Ride Comfort Characteristics with Different Tire Pressure of Passenger Vehicle Featuring MR Damper

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Abstract. This paper presents ride comfort characteristics of a quarter-vehicle magneto-rheological (MR) suspension system with respect to different tire pressure. As a first step, controllable MR damper is designed and manufactured based on the optimized damping force levels and mechanical dimensions required for a commercial mid-sized passenger vehicle. After experimentally evaluating dynamic characteristics of the manufactured MR damper, the quarter-vehicle suspension system consisting of sprung mass, spring, tire and the MR damper is constructed in order to investigate the ride comfort. After deriving the equations of the motion for the proposed quarter-vehicle MR suspension system, vertical tire stiffness with respect to different tire pressure is experimentally identified. The skyhook controller is then implemented for the realization of quarter-vehicle MR suspension system. Ride comfort characteristics such as vertical acceleration RMS of sprung mass are evaluated under bump road condition and presented in time domain.

1. Introduction

The vehicle dynamic characteristics such as ride comfort and steering stability can be normally improved by suspension systems [1-2]. So far, passive oil damper, which provides design simplicity and cost-effectiveness, is widely employed for conventional vehicles. However, the performance limitation is inevitable due to uncontrollable damping force. Recently, active damper using a motor and electro-servo hydraulic valve is gradually used to achieve ride comfort as well as steering stability. However, design complication and high cost prevent the popularization of advanced active suspension systems. Therefore, alternative mechanisms for vehicle suspension systems have been studied to replace the passive and active dampers. For the last decade, the research work on ride comfort and steering stability of a vehicle system using semi-active suspension has been significantly increased. The semi-active suspension system using magnetorheological (MR) damper can offer a desirable performance in the active mode without requiring large power consumption and expensive hardware. MR fluid, whose rheological properties can be rapidly changed by applying a magnetic field to the fluid domain, has been used in many devices such as shock absorbers, valves, engine mounts and clutch/brake systems. Recently, Choi et al. [3] manufactured a MR damper for a small-sized passenger vehicle and presented its control characteristics of the damping force. On the basis of this
work, they extended their research and evaluated control performance of the proposed MR damper via hardware-in-the-loop simulation [4].

The main contribution of this work is to present ride comfort characteristics of a quarter-vehicle suspension system equipped with continuously controllable MR damper with respect to different tire pressure. As a first step, controllable MR damper is designed and manufactured based on the optimized damping force levels. After experimentally evaluating dynamic characteristics of the manufactured MR damper, equation of the motion for the proposed quarter-vehicle suspension system consisting of sprung mass, spring, tire and the MR damper is derived in order to investigate the ride comfort. Subsequently, vertical tire stiffness with respect to different tire pressure is experimentally identified. Ride comfort characteristics such as vertical acceleration RMS of sprung mass are evaluated under bump road condition.

2. MR Damper

The schematic configuration of the MR damper proposed in this work is shown in figure 1. The MR damper is composed of the cylinder, piston and gas chamber. The floating piston is also used in order to compensate the volume induced by the motion of the piston. The cylinder and the gas chamber in the MR damper are fully filled with the MR fluid and the nitrogen gas, respectively [5].

In the absence of the magnetic field, the MR damper produces the damping force only caused by the fluid viscous resistance. However, if a certain level of the magnetic field is supplied to the MR damper, the MR damper produces additional damping force owing to the yield stress of the MR fluid. This damping force of the MR damper can be continuously tuned by controlling the intensity of the magnetic field. Therefore, the damping force of the proposed MR damper can be written as follows:

$$F_D = k_e x_e + c_e \dot{x}_e + F_{MR}$$

where $k_e$ is the effective stiffness due to the gas pressure, $c_e$ is the effective damping due to the fluid viscosity, $x_e$ is the excitation displacement, and $F_{MR}$ is the field-dependent damping force. The controllable damping force $F_{MR}$ can be expressed by

$$F_{MR} = (A_p - A_r) \frac{c L_p}{h} \tau_y(B) \cdot \text{sgn}(\dot{x}_e)$$

where $c$ is a coefficient which depends on flow velocity profile, $L_p$ is the length of the magnetic pole, $h$ is the gap between outer and inner cylinder and $\tau_y(B)$ is the yield stress caused by the magnetic flux density. $A_p$ and $A_r$ represent piston and piston rod areas, respectively; and $\text{sgn}(\cdot)$ is a signum function.

The size and the level of required damping force is determined so that the MR damper can be applicable to a commercial passenger vehicle. In order to obtain optimal geometric dimensions of the proposed MR damper, commercial FEM software is adopted [5]. Figure 2 presents the damping force characteristics of the optimal MR damper with respect to the piston velocity at various magnetic fields. This is obtained by calculating the maximum damping force at each velocity.

3. Quarter-Vehicle Model

In this study, a quarter-vehicle MR suspension system is established to evaluate control performances of the manufactured MR damper with optimal parameters. Figure 3 shows the quarter-vehicle model
Figure 3. Quarter vehicle model of MR suspension

Figure 4. Stiffness test of the tire

Figure 5. Vertical stiffness of the tire

of the semi-active MR suspension system which has two degree-of-freedom. \( m_s \) and \( m_u \) represent the sprung mass and unsprung mass, respectively. The spring for the suspension is assumed to be linear and the tire is also modeled as linear spring component. Now, by considering the dynamic relationship, the state space control model is expressed for the quarter-vehicle MR suspension system as follows:

\[
\dot{x} = Ax + Bu + Lz_r
\]

\[
\dot{y} = Cx
\]

where

\[\begin{bmatrix}
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & F_{MR} \\
-k_s & -c_s & -k & -c_m & -1 & 0 & 0 & 0 & 0 \\
m_s & m_s & m_s & m_s & m_s & m_s & k & c & 0 \\
0 & 0 & 0 & 0 & 0 & m_s & m_s & c_m & 0 \\
m_u & m_u & m_u & m_u & m_u & m_u & k & c & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
\end{bmatrix},
\]

\[\begin{bmatrix}
0 & 0 & 0 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 \\
\end{bmatrix}
\]

\[B = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 \end{bmatrix}^	op
\]

\[L = \begin{bmatrix} 0 & 0 & 0 & 0 & k \end{bmatrix}
\]

\[C = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \end{bmatrix}
\]

where \( k_s \) is the total stiffness coefficient of the suspension including the effective stiffness \( k_s \) of the MR damper in equation (1). \( c_s \) is the damping coefficient of the suspension and it is assumed to be equal to \( c_m \). \( k \) is the stiffness coefficient of the tire. \( z_s \), \( z_u \), and \( z_r \) are the vertical displacement of sprung mass, unsprung mass and road excitation, respectively. In addition, \( \tau \) is the time constant of the MR damper and \( u \) is the control input.

The photograph of vertical stiffness test for the tire is shown in figure 4. Figure 5 presents the vertical stiffness characteristics of the tire with respect to different tire pressure. It is clearly observed that the vertical stiffness is increased as tire pressure increases. The vertical stiffness of 153kNm at tire pressure 124kPa is increased up to 225kNm at 227kPa.

Among many potential candidates for control algorithms, the skyhook controller is adopted in this work. It is well known that the logic of the skyhook controller is simple and easy to implement to real field [5]. The desired damping force is set by

\[u = C_{so} \cdot \dot{z}_s \]

where \( C_{so} \) is the control gain which physically indicates the damping coefficient. In above equation, control input \( u \) directly represents the damping force of \( F_{MR} \).

4. Performance Evaluation

Ride comfort characteristics of the quarter-vehicle MR suspension system with respect to different tire pressure are evaluated under bump road excitations [3]. Figure 6 and 7 presents ride comfort comparison of the quarter-vehicle MR suspension system under bump road excitation with constant vehicle velocity of 3.08km/h. As shown in figure 6, it is obvious that unwanted vibrations induced from the bump excitation have been well reduced by adopting the MR damper associated with the
skyhook controller. It is also observed from the vertical displacement of sprung mass that the control performance of the controlled case is better than that of the uncontrolled case independent of tire pressure. The RMS (root mean square) of vertical acceleration ($\ddot{z}$) shown in figure 7 indicate that ride comfort characteristics of a vehicle system can be substantially improved by employing the MR damper and skyhook controller even though tire pressure is changed during driving.

5. Conclusion
In this work, ride comfort characteristics of a quarter-vehicle magneto-rheological suspension system with respect to different tire pressure was undertaken. The MR damper was designed and manufactured based on the optimized damping force levels and mechanical dimensions required for a commercial mid-sized passenger vehicle. After experimentally evaluating dynamic characteristics of the manufactured MR damper, the quarter-vehicle suspension system consisting of sprung mass, spring, tire and the MR damper was constructed in order to investigate the ride comfort. After deriving the equations of the motion for the proposed quarter-vehicle MR suspension system, vertical tire stiffness with respect to different tire pressure was experimentally identified. The skyhook controller has been then implemented for the realization of quarter-vehicle MR suspension system. Ride comfort characteristics such as vertical displacement and acceleration RMS of sprung mass were evaluated under bump road condition. The results presented in this work are quite self-explanatory justifying that the MR damper with the controller can provide improved ride comfort independent of tire pressure.

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References
[1] T.D. Gillespie, Fundamentals of Vehicle Dynamics, Society of Automotive Engineers, Warrendale, PA, 1992.
[2] E.M. Elbeheiry, D.C. Karnopp, M.E. Elaraby and A.M. Abdelraaouf, “Advanced ground vehicle suspension systems - a classified bibliography”, Vehicle System Dynamics 24, 231-258 (1995).
[3] H.S. Lee and S.B. Choi,” Control and response characteristics of a magnetorheological fluid damper for passenger vehicles”, Journal of Intelligent Material Systems and Structures 11, 80-87 (2000).
[4] S.B. Choi and K.G. Sung, “Vibration control of magnetorheological damper system subjected to parameter variations”, International Journal of Vehicle Design 46, 94-110 (2008).
[5] K.G. Sung, Y.M. Han and S.B. Choi, “Geometric Optimization of Controllable Magnetorheological Shock Absorber for Commercial Passenger Vehicle”, Proceedings of the SPIE’s 11th Annual Symposium on Smart Structures and Materials, 9-13 March, 2008, San Diego, California.