

Assessment of Ride Comfort and Braking Performance Using Energy-Harvesting Shock Absorber

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ABSTRACT

Conventional viscous shock absorbers, in parallel with suspension springs, passively dissipate the excitation energy from road irregularity into heat waste, to reduce the transferred vibration which causes the discomfort of passengers. Energy-harvesting shock absorbers, which have the potential of conversion of kinetic energy into electric power, have been proposed as semi-active suspension to achieve better balance between the energy consumption and suspension performance. Because of the high energy density of the rotary shock absorber, a rotational energy-harvesting shock absorber with mechanical motion rectifier (MMR) is used in this paper. This paper presents the assessment of vehicle dynamic performance with the proposed energy-harvesting shock absorber in braking process. Moreover, a PI controller is proposed to attenuate the negative effect due to the pitch motion. The simulation result implies that the active damping control based on the MMR shock absorber can optimize the performance indices for both the suspension and braking system during the braking process. An average total power amount of 29.4W can be harvested with deceleration level of 0.4g on the Class C road.

CITATION: Yu, L., Huo, S., Xuan, W., and Zuo, L., "Assessment of Ride Comfort and Braking Performance Using Energy-Harvesting Shock Absorber," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(2):2015, doi:10.4271/2015-01-0649.

INTRODUCTION

The vehicle suspension system attenuates or even vanishes the vibration excited from the road irregularity, offering ride comfort to the passengers and at the same time trying to remain steady dynamic vertical load for the vehicle dynamic control of other systems, like ABS, ESP, AFS, etc. [1, 2, 3]. To expand the application potential of the suspension system, active suspension has been studied for decades [4, 5, 6]. Despite the control flexibility and the good performance of the active suspension, the large amount of energy consumption [7] is the main limitation for widely applications. Since energy saving becomes global-wide issue, this problem becomes more intense, arising the trade-off between the vehicle fuel consumption and the suspension high performance. Thus, the semi-active suspension or energy-recovery actuator, which can achieve the similar performance of active suspension and meanwhile consume far less energy, or even can harvest energy from the suspension vibration, arouses the focus of studies.

The traditional damper dissipates the vibration energy by transferring it into heat waste, which cannot be reused. Many methods have been proposed to develop the use or harvest of this potential kinetic energy. Karnapp [8] proved the feasibility of using permanent magnet linear motor as active damper by changing the external resistance. Hsu [9]

indicated the recoverable power amount dependent on driving and road conditions and control algorithm. He also pointed out that the active suspension had few effect on the suspension performance while harvesting energy. Suda et al. [10] developed a self-powered active suspension with two linear DC motors serving as an actuator and an energy regenerative damper respectively. The experiment proved that the proposed hybrid suspension improved the isolation performance especially at high frequency. Kawamoto et al. [11] used the ball screw mechanism based electro-mechanical actuator as active suspension, which consumed energy around sprung mass vibration mode below about 2 Hz, and regenerated energy around unsprung mass vibration mode above about 2 Hz. The power of 33W was regenerated according to the experimental result. Gysen et al. [12] developed a direct-drive brushless tubular permanent-magnet actuator to reduce the power consumption for active suspension, only 47W per wheel for comfort objective and 37W per wheel for tire load objective. Chen [13] proposed a self-sensing magnetorheological (MR) damper containing the functions of energy harvesting, controllable damping and vehicle dynamic sensing.

Zuo et al. [14] assessed the compatibility among energy harvesting, ride comfort and road handling. The analysis revealed that road roughness, tire stiffness and vehicle speed exert great influence on the harvesting power potential. Considering that rotary shock absorber

which transferred linear suspension vibration motion to DC generator rotatory motion, had advantage of harvesting more power and obtaining large damping coefficient, Li et al. [15] designed an energy-harvesting shock absorber using rack-pinion mechanism. An average power of 19.2 W can be harvested at 48km/h using such shock absorbers. And later Li introduced the mechanical motion rectifier [16] to improve the performance and reliability of the previous shock absorber, which is used in this paper.

Although the dynamic performance in vehicle and the energy-harvesting potential of the proposed energy-harvesting shock absorber have been evaluated under constant speed by the prior work [17], the performance under different driving conditions and the interaction with vehicle dynamics still remain unknown. Particularly this paper focuses on the pitch motion improvement to control vehicle attitude and provide better ride comfort during the braking process, using the proposed MMR shock absorber. On the other hand, the energy harvesting power due to the impact on the suspension during braking process would be evaluated as well. Hence, this paper will firstly present the vehicle half car model with MMR based energy-harvesting shock absorbers. Then, the active damping control will be developed with introducing a PI controller to adjust the external resistance. At last, the simulation will be carried out to assess the ride comfort, effects on braking process and the average harvesting power.

VEHICLE MATHEMATICAL MODEL

The main purpose of this paper is to evaluate the energy harvesting feasibility and the effect exerting on the braking performance by the innovative energy-harvesting shock absorber during the braking process. Hence, the equivalent model of the energy-harvesting absorber is introduced in this section, especially the MMR mechanism and the damping coefficient variable characteristics. Regarding ride comfort during the braking process, sprung mass acceleration is not the only concerned index, but also the pitch acceleration matters. Therefore, the half car model considering suspension and braking system is suitable for the simulation and comparison. Also the tire-road interface model and the road excitation model are included, which are of significance to describe the empirical tire-road interaction and to represent the road unevenness respectively.

Energy-Harvesting Shock Absorber Model

The MMR based energy-harvesting shock absorber is implemented between the chassis and the wheels, to provide damping force by the generator electromagnetic force, i.e. to dissipate vibration by transferring kinetic energy to electric energy. Figure 1 shows the design of the absorber [16].

The MMR mechanism enables the unidirectional rotation of generator rotor from the bidirectional suspension vibration input. With the rack irregular reciprocating movement input, only one of the two roller clutches, which are mounted between the big bevel gears and the shaft, will be engaged, resulting in the small coupled bevel gear rotating in

one direction all the time. This kind of design eliminates the impact between the mechanical components caused by backlash when direction changing. Thus, the reliability would be improved.

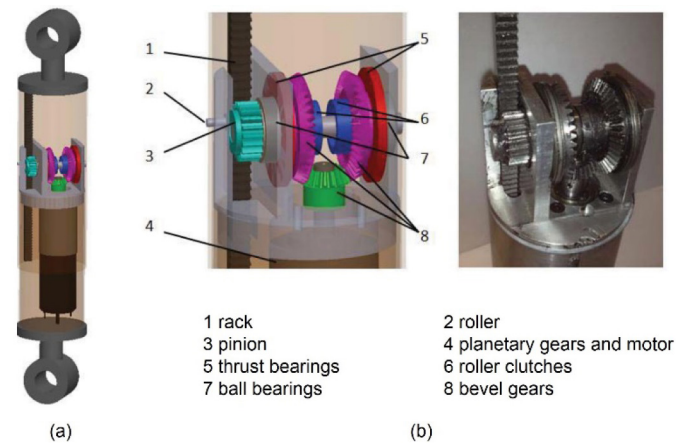


Figure 1. 3D model and actual prototype of the energy-harvesting shock absorber, (a) overall view, (b) inner structure.

The MMR based energy-harvesting shock absorber can be simplified for modeling as shown in Figure 2 [16].

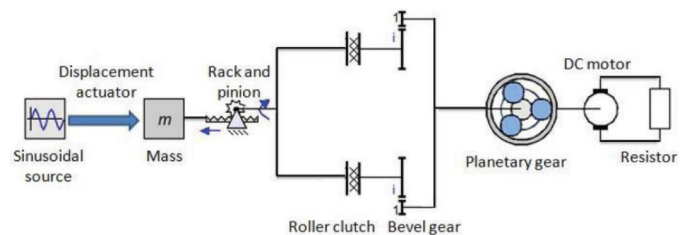


Figure 2. Simplified schematic of the MMR based energy-harvesting shock absorber.

For the DC generator, we can get the physical equations

$$U_{ef} = k_e \omega_g = k_e v_r \frac{k_h k_b}{r} = k_e v_r n, \quad (1)$$

$$i = \frac{U_{ef}}{R}, \quad (2)$$

where U_{ef} is the generator output voltage, k_e is the back electromotive voltage constant, ω_g is the rotor speed and v_r is the rack reciprocating speed. k_h , k_b are the transmission ratios of the planetary gear head and bevel gears and r is the pinion radius, which altogether form the total transmission ratio n . i is the electric current and R is the total resistance, including the internal resistance R_i and external resistance R_e . The inductance in the circuit is neglected. Hence, the output power can be written as

$$P_{out} = \frac{U_{ef}^2}{R_e}.$$

The MMR mechanism introduces its non-linear behavior into the system as well. When the input shaft rotatory speed is smaller than that of the output shaft, the roller clutches would disengage so that no force is transmitted. When the roller clutch is engaged, the damping force can be equivalent to the following:

$$F_d = -cv_r - m_e a_r, \quad (3)$$

where $c = \frac{k_e k_t k_h^2 k_b^2}{R r^2}$ is the equivalent damping coefficient, and k_t is the generator torque constant. m_e is the equivalent mass of all moments of inertias of the motor J_m , planetary gear head J_h and the large bevel gear J_b . When J_h and J_b can be neglected compared to J_m , the equivalent mass can be derived as the following:

$$m_e = \frac{J_m k_h^2 k_b^2 + J_h k_b^2 + J_b}{r^2} \approx \frac{J_m k_h^2 k_b^2}{r^2}. \quad (4)$$

When the roller clutch is disengaged, the damping force decreases to zero. The output shaft speed will decrease due to friction until both shafts speed equal, i.e. the roller clutch reengages. The dynamic equation of output shaft during disengagement is given as the following:

$$\dot{\omega}_{out} = -\tau \omega_{out}, \quad (5)$$

$$\omega_{out} = \omega_{out0} e^{-\tau t}, \quad (6)$$

where ω_{out0} and t denote the output shaft speed and the time when disengagement happens. τ is the time coefficient dependent on the equivalent coefficients, which is given as the following:

$$\tau = \frac{c}{m_e} = \frac{k_e k_t k_h^2 k_b^2}{R r^2 m_e} \approx \frac{k_e k_t}{R J_m}. \quad (7)$$

The MMR based shock absorber enables the active damping control by changing the external resistance. Figure 3 shows the force-displacement characteristics of the absorber under sinusoid displacement input. When the displacement is close to the amplitude, i.e. the velocity becomes very small but the acceleration becomes relative large, the inertia part of the absorber is dominant and exerts negative effect on the damping part of Eq. (3). From the physical point of view, the output shaft speed is faster than that of the input shaft. Thus, the roller clutch disengages and the output force falls to zero. On the other hand, when the movement starts to reverse after the end of expansion or compression, the accumulated inertia power releasing direction matches the velocity direction, resulting in the output force jumping to a certain value, which equals the maximum inertia force. Besides, the external resistance affects not only the damping coefficient, but also the disengagement duration since it also changes the relative relationship between equivalent damping coefficient and equivalent mass.

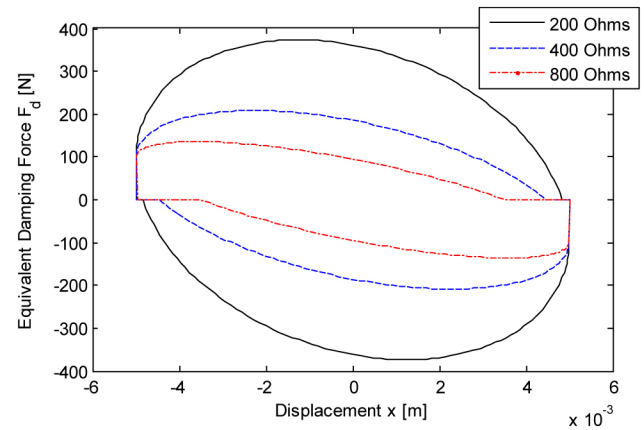


Figure 3. Force vs Displacement for MMR based shock absorber under 5 Hz sinusoid displacement input with $m_e = 20$ kg and different total resistances.

Half Car Suspension and Braking System Model

Because of concentration on vertical and longitudinal vehicle dynamics, lateral characteristics and the difference between the left and right side of the vehicle are neglected in this paper. The vehicle body is considered as rigid and with sprung mass. The suspension is equivalent to the tire side with spring stiffness and damping coefficient. The tire and related components are equivalent to unsprung mass and tire spring stiffness and damping coefficient. A 7-DOF half car model considering suspension and braking system is shown in Figure 4.

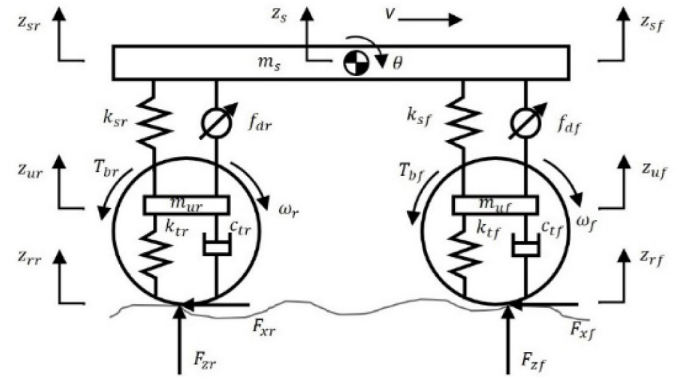


Figure 4. Schematic of half car model.

Suspension Mathematical Model

In the suspension model, the damping force generated by the MMR based energy-harvesting shock absorber is modeled as an active damping actuator mounted in paralleled with the passive spring. The equations of the vertical and pitch motion for the sprung mass can be written as the following:

$$m_s \ddot{z}_s = f_f + f_r, \quad (8)$$

$$I_p \ddot{\theta} = f_r l_r - f_f l_f + F_{xf}(h + z_s - z_{rf}) + F_{xr}(h + z_s - z_{rr}), \quad (9)$$

with