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Passenger Comfort Analysis in an Automotive Considering a Magneto-Rheological Damper based Suspension

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ABSTRACT:

Passenger's comfort in moving vehicles depends on the quality of the ride. The major cause of discomfort is the vibration transmitted to passengers due to the road irregularities. For a comfortable ride on a vehicle, vibration must stay within prescribed standards. In the present work, an attempt was made to show that, the vibrations can be limited with the use of Magneto-rheological (MR) dampers for varying road profiles than the passive damping methods. MR dampers are semi-active control devices that use MR fluids to produce controllable damping force as they are known to exhibit nonlinear behaviour. Multi body dynamic studies were done to study the response of the system using a quarter car model. In this paper, passive damping (viscous damping) was considered at natural frequency of 1.012Hz, the response of damping was observed after 10s and the acceleration was found to be 6m/s^2. When MR damper is employed as the magnetic force increases, the response of the damping was better than the passive damping, at 1.2A it comes down to 0.55m/s^2, and the vibration gets dampened after 1.75s. Hence, from this study it is concluded that the MR damper can be employed in automobile for better ride comfort.

KEYWORDS:

Magneto-rheological damper; Riding comfort; MSC ADAMS; Nonlinear damping

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1. Introduction

Magneto-Rheological (MR) dampers are termed as semi-active control devices that use MR fluids to produce controllable dampers. They offer significantly reliable operation and can be viewed as fail-safe in that they become passive dampers if the control hardware doesn't function. Normal damper has got compression cycle and expansion cycle. In the compression cycle, the fluid inside one chamber moves to the other chamber through holes in the piston, the size of which is very small which restricts the motion of the motion of the fluid and because of the viscosity of the fluid the damping effect is obtained. This is known as passive damping. But this has got its own limitations, like a high damping system performs well in the resonant frequency and poorly away from it; but it is converse in case of low damping system. To overcome this, there was a need for the controlled suspension system where spring constant and damping coefficient happens to be closed loop.

The external energy consumed to control this smart suspension is an important issue to be considered. The controller design must be such that it balances between the effectiveness of the controller and the energy consumption. Active suspension system performs highly than any other system but it consumes a lot of energy. This can achieved only at the expense of complex and expensive system. Semi-active control devices are reliable compared to passive dampers and are as versatile as active suspension system without the requirement of large amount of energy. MR dampers interact with the electronic system and controls the mechanical properties of the fluid inside the damper and thus the damping force can be continuously adjusted. In addition, because of its small size it can fit next to the existing passive damper that is in the working state. The yield stress of MR fluid varies with applied magnetic field. MR fluids are non-colloidal suspension of magnetic particles of size 5-10 micron in the base fluid like water or silicone oil. These are the fluid which changes their viscosity with varying magnetic field. On application of magnetic field these micro particles form a linear chain like structure parallel to the applied field as shown in Fig. 1 and they become thicker causing a resistance to the fluid motion.

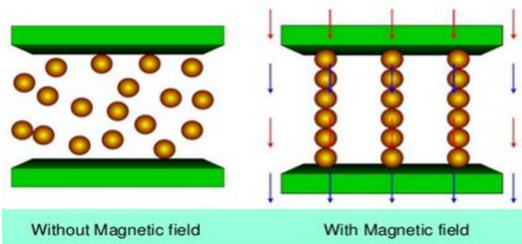

Fig. 1: MR fluid used in MR damper

1.1. MR dampers and their characteristics

The dependency of the MR driving force on the applied magnetic field intensity, linear dependency of the driving force on piston velocity and in connection to its construction that it is robust, flexible and therefore easy to mount and maintain [1]. A new model format is proposed which represents an extension of earlier work by the authors. The proposed model is more general and yet maintains the physical significance of key parameters. A novel model updating (or system identification) technique is developed so that the model can account for the behaviour of various configurations of device without the need for prior knowledge of the fluid properties. The modeling technique is completed by establishing empirical shape relationships between the pre-yield parameters, post-yield parameters, yield force and the applied excitation conditions. The modeling and identification procedures are applied to an MR damping device and the results are validated by comparing predicted and experimental responses under both non-sinusoidal and broadband excitation conditions [2].

A comparative evaluation of the currently available parametric models showed that the simple algebraic parametric models exhibited considerably better predictions than the much more complicated ordinary differential parametric models [3] [9]. MR vibration damper suitable for vehicle suspension was designed and fabricated using the MR fluid. The results indicate that the two damper configurations exhibit different force-displacement characteristics during impulse loading. For the single stage, double-ended damper, the peak force occurs close to the beginning of the impact. Conversely, the two stages, mono-tube damper does not reach the peak force until after the nitrogen accumulator bottoms out [4] [5]. The predictions from theoretical simulations based on the mathematical model [6] are validated using the data collected from the experiments. This led to the inference that the modeling procedure represents the MR damper very satisfactorily.

The energy dissipated and equivalent damping coefficient of the MR damper in terms of input voltage, displacement amplitude and frequency are investigated. The relative displacement with respect to the base excitation is also quantified and compared with that of the conventional viscous damper through updating the equivalent damping coefficient with changing driving frequency. In addition, the transmissibility of the MR damper system with semi-active control is also discussed. The results of this study are valuable for understanding the characteristics of the MR damper to provide effective damping for the purpose of vibration isolation or suppression [7]. A mathematical model based on the flow of MR fluids through an annular gap is developed. Central to the model is the solution for the flow of MR fluid model with a yield stress through the annular gap inside the damper.

The physical parameters of a MR damper designed and fabricated at the University of Manchester are used to evaluate the performance of the damper and to compare with the corresponding predictions of the parallel plate model. Simulation results incorporating the effects of fluid compressibility are presented, and it is shown that this model can describe the major characteristics of such a device, nonlinear, asymmetric, and hysteretic behaviours successfully [8]. The feasibility and applicability of a semi active MR shock isolation system to replace a conventional passive shock isolation system for commercial off-the shelf (COTS) equipment, improved shock mitigation performance, semi active control strategies such as skyhook and sliding mode control were incorporated in the analysis. Controlled responses of the semi active MR shock isolation system were simulated and compared with those of a conventional passive shock isolation system for two representative shock loads for COTS equipment [10].

1.2. Automotive applications of MR dampers

The control of the stationary response of a half car vehicle model moving with a constant velocity over a rough road with MR dampers is considered. The MR dampers were modelled by the modified Bouc-Wen model. The MR damper parameters adopted in the vehicle model correspond to an actual fabricated damper and are determined so that the MR damper model characteristics match with experimental characteristics. The random road excitation is considered as the output of a first order linear shaping filter to white noise excitation. The control and response statistics of the nonlinear vehicle model with MR damper are obtained using the equivalent linearization method in an iterative manner and the results are verified by Monte-Carlo simulation [11].The theoretical and experimental studies taken up for the design, development and testing of a completely new MR damper model that can be used for the semi-active control of automotive suspensions.

The MR damper technology presented in this paper is based on an approach where, in contrast to the conventional solutions where the coil axis is usually superposed on the damper axis and where the inner cylindrical housing is part of the magnetic circuit, the coils are wound in a direction perpendicular to the damper axis. The paper investigates approaches to optimizing the dynamic response and provides experimental verification [12]. An optimal design of a passenger vehicle MR damper was based on finite element analysis. The MR damper is constrained in a specific volume and the optimization problem identifies the geometric dimensions of the damper that minimize an objective function. The objective function consists of the damping force, the dynamic range, and the inductive time constant of the damper. After describing the configuration of the MR damper, the damping force and dynamic range are obtained on the basis of the Bingham model of an MR fluid. Then, the control energy (power consumption of the damper coil) and the inductive time constant are derived. The objective function for the optimization problem is determined based on the solution of the magnetic circuit of the initial damper. Subsequently, the optimization procedure, using a golden-section algorithm and a local quadratic fitting technique, is constructed via commercial finite element method parametric design language. Using the developed optimization tool, optimal solutions of the MR damper, which are constrained in a specific cylindrical volume defined by its radius and height, are determined. A comparative work on damping force and inductive time constant between the initial and optimal design is undertaken. Innovative designs of MR damper were also attempted in [13] - [16].

1.3. Passenger comfort

Quality of the ride and driving pleasure depends on the passenger comfort in the moving vehicle. Vibration transmitted to the vehicle because of the road unevenness, aerodynamic forces etc. cause the discomfort. The major cause being the road unevenness, the vibration must stay well within the desired limits. Analysis of automotive involves mathematical models of variable level of complexity based on the parameters under focus and are correspondingly called as full car, half-car and quarter car models. Since, the acceleration of the sprung mass plays a key role in the assessment of comfort, a quarter car model would be sufficient. Therefore in the present work a quarter car models with an MR damper is used to study the system response.

2. Quarter car simulation model

In this research, 1999 Mercedes ML-430 sport/utility vehicle has been considered where passive damper and semi-active damper is focused (Fig. 2). The physical testing data of an MR damper proposed for the same car has established the force-relative velocity characteristics. Then, the 2 DOF, quarter car model is considered for the analysis with the acceleration of the sprung mass being the point of interest. A simpler model would be computationally efficient for modelling and simulation using MSC ADAMS. The parameters used in the simulated quarter car model are given in Table 1.

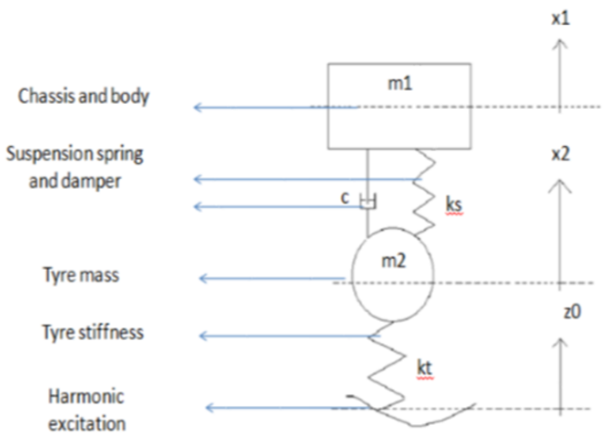

Fig. 2: Quarter car model considered for the analysis

Table 1: Parameters of the quarter car simulation model [7]

Parameter	Description	Value
M1 kg	Sprung mass	600
M2 kg	Unsprung mass	77
K1 N/m	Suspension stiffness	30000
K2 N/m	Tyre suspension stiffness	300000
C1 Ns/m	Suspension damping coefficient	425
g m/s^2	Acceleration due to gravity	-9.81

Modal analysis of the system is carried out to find out the natural frequencies of the system falling in the range of 0-80 Hz since human perception is sensitive in this range. For quantification of the comfort, harmonic analysis is carried out with input in the form of sinusoidal excitations in the range 0 to 80Hz with 0.1m amplitude for simulating the road conditions. Frequency response of the passive damping is considered first and then the MR damper characteristics being the input for different levels of magnetization of the MR fluid achieved by introducing electric current at small constant increments. Vehicle is assumed to move only in the vertical direction. Excitation road profile is input as a continuous time variant excitation along the same direction. Response of the system to harmonic excitation is considered. The specifications of a 1999 Mercedes ML-430 sport/utility vehicle are chosen. The following sinusoidal function is used for the road profile:

$$Y = 0.1^{*}\sin(2t) \tag{1}$$

The maximum amplitude the profile is 0.1m. Fig. 3 shows the response of the system in terms of acceleration of the sprung mass for different damping conditions. Simulation results are obtained in time domain and then converted into frequency domain using FFT transformation in MSC ADAMS. All the generated results for various current values of MR damper are then compared with the passive damping value.

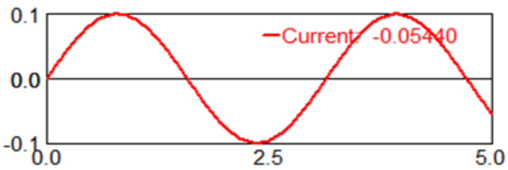

Fig. 3: Road profile for simulation

3. Results and discussion

The output of simulation model with passive damping input of c = 425 Ns/m is shown in Fig. 4. The amplitude 0.1m considered for the present analysis can be assumed to simulate, bumps on the road at smaller frequency to finer road irregularities at higher frequencies. The variation of acceleration of the sprung mass, a maximum with acceleration $7 m/s^2$ at around 1Hz in the presence of the passive damper but conceded to a constant $0.1\ m/s^2$ at a frequency less than 5 Hz. When MR damper is used, the simulations are carried out for the damping varied with the current supply of 0.4A, 0.6A, 0.8A, 1A and 1.2A [7] as shown in Fig. 5(a) to (d). Respective frequency response outputs of the simulation results are shown in Fig. 6(a) to (e).

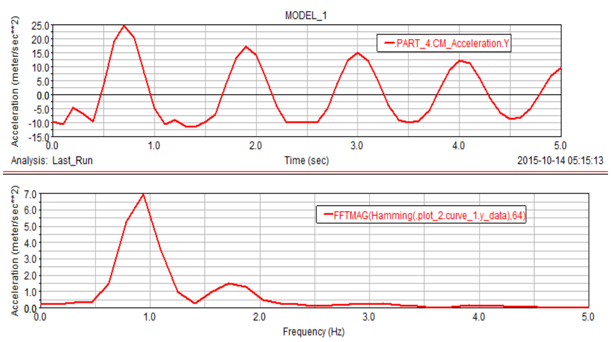

Fig. 4: Frequency response of the quarter car model with a constant passive damping

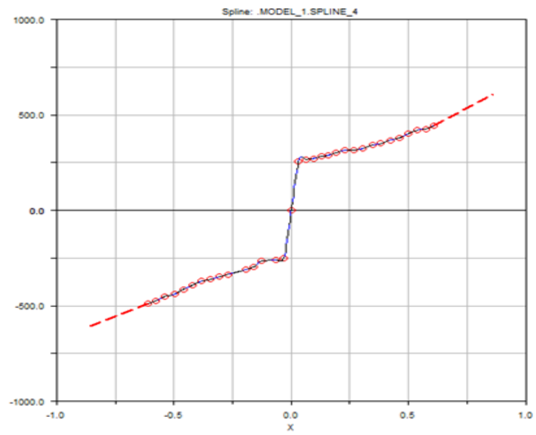

Fig. 5(a): Force vs. Relative velocity at 0.4A for MR damper

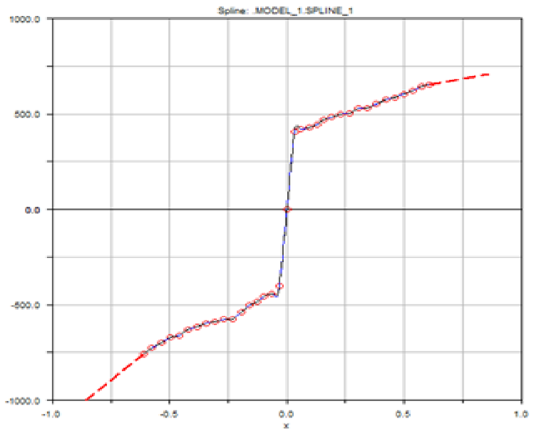

Fig. 5(b): Force vs. Relative velocity at 0.6A for MR damper

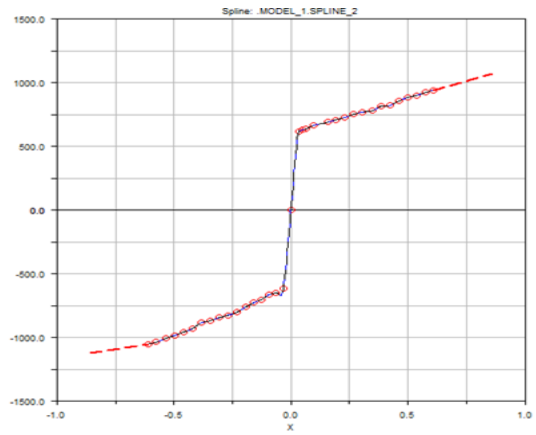

Fig. 5(c): Force vs. Relative velocity at 0.8A for MR damper

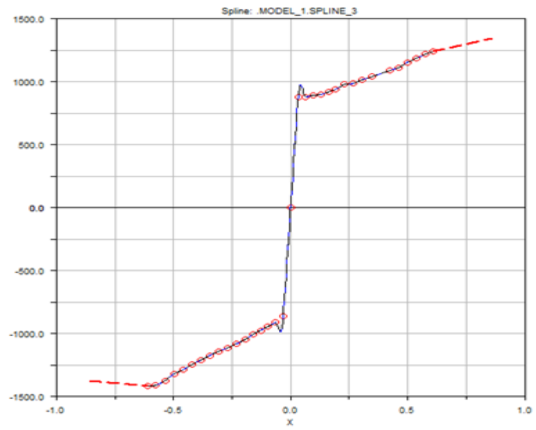

Fig. 5(d): Force vs. Relative velocity at 1A for MR damper

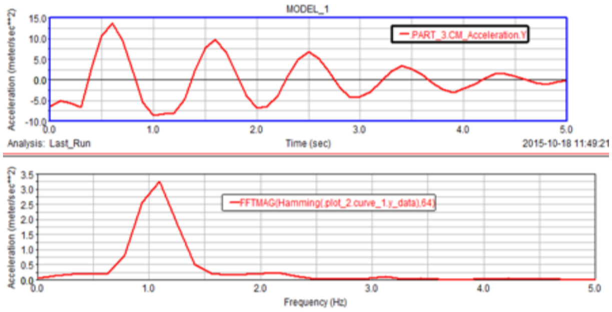

Fig. 6(a): Frequency response of the quarter car model with a nonlinear damping when the current supplied is 0.4A

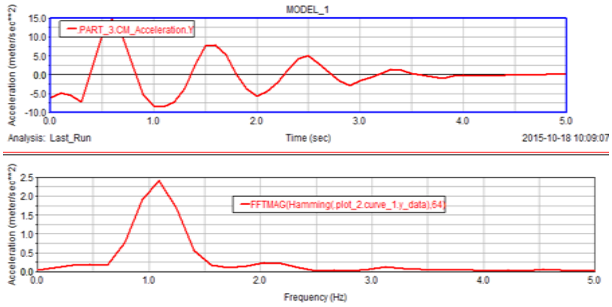

Fig. 6(b): Frequency response of the quarter car model with a nonlinear damping when the current supplied is 0.6A

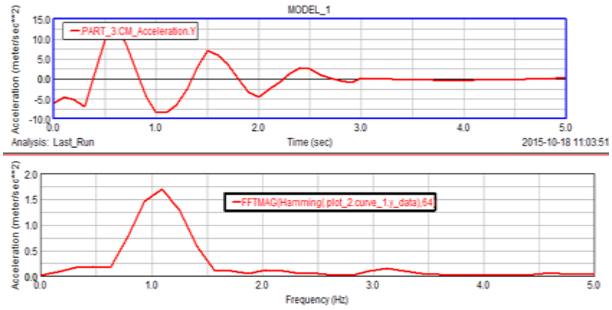

Fig. 6(c): Frequency response of the quarter car model with a nonlinear damping when the current supplied is 0.8A

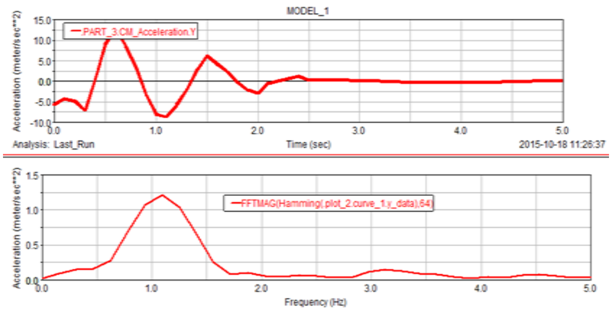

Fig. 6(d): Frequency response of the quarter car model with a nonlinear damping when the current supplied is 1A

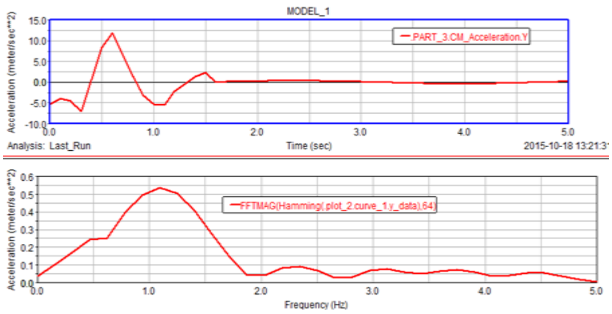

Fig. 6(e): Frequency response of the quarter car model with a nonlinear damping when the current supplied is 1.2A

When MR damper is used at resonance frequency of 1.012Hz, the acceleration would come down to 25m/s^2 when current is 0.4A and vibration gets dampened after 7s. At 0.6A it comes down to 2.25m/s^2, and the vibration gets dampened after 4s. At 0.8A, it comes down to 1.75m/s^2 again and the vibration gets dampened after 2.5s. At 1A it goes down to 1.25m/s^2, and the vibration gets dampened after 2.25s. At 1.2A it comes down to 0.55m/s^2, and the vibration gets dampened after 1.75s. It is clear from the graphs that with the installation of the MR damper, acceleration of the vehicle at the natural frequency which is high when just passive damping is done comes down as the current in the MR damper increases and also the damping happens quickly, i.e. the systems comes to equilibrium position quickly. It can also be observed that, as the applied current is increased the system becomes more and more sensitive to the excitations even at higher frequencies, indicating the rise in the overall rigidity of the system.

4. Conclusion

Quarter car model was used for the comfort analysis by considering the acceleration of the sprung mass. The comfort was quantified through harmonic analysis with an input in the form of sinusoidal excitations in the range 0 to 80 Hz with 0.1 m amplitude for simulating the road conditions. Further, analysis has been carried out for different values of damping. The passengers' comfort increases considerably in the presence of an MR damper as it reduces the acceleration. For general human comfort the level of magnetization of the MR fluid damper should be set between 0.6A to 0.8 A of electric current supplied to the damper as it provides comparatively good comfort by consuming maximum amount of the energy. The passive damper, which is relatively softer, works poorly near the excitation frequency with acceleration crossing 15 m/s^2 and well away from it. The MR damper, which is relatively harder, works fairly well at the resonant frequency with the maximum acceleration limited to values closer to 2m/s^2 and not as well as a passive damper away from it.

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