Performance Evaluation on Vibration Control of MR Landing Gear

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Abstract. This paper is concerned with the applicability of the developed MR damper to the landing gear system for attenuating undesired shock and vibration in the landing and taxing phases. First of all, the experimental model of the MR damper is derived based on the results of performance evaluations. Next, a simplified skyhook controller, which is one of the most straightforward, but effective approaches for improving ride comport in vehicles with active suspensions, is formulated. Then, the vibration control performances of the landing gear system using the MR damper are theoretically evaluated in the landing phase of the aircraft. A series of simulation analyses show that the proposed MR damper with the skyhook controller is effective for suppressing undesired vibration of the aircraft body. Finally, the effectiveness of the simulation results are additionally verified via HILS (Hardware-in-the-loop-simulation) method.

1. Introduction
A magneto-rheological (MR) fluid is a non-colloidal solution which is composed of ferromagnetic particles with micrometers in diameter dispersed in a non-conductive carrier fluid [1]. MR fluids have the characteristic that their rheological properties can be continuously and reversely changed within several milliseconds solely by applying or removing external magnetic fields. Due to this feature, MR fluids are increasingly being considered in a large variety of devices such as shock absorbers, vibration insulators, brakes or clutches [2]. Compared to conventional actuation methods, devices using the MR fluid can be made simple in construction, high in power, and low in inertia. Therefore, the use of the MR fluid actuators is very effective for improving the function and performance of conventional control systems. Recently, the authors have proposed an electromagnetic design methodology for the magneto-rheological (MR) damper [3]. In order to improve the performance of the MR damper, the magnetic field should be effectively supplied to the MR fluid. For this purpose, two effective approaches were proposed: one is to shorten the magnetic flux path by removing the unnecessary bulk of the yoke in order to improve the static characteristic of the MR damper, and the other is to increased the magnetic reluctance of the ferromagnetic material by minimizing its cross-sectional area through which the magnetic flux passes in order to improve the dynamic and hysteretic characteristics. The superiority of the proposed design methodology to a conventional one was verified through the magnetic field analysis and a series of basic experiments. As further study, this paper is concerned with the applicability of the developed MR damper to landing gear system for attenuating undesired vibration in the landing phase.
2. MR damper

Fig. 1 depicts the schematic configuration of an MR damper considered in this work. This MR damper consists of a MR fluid, a casing, a gas accumulator, and a piston head in which the magnetic circuit is installed. The casing (SS41) is fully filled with the MR fluid (MRF-132AD) and divided into upper and lower chambers by the piston head. When the piston head moves, the MR fluid is moved through the annular gap between the casing and the piston head from one chamber to the other, where the MR fluid is exposed to an applied magnetic field. In order to compensate for changing the fluid volume occupied by the piston rod in the upper chamber, the gas accumulator is located in the lower chamber. The bush and the washer (MC nylon) are employed for preventing leakage of magnetic field, and the guide piston with the porous structure (MC nylon) are installed to keep the gap size constant. When an external current is supplied to the electromagnet coil, a magnetomotive force is generated in the magnetic circuit and the corresponding magnetic field is applied to the MR fluid. Then, the dynamic yield stress of the MR fluid is changed depending on the magnetic field intensity, and the resultant damping force is activated in the opposite direction to the motion of the piston head. In the absence of a magnetic field, the damping force occurs only due to the viscosity of the MR fluid itself. However, in the presence of a magnetic field, an additional damping force occurs due to the MR effect of the MR fluid. Therefore, the damping force of the MR damper can be continuously and reversely controlled by adjusting the applied coil current. The external appearances of the manufactured dampers are shown in Fig. 2, and its full stroke is 80 mm (ruler scale: 30cm). Now, the experimental model of the MR damper is developed based on the results of performance evaluations obtained from the previous research. The magnetic field-dependent damping characteristic of the MR damper can be effectively described by the Bingham plastic model. Although an MR fluid behavior actually exhibits some departures from the Bingham model due to its viscoelastic property, it has been proved by numerous researchers that this simple model is very useful for the design and modeling of MR fluid-based devices [4][5]. Then, the constitutive equation of the MR damper can be expressed as

\[ \tau \ddot{d} + F_d = F_{MR} + F_{\eta} + F_g + F_f \]  

The above equation is written in SI units. The first term in right hand side is the damping force due to the magnetic field-dependent characteristic of the MR fluid (the MR effective force), and was found by \( F_{MR} = (426.8 I - 3.833) \text{ sgn}(v) \approx 426.8 I \text{ sgn}(v) \), where \( I \) is the coil current and \( v \) is the excitation velocity of the piston head. Especially, it is notable that the MR effective force of the considered MR damper is nearly linear to the coil current. The second term represents the viscous damping force due to the viscosity of the MR fluid itself, and was obtained as \( F_{\eta} = 702.2 v \). Also, by assuming that the gas in the accumulator undergoes a polytropic process during operations, the gas spring force was found by \( F_g = 78.54 (11634 I (11634 - 78.54 d))^{1.4} \), where \( d \) is the displacement of the piston head. The frictional force due to the oil seals and the guide piston was found by \( F_f = 63 \text{ sgn}(v) \). Finally, the differential term in the left hand side of (1) represent the dynamic characteristic of the MR damper. In general, the damping force of the MR damper for the step input current exhibits the dynamic behavior similar to a 1\(^{st}\) linear system. Note that the characteristic time constant \( \tau \) is
dependent on the excitation velocity of the MR damper as well as the field-induced condition whether is field-on or field-off; \( \tau = \tau(v) \).

3. Simplified skyhook control

In the literature, several control schemes have been developed to improve ride quality of active or semi-active damper systems. One of the most straightforward, but effective approaches is the skyhook controller initially introduced by Karnopp and Cosby [6]. The main advantages of this controller are its simple implementation and easy understanding of the relationship between design and performance. Especially, this controller is applicable for both a semi-active system and an active system. These merits have made the skyhook controller adopted in a large number of real applications and used as the reference control scheme in the literature. At this control, the damping force of the actuator is given proportionally to the vertical velocity of the upper mass. However, most previous researches have found the proportional gain by the trial and error method. In this chapter, a simplified skyhook controller with its control gain determined analytically is proposed for the vibration control of the aircraft landing gear system. In general, a landing gear system featuring the controllable damper can be constructed by a simplified two rigid body model, where one rigid body is the aircraft and the other is the landing gear. Its schematic diagram is shown in the first column of Table 1, where the effect of the tire deflection is neglected. Then, the dynamic governing equation of the landing gear system can be expressed as

\[
\begin{align*}
\dot{m}_1 x &= -k_1 (x - y) - F_d + L_f - m_1 g \\
\dot{m}_2 y &= -k_2 (y - z) + F_d - m_2 g
\end{align*}
\]  

(2)

where \( m_1 \) is the mass of the aircraft body, \( m_2 \) is the mass of the landing gear, \( x \) is the displacement of the aircraft body, \( y \) is the displacement of the landing gear, \( z \) is the surface profile amplitude, \( k_1 \) is the linearized stiffness coefficient of the additional coil spring, \( k_2 \) is the linearized stiffness coefficient of the tire, \( F_d \) is the damping force of the active damper, and \( g \) is the gravitational acceleration. Also, the lift force can be described as \( L_f = [1.2 - 0.9 \tanh(3t)](m_1 + m_2)g \), where \( t \geq 0 \) is the time in seconds. To obtain the vibration transmissibility of the upper mass, linearizing (2) and rearranging in the Laplace domain yield

\[
[m_1 m_2 s^4 + (k_1 m_1 + k_2 m_2) s^2 + k_1 k_2] s^2 + (m_2 s^2 + k_2) \delta x(s) + (m_2 s^2 + k_2) \delta F_d(s) = k_1 k_2 \delta z(s)
\]

(3)

where \( s \) is the Laplace variable, and \( \delta \) represents the variation of the corresponding variable at the static equilibrium state. Similarly, the vibration transmissibility of the 2-dof skyhook model shown in the second column of Table 1 can be expressed as

\[
[m_1 m_2 s^4 + c m_2 s^3 + (k_1 m_1 + k_2 m_2 + k_2 m_2) s^2 + c(k_1 + k_2) s + k_1 k_2] \delta x(s) = k_1 k_2 \delta z(s)
\]

(4)

Then, through the comparison between (3) and (4), the desired damping force can be obtained as follows

\[
F_d(s) = c [1 + H(s)] s x(s)
\]

(5)

where \( H(s) = k_1 l/(m_2 s^2 + k_2) \). With the above control input, the 2-dof general model has the identical dynamic behavior to the 2-dof skyhook model. For the simple implementation of the controller (5), the following assumption is introduced; the mass of the lower body is much smaller than that of the upper body, and therefore its effect is negligible. Then, the resultant damping force, which is called the simplified skyhook controller in this work, is given by

\[
F_d(s) = F_d(s) \bigg|_{m_2 = 0} = c [1 + H(0)] s x(s)
\]

(6)

With the aid of the above assumption, the 2-dof skyhook model can be simplified to the 1-dof skyhook model with two coil spring serially connecting to each other and the ideal skyhook damper, as shown in the third column of Table 1. Then, the skyhook damping coefficient can be analytically determined as \( c = 2 \zeta \sqrt{m_1 k} \), where \( k = k_1 k_2 / (k_1 + k_2) \). On the other hand, the validity of the introduced assumption can be investigated in the frequency domain, as follows.
\[ H(s)/H(0) = 1/(\dot{s}^2 + 1) \]  
(7)

where \( \dot{s} = s/\sqrt{k_2/m_2} \). Fig. 3 shows the effect of the lower mass on the vibration control performance of the simplified skyhook controller. From this figure, it can be seen that the simplified skyhook controller (6) give the similar control performance to the ideal one (5) within the frequency bandwidth less than the natural frequency of the landing gear \( \omega_2 = \sqrt{k_2/m_2} \). In real situation, the natural frequency of the aircraft body \( \omega_3 = \sqrt{k_1/m_1} \) is much smaller than that of the landing gear. In other words, the lower mass has little effect on the performance of the skyhook controller intended to attenuate the vibration of the aircraft body. Therefore, it is concluded that the simplified skyhook controller is useful for the landing gear system, in terms of the practicality and implementability of the algorithm rather than its perfection.

**Table 1. Schematic diagrams of the vibration systems**

| 2-dof general model | 2-dof skyhook model | Simplified model | 1-dof skyhook model |
|---------------------|---------------------|------------------|---------------------|

![Fig. 3. Relative transmissiblity](image)

4. Simulation result

The vibration control performance of the landing gear system featuring the MR damper is investigated through a series of simulations. The specifications of the commercial aircraft ‘Boeing 707’ are used for simulations; \( m_1 = 68,210 \) kg, \( m_2 = 1,140 \) kg, \( k_1 = 11,250,000 \) N/m, \( k_2 = 8,461,000 \) N/m, and \( c = 105,400 \) Ns/m [7]. The sink velocity of the aircraft is set to 0.76 m/s, which can be considered as the practical condition that the aircraft is landing softly [8]. Also, the feasibility of the proposed skyhook controller is investigated through the comparison with the Min-Max controller which is a kind of bang-bang controls. Considering the damping force of the passive-type damper in the real aircraft, the capacity of the MR damper is scaled by 18.5. The input coil current is limited to 5.0 A.

On the other hand, the simplified skyhook controller (6) is designed in the active actuator manner. However, since the MR damper is a semi-active actuator, the following semi-active condition should be superposed on the input coil current:

\[ I = \begin{cases} 1 & \text{for } \ddot{x} (\ddot{x} - \ddot{y}) \geq 0 \\ 0 & \text{for } \ddot{x} (\ddot{x} - \ddot{y}) < 0 \end{cases} \]  
(8)

This semi-active condition ensures the increase in energy dissipation of the MR landing gear system. It is well known that semi-active dampers can attenuate vibrations as well as active dampers without additional cost and complication required in the active damper system.

Fig. 4 shows the vibration control performance of the landing gear system according to the controller. It can be seen that compared with the passive damper, the MR damper gives better performances in attenuating the displacements. Also, the accelerations of the upper mass are most significantly reduced by employing the simplified skyhook controller to the MR landing gear system. The prime purpose of adopting the skyhook controller is to improve the ride comfort of the aircraft by minimizing the acceleration of the upper mass. Since this strategy does not pay attention to the vibration of the lower mass, the maneuverability of the vehicle might be deteriorated due to excessive vibrations of the lower
mass. Obviously, the ride quality and the drive stability are two conflicting criteria in the vibration control of the landing gear system. For the reason, the use of the MR damper increases the acceleration of the lower mass. Nevertheless, the skyhook controller gives better performance in attenuating the acceleration of the lower mass than the Min-Max controller. Also, it is notable that the skyhook controller requires less energy than the Min-Max controller. Therefore, it is concluded that the MR damper featuring the simplified skyhook controller is quite effective to improve the vibration control performance of the aircraft landing gear system in the landing phase.

Fig. 4. Control performance of the landing gear system according to the controllers

Fig. 5. Control performance of the landing gear system according to the time constants of the MR damper

Fig. 5 shows the vibration control performance according to the characteristic time constant of the MR damper featuring the simplified skyhook controller. When the time constant is small, the displacements of the upper and lower masses as well as the acceleration of the upper mass are reduced. However, its opposite is true in the case of the acceleration of the lower mass. In other words, fast response of the MR damper can deteriorate the vehicle stability of the aircraft. If a low-pass filter is employed to adjust the response time of the MR damper, then the relative importance of the ride comport versus the vehicle stability can be controlled considering the operating condition of the landing gear system. Among many other variations of the skyhook controller proposed to overcome the drawback of the original skyhook controller, the groundhook and hybrid controller have been paid much attention in the literature. The groundhook controller is similar to the skyhook controller, except that it is intended to improve the vehicle stability by minimizing the acceleration of the lower mass. The hybrid controller combines the concept of skyhook and groundhook controller to take advantage of the benefits of both, where a weighing factor is employed to determine the relative ratio between both controllers. Note that the time constant of the MR damper plays a similar role to this weighting factor. Therefore, the time constant of the MR damper can be considered as one control gain which is controllable with the low-pass filter.

5. Experimental results: HILS
In this chapter, HILS (Hardware-in-the-loop simulation) is used to evaluate the vibration control performance of the MR landing gear system. In general, the theoretical analysis is rather difficult to precisely predict the performance of the system, since many real situations are often neglected or approximated during its modeling and emulating. Therefore, the HILS method is frequently adopted in order to overcome this limitation. Especially, this method has major advantages such as easy modification of system parameters and wide application of operating conditions [9][10]. Fig. 6 shows the apparatus of HILS constructed in this study. The MR damper is excited by a hydraulic cylinder at the given relative displacement, which is calculated from the difference in displacement between the sprung and unsprung masses. Here, a proportional hydraulic valve is employed for the hydraulic cylinder to track its desired displacement even in the presence of the varying damping force of the MR damper. The coil current input to the MR damper can be adjusted by the current amplifier which is controlled by the personal computer with the DC converter. During operations, the damping force and displacement of the MR damper are measured by a loadcell mounted on the base frame and a linear potentiometer connected with the hydraulic cylinder, respectively. The sensor signals are transmitted to the personal computer via the AD converter with its sampling frequency of 1000 Hz. Fig. 7 shows the results of the HILS conducted under the same conditions to the simulations. It can be seen that the HILS results are well agreed with the simulation ones. However, there is little difference in the displacement. It is considered that it results from several assumptions introduced in the modeling of the MR damper. In all the simulations, the time constant of the MR damper is assumed to be constant, but it can be varied according to the excitation velocity or current-induced condition as mentioned above. Also, MR fluid behavior actually exhibits some departures from the Bingham model due to its viscoelastic and hysteresis property. Nevertheless, we can conclude that the proposed MR damper featuring the simplified skyhook controller is effective for attenuating undesired vibration of the landing gear system, with the viewpoint of the practicability and implementability rather than the perfection.

6. Conclusions
With the help of the simplified skyhook controller proposed in this work, the applicability of the MR damper constructed as Pusan National University to the aircraft landing gear system was investigated through a series of simulations. The obtained results can be summarized as follows; (i) the MR damper featuring the simplified skyhook controller is effective for attenuating undesired vibration of the landing gear system. (ii) The control gain of the skyhook controller can be analytically obtained with the assumption that the mass of the landing gear system is much smaller than the mass of the aircraft body and therefore is negligible. (iii) The time constant of the MR damper is one important factor determining the vibration control performance of the MR landing gear system, and plays a role similar to the weighting factor in the hybrid controller combining the function of the skyhook and groundhook controllers. (iv) Using the HILS method, it is verified that the proposed MR damper featuring the simplified skyhook controller is effective for attenuating undesired vibration of the landing gear system.

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