

SDOF	Single Degree of Freedom System
TDOF	Two Degrees of Freedom
LVD	Linear Viscous Damper
SAVFD	Semi Active Variable Friction Damper

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