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MR Damper Characterization for Implementation of Semi-active Suspension Control

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Abstract

The paper proposes experimental method to characterize the Magnetorheological (MR) damper for realization of the suspension control for multi-axle military vehicle. An accurate model of MR damper based on the laws of physics to be embedded in real time controller for suspension system increases the computational load and implementation intricacies attracting higher costs and attendant issues. This paper presents a novel practical based approach to characterize MR dampers using simple experimental set up which, additionally, with the help of simulation technique, aids to assess the dynamic range of MR damper for various parameters of suspension system for vibration control in respect of multi-axle vehicle vehicles. The paper reports experimental investigation for characterization of MR damper through quarter car model. Equivalent damping coefficient is estimated using work diagrams. The governing equations of a quarter car model are formulated analytically and the damping coefficients obtained by experimental investigation have been used for simulation study. Performance analysis in terms of body acceleration, amplitude gain, suspension working space and normalized tyre forces has been carried out. The analysis reveals that with variation of current, there is effective reduction in amplification of the sprung mass displacement at the sprung mass fundamental frequency and improvement in ride comfort. Proposed approach is helpful to achieve better controllability of the MR damper without relying on the software based techniques for accounting the behaviour of MR fluid. The method finds practical application for implementation in military multi-axle vehicles.

Keywords: MR Damper, Military Multi-axle Vehicles, Quarter Car, Simulation, Suspension Control

1. Introduction

Dynamics of an automobile is greatly influenced by suspension system. The suspension parameters in the passive systems are tuned optimally to either achieve ride comfort or vehicle stability. However, the passive systems have numerous limitations which made the researchers to introduce semi-active and active suspension controls. Semi-active suspension systems which have inherent advantages make use of active dampers to obtain improved performance. MR dampers are featured to deliver variable damping when supplied with controlled electrical signal. MR dampers are preferred choice owing to their high yield stress, dynamic fluid properties and fast response time. They also have mechanical simplicity, low power consumption and robustness. The design of better quality suspension is the aim of the vehicle designer. The well accepted vehicle suspension needs to reduce the amplitude of displacement as well as acceleration in vertical direction. This will facilitate ride comfort for the occupants. Also the aim of good suspension is to get better contact of tyre with the terrain. A suspension element

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isolates the sprung mass from road inputs by large suspension deflection through softer damping. As against this, harder damping and less suspension deflection brings good ground contact. Therefore, the quality of ride and the stability of the vehicle during driving are mutually opposite factors which need to have proper trade-off studies. The actual damping values of MR dampers are the very vital parameters in the vibration isolation with respect to body acceleration, amplitude gain, suspension working space and NTF. Hence characterization of damper has been the prime consideration for the suspension researchers. For semi active control systems of the suspension systems, accurate analytical models are required to implement robust control strategies and achieve desired performance indices. Unlike passive dampers, MR dampers cannot be represented by simple models due to inherent nonlinearities. Modelling of MR dampers and its effect on the system performance has been widely reported in literature.

Various illustrations of research pertaining to seat suspensions incorporating MR dampers were introduced[1–3]. A study on system having one degree of freedom with MR damper perturbed by harmonic excitations was also reported[4]. The nonlinear force versus velocity graph of MR damper for semi active suspension system is used for the designated control law[5]. Effect of MR damper constraints on the performance of the MR damper while quantifying the same through control techniques has been reported[6]. Herschel-Bulkley constitutive model for MR damper was presented[7]. Integration of current function in modified linear biviscous, polynomial extended Bouc-Wen and generalized sigmoid function models to predict the current dependent hysteretic damping force was presented[8]. The modelling, theoretical analysis and characterization of small MR fluid damper for typical application and trade off studies between ride comfort and handling for off road vehicles with reference to damping characteristics respectively was also given[9,10]. A comparison of the damping performance of the two MR damper designs using Bingham-Plastic model and low speed hysteresis model was reported[11]. 'Herschel-Bulkley model' with a field dependent yield stress and the impact of shear rate dependent viscosity on damping capacity of MR damper was assessed[12]. Lumped mass model for characterization of MR damper which includes frictional and temperature effects was studied[13]. Field dependent characterization of MR dampers through geometrical dimension optimization of electromagnetic circuit was presented[14]. Two models, viz. 'normalized version of Bouc-Wen model' and the 'Dahl friction model' were tested and their experimental validation have been explored[15]. Experimental evaluation of 'field dependent damping force' and controllability of cylindrical MR damper, subjected to parameter variations and its employment for control of vibration of semi-active suspension system was given[16]. Neural network to estimate the controlled voltage to the input of MR damper to produce the force in optimal way, predicted by the controller, in order to reduce vibrations was developed[17]. Analytical and experimental evaluation of the MR damper through continuum electromechanical model was presented[18]. Time domain hydro mechanical model based on lumped parameters to take in account the non-linear hysteresis behavioral pattern of the MR fluid in order to simulate force versus displacement output response and force-time history for various excitations was given out[19]. Modelling of the dynamic performance of linear MR damper using flow model is reported[20] which is further modelled for hysteretic behavior using mAlg and modified Bouc-Wen model. Four modelling approaches, viz. two static, nonlinear with dynamic filter and probabilistic model to predict multi input and single output behavior of MR dampers were discussed[21]. Recursive lazy learning method based on neural network for modeling MR damper behavior was reported[22]. Experimental method for identifying the MR damper performance was also published[23]. The importance of analytical modelling including parameter optimization of MR dampers was emphasized[24]. MR damper characterization is useful specially in modern control techniques such as those based on neural network based vibration control system[25]. The fire power performance using active type suspension system using linear state feedback system is also presented for combat role[26]. MR damper force dynamics study through variety of experiments were outlined[27] which suggested better modelling for MR damper controllability through rod displacement and electric current sequencing. A robust set of non dimensional parameters to model MR damper and device characterization was proposed[28] and the analysis was done on various constitutive models of 'non Newtonian' fluid models. The identification of optimal parameters of MR damper for semi active logic using non-dominated sorting Genetic Algorithm was presented[29] and the hysteric nature of the MR dampers had been characterized using 'Bingham' and 'modified Bouc-Wen' model. Modelling techniques of rotary MR dampers which have a distinct advantage in terms of smaller MR fluid, compactness and weight penalty was presented[30]. The non-linear dynamics is investigated by numerical solutions where quarter car with non-linear damper is formulated[31] and studied for model of chaotic type for successive speed breaker bumps on highways.

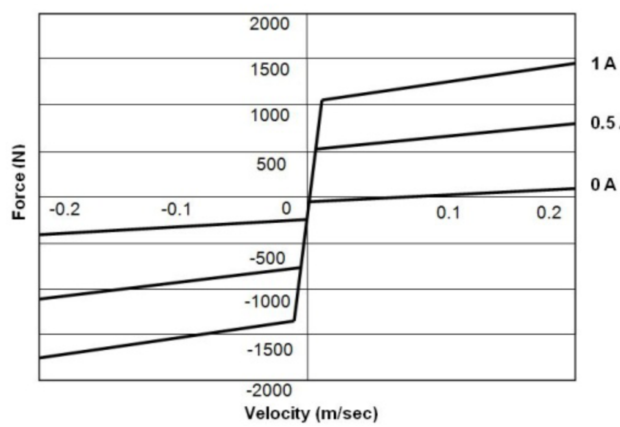
Even though various studies have been proposed in literature to obtain an accurate model of the damper, all of them appear to increase the computational burden on the real time controllers more than what is required. Moreover, modelling of these devices in precise way using the physical laws is an uphill task. Thus there have been limitations in the control implementation for semi active suspension owing to these constraints. Characterization of MR dampers is, therefore, yet to be fully explored. This paper presents a novel practical based approach to characterize MR dampers using simple experimental set up which additionally, with the help of simulation technique, aids to assess the dynamic range of MR damper for various parameters of suspension system for vibration control.

The present work carries out experimental investigation for characterization of MR damper through quarter car model. Equivalent damping coefficient is estimated using work diagrams. The MR damper is assessed for body acceleration, amplitude gain, suspension working space and NTF using simulation techniques. The proposed approach is felt helpful for achieving better controllability of the MR damper in real time applications without relying on the software based techniques for accounting MR fluid behaviour.

2. Experimental Characterization of MR Damper

The MR damper (RD-8041) discussed here as shown in Figure 1a is sourced from Lord Corporation and is a mono-tube shock absorber containing high-pressure nitrogen gas with a pressure of 300 psi. MR dampers typically consist of a piston, magnetic coils, accumulator, bearing, seal and damper reservoir filled with MR fluid. Damper has stroke of 7..4 cm and fully extended length at 24.8 cm. The minimum typical damping force achieved is 2447 N peak-to-peak at 5 cm/sec with 1 A current. It can also be seen from Figure 1b that the output of the damper i.e. force does not exactly get through zero. This is due to the fact that there is an

(a)



(b)

Figure 1. MR damper RD-8041 (a) Photograph of MR damper used for the experimental test (b) Variation of velocity with damper force.

accumulator in the MR damper which is filled with air under compressed condition.

The RD-8041 MR damper has a conventional cylindrical body configuration filled with MR fluid comprising the piston, the magnetic circuit with a coil resistance of 5 Ω and 7 Ω at ambient and at 71°C temperature. Figure 2 shows the experimental test rig used for data generation. The test rig shows the MR damper fixed on servo-hydraulic actuator under computer controlled environment. The servo-hydraulic actuator incorporates load cell to measure the force produced by the MR damper and a displacement sensor to measure the piston displacement. The MR damper was excited using servo-hydraulic actuation system. Tests were run for a combination of frequencies and input current. The sinusoidal excitation was controlled using a controller device and the response was measured via in-built instrumentation of the servo-hydraulic actuation system. The data processing was done in Matlab/Simulink Software.

The sinusoidal excitation was used as input in the frequency range of 0.4 Hz to 3.6 Hz in the steps as given

Table 1. Sinusoidal excitation parameters used for RD-8041 damper characterization

Parameter	Values
Frequencies (Hz)	0.4, 1.2, 2.0,2.8,3.6
Amplitude (mm)	10
Current (A)	0, 0.1, 0.2, 0.3, 0.4,0.5

in Table 1 and the current was varied from 0 to 0.5 A. The force and displacement responses of the damper were sampled simultaneously. Velocity response was obtained by mathematical operation of differentiation. Sample response at excitation frequency of 1.2 Hz under five constant currents is given in Figure 3. It can be seen that with the increase of the applied electric current, the damping force is increased remarkably. However, the input current was increased upto 0.5 A, as the hike in the damping force is insignificant after it reaches saturation of the MR effect which occurs at 0.75 A.

Figure 2. Test rig used for MR damper characterization.

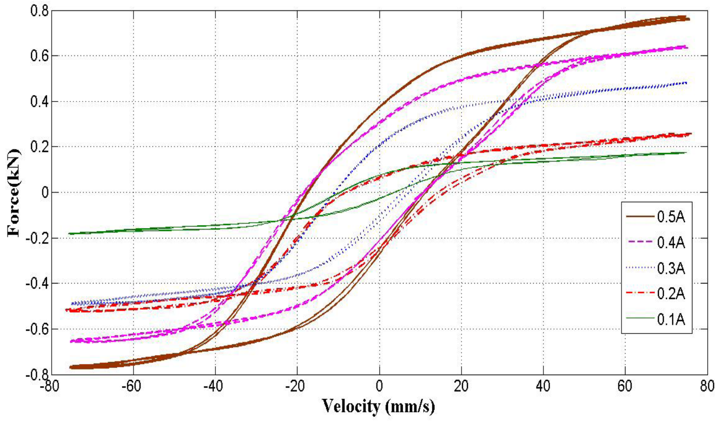


Figure 3. Effect of applied current on the variation of velocity with force of MR damper measured at a frequency of 1.2 Hz and amplitude of 10mm.

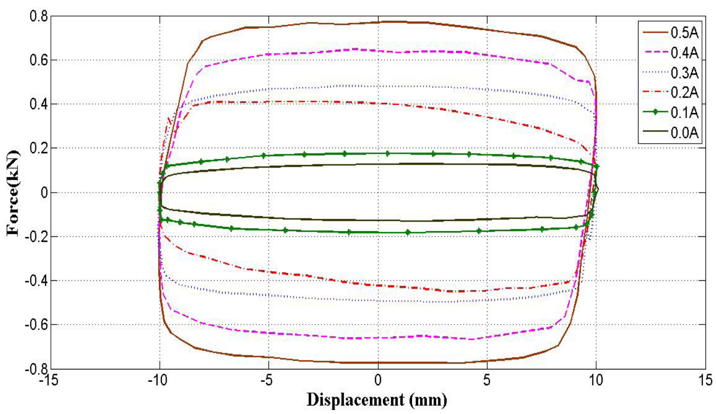


Figure 4. Effect of applied current on the variation of displacement with force of MR damper measured at a frequency of 1.2 Hz and amplitude of 10mm.

Figure 4 gives the work diagram for 1.2 Hz with amplitude (X) of 10 mm. The area inside work diagram (W) is calculated from the area under the curve of the force–displacement diagram for a particular current input, frequency and excitation amplitude. Equivalent damping coefficient C_{eq} is calculated as follows:

$$C_{eq} = \frac{W}{\pi \omega_d X^2} \tag{1}$$

where, W is area inside work diagram (kN-mm); ω_d is the rotational frequency (rad/s) and X is the excitation amplitude (mm).

Figure 5 shows the variation of equivalent damping coefficients of the MR damper with forcing frequency under various currents. It is observed that at lower frequencies, damping coefficient (equivalent) increases significantly. However, damping coefficient (equivalent) under high current decreases significantly as the fre-

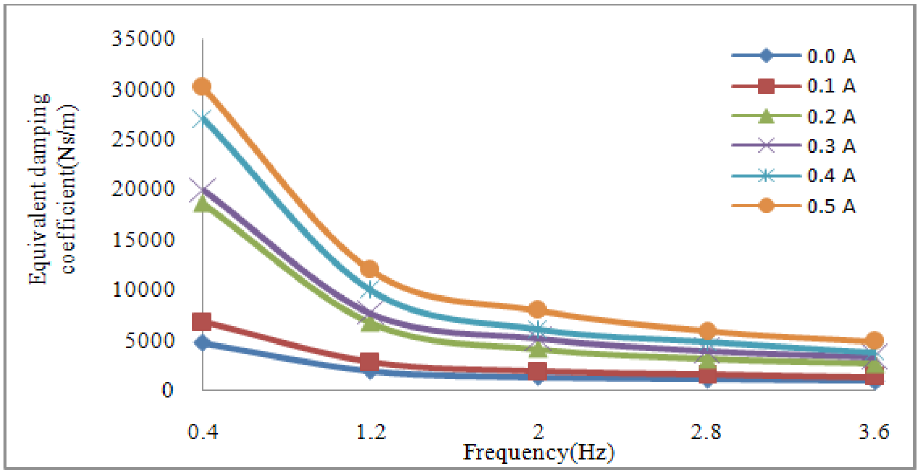


Figure 5. Variation of equivalent damping coefficients at various excitation frequencies with various applied current.

quency increases. The same decreases slowly without the current. It is also observed that the effect of current on equivalent damping coefficient is not significant at higher frequencies. The experiment reveal that the designed MR damper has very high variable damping force under the presence of magnetic field ensuing better controllability.

3. Simulation of Quarter Car Model

The suspension system is aimed at achieving two main functions. First, it has to provide a good amount of vehicle body isolation from the inputs applied between the wheels and road, irregularities of tyre and out-of-balance forces arising out of wheels to ensure operator comfort. Also it should keep the wheels and road surface in close contact (i.e. tyre forces) in order to ascertain proper adhesion when accelerating, braking or cornering. Effect of change in damping coefficient on suspension performance as regards body acceleration/amplitude, working space of suspension and normalised tyre force can be affectively analysed using a quarter car model having the characterisics of MR damper as shown in Figure 6. M_s represents the sprung mass M_u is the unsprung mass, k represents the stiffness of suspension system and k_t is the stiffness of tyre and C is the variable damping coefficient,

Table 2. Quarter car model parameters used for simulation

Parameter	Values
Sprung mass (ms)	205 kg
Un sprung mass (mu)	30kg
Spring stiffness (Ks)	20570 N/m
Tyre Stiffness (Kt)	142000 N/m

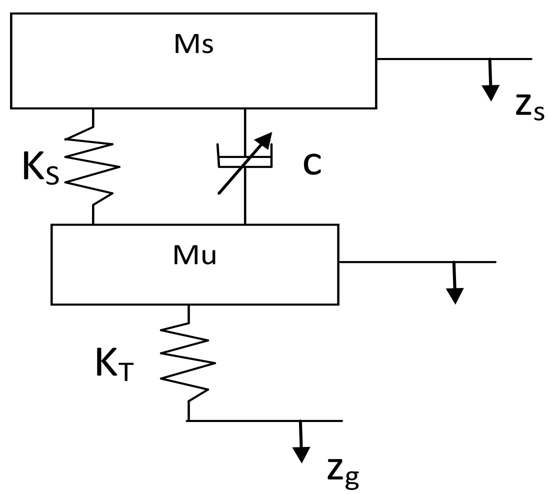


Figure 6. Simplified quarter car model.

Z_s is sprung mass displacement, Z_u is unsprung mass displacement and Z_g is road input displacement. An electric current applied to MR damper is of varying magnitudes such as 0.1 A, 0.3 A and 0.5 A. MATLAB environment is used for simulation. The various parameters used in the simulation of the quarter car model are provided in Table 2. In the simulation, it is assumed that response of MR damper is very fast and thus time delay is neglected.

Following governing equations are modeled in Matlab-Simulink software.'

$$M_S\ddot{Z}_S = -K_S\left(Z_S - Z_U\right) - C_S\left(\dot{Z}_S - \dot{Z}_U\right) \quad (2)$$

$$M_U\ddot{Z}_U = K_S\left(Z_S - Z_U\right) + C_S\left(\dot{Z}_S - \dot{Z}_U\right) - K_T\left(Z_U - Z_G\right) \quad (3)$$

4. Results and Discussions

Figure 7 depicts how the body acceleration gain varies under various amplitudes of current applied to MR damper at various excitation frequencies. It is observed that when constant current control is applied, the acceleration between body resonance and wheel hop is very high compared to those with passive control i.e. at 0 A in the given frequency range. Also it is observed that the body acceleration response around the body resonance is isolated effectively with higher current values. It is also noticed that body acceleration gains around the wheel hop increases with increase in the current.

Figure 8 shows the variation of body amplitude gain with damper current at various excitation frequencies of the MR damper. It can be seen that the response is quite similar to that of the body acceleration gain. When the current is constant, the amplitude gains in the frequency range between body resonance and wheel hop is very high in comparison with passive control i.e. at 0 A. Body amplitude gain around the sprung mass fundamental frequency (body resonance) is isolated effectively using higher current values. However, around the unsprung mass fundamental frequency (wheel hop) body amplitude gain increases with increase in current.

Figure 9 shows the suspension working space under different values of constant current. Suspension working space refers to the motion between bump and rebound stops. It is seen that suspension travel response around

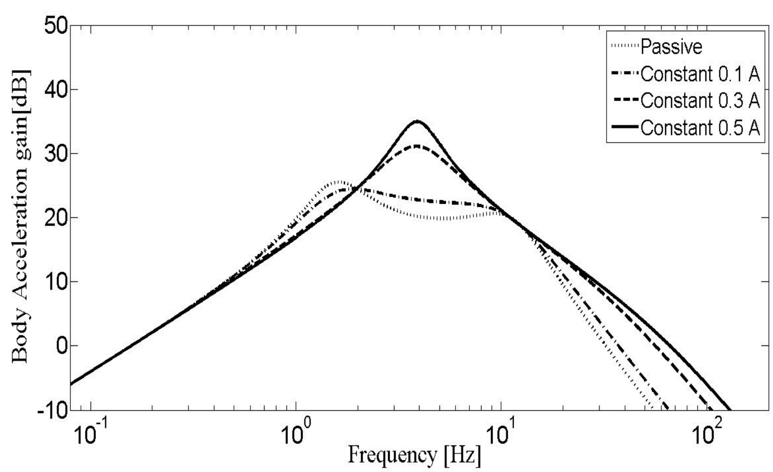


Figure 7. Variation of body acceleration gain with applied current at various excitation frequencies.

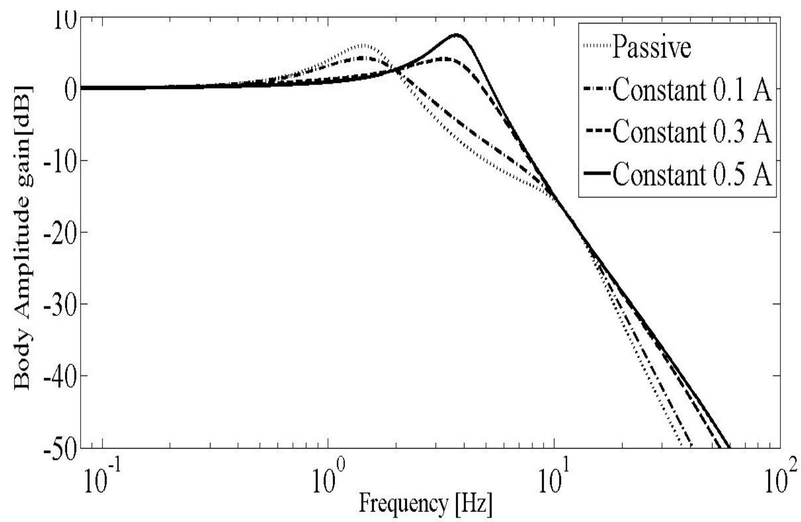


Figure 8. Variation of body amplitude gain with applied current at various excitation frequencies.

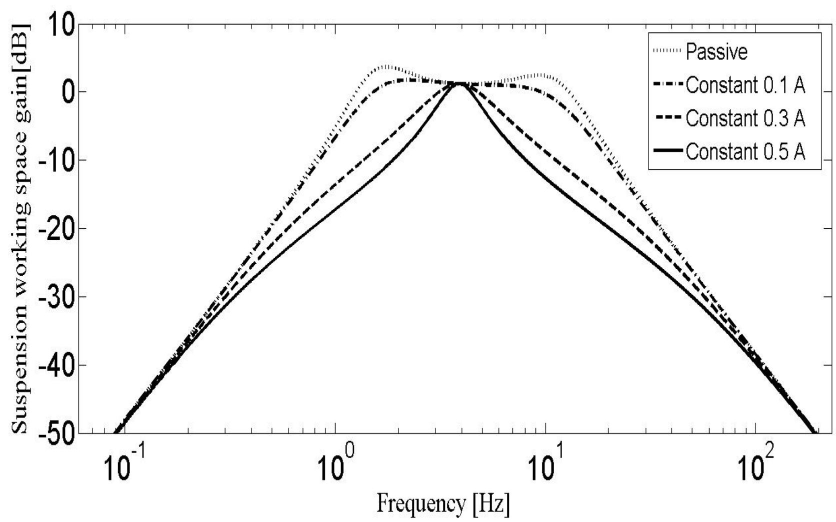


Figure 9. Variation of suspension work space with applied current at various excitation frequencies.

the body resonance and around the wheel hop is reduced to a significant extent with constant current control.

Figure 10 shows the variation of normalized tyre force (NTF) with applied current to MR damper at various excitation frequencies. This can result in considerable time variations of the tyre normal force around (quasi) static equilibrium, which in turn can lead to a loss of contact with the ground and reduced handling ability. Under

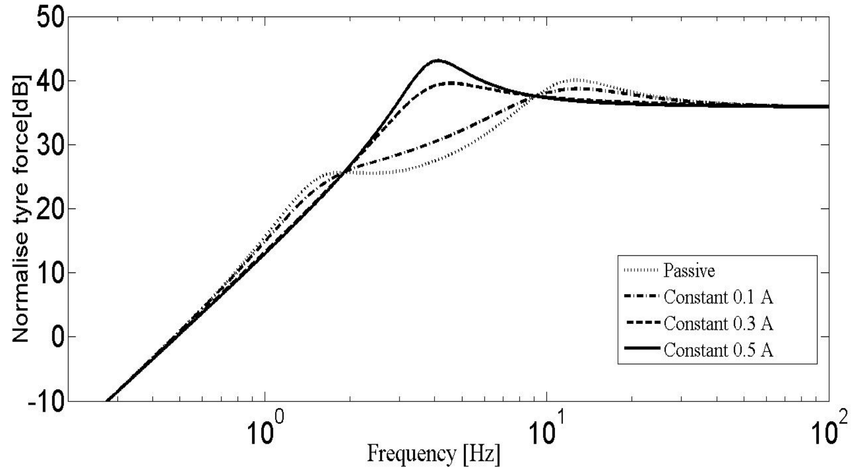


Figure 10. The variation of normalized tyre force with frequency.

constant current control, the NTF gain in the frequency range between body resonance and wheel hop is very high compared to current at 0 A. The NTF gain around the body resonance and the wheel hop is isolated effectively using higher current values.

5. Conclusions

Ride quality and stability during driving of the vehicle are two conflicting requirements to be achieved by implementing control using MR dampers. Hence characterization of MR damper is prime consideration for the suspension researchers. In the present study, characterisation of MR damper using experimental technique has been carried out. Dynamic range estimation of the MR damper is presented which shows that the MR damper's range for changeable damping force is very large. The damping coefficient is experimentally arrived at which is also ascertained for the current range to ensure adequate controllability for the given application.

The governing equations of a quarter car model are formulated analytically and damping coefficients obtained by experimental investigation have been used for simulation study. Performance analysis in terms of body acceleration, amplitude gain, suspension working space and normalized tyre forces has been carried out for the quarter car model. From the quarter car analysis, it is inferred that by varying the current, effective reduction in amplification of the sprung mass displacement at the sprung mass fundamental frequency, along with improvement in ride comfort can be achieved by implementation of suitable semi-active control strategy with the said MR damper. Simulation results reveal that the suspension performance in terms of sprung mass acceleration and displacement, working space of suspension and normalized tyre forces for the vehicle can be effectively improved through employment of suitable control strategy using candidate MR damper.

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