

Handling Analysis of a Vehicle Fitted with Roll-Plane Hydraulically Interconnected Suspension Using Motion-Mode Energy Method

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ABSTRACT

This paper employs the motion-mode energy method (MEM) to investigate the effects of a roll-plane hydraulically interconnected suspension (HIS) system on vehicle body-wheel motion-mode energy distribution. A roll-plane HIS system can directly provide stiffness and damping to vehicle roll motion-mode, in addition to spring and shock absorbers in each wheel station. A four degree-of-freedom (DOF) roll-plane half-car model is employed for this study, which contains four body-wheel motion-modes, including body bounce mode, body roll mode, wheel bounce mode and wheel roll mode. For a half-car model, its dynamic energy contained in the relative motions between its body and wheels is a sum of the energy of these four motion-modes. Numerical examples and full-car experiments are used to illustrate the concept of the effects of HIS on motion-mode energy distribution. The obtained simulation results show that the installed HIS system is able to reduce the energy level in the body-dominated roll motion-mode, and it has negligible effect in the bounce mode. Experimental results match the simulation results well.

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INTRODUCTION

Vehicle rollovers are dangerous and potentially fatal events. America, Australia, Britain, Canada and other western countries have attached great importance to the severity and frequency of rollover accident. The statistics from National Highway Traffic Safety Administration (NHTSA) show that, in all of the traffic accidents, the harmful level of rollover accident ranks the second place [1]. The reason of vehicle rollover is very complex, many factors can affect the motions of a car. The suspension system plays an important role in reducing the probability of rollover accidents [2]. A good suspension system can largely reduce the probability of vehicle rollover accident.

Recently, a passive roll-plane hydraulically interconnected suspension system (HIS) has been proposed to provide the essential roll stiffness to a vehicle to replace anti-roll bars [3]. The proposed HIS is one of the passive hydraulically

interconnected suspension systems, and is composed of hydraulic cylinders, accumulators, connected through hoses and pipes [4].

VEHICLE DYNAMIC MODELING

Model Description

In order to retain simplicity, while still account for fluid interconnections between wheel stations, a bilateral symmetry 4-DOF half-car model, as shown in [Figure 1](#), is employed in the investigation. The vehicle model consists of sprung and unsprung mass, connected by spring and dampers. The hydraulic system is coupled kinetically to the mechanical system through double-acting hydraulic cylinders [4], shown in [Figure 1\(b\)](#). The parameter of the 4-DOF vehicle dynamic model are presented in [Table 1](#).



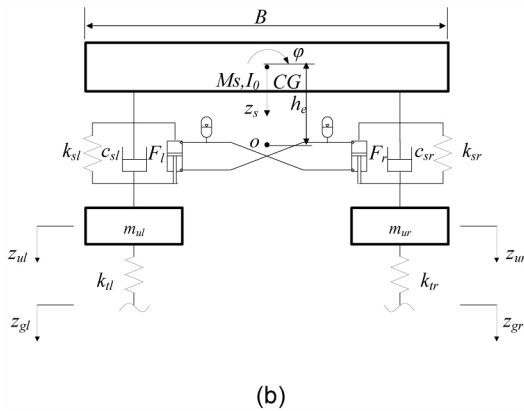
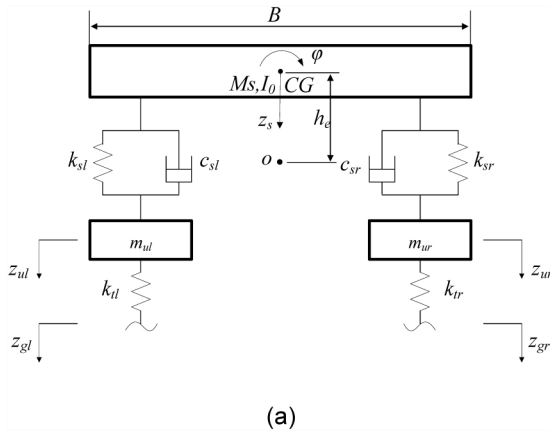


Figure 1. Schematic drawing of the half-car model: (a) fitted with a conventional independent suspension; (b) fitted with an HIS.

Table 1. Parameter values of the 4-DOF vehicle dynamic model.

Sprung mass	750 kg
Unsprung mass(subscript l=left, r=right)	35 kg
Sprung mass moment of inertia about roll axis	320 kg m ²
Track width	1.5 m
Suspension spring stiffness (subscript l=left, r=ri)	20 kNm ⁻¹
Tire spring stiffness (subscript l=left, r=right)	200 kNm ⁻¹
Suspension damping coefficient(subscript l=left, r=right)	1500 Nsm ⁻¹
Distance between centre of gravity and vehicle roll centre	0.3 m
he height of the upper and lower chambers at the initial time	0.20 m
Pre-charge gas volume of accumulator	1.607×10 ⁻⁴ m ³
Pre-charge gas pressure of accumulator	1.5 Mpa
Density of the hydraulic oil	870 kg m ⁻³
Bulk modulus of the hydraulic oil	1400 Mpa
Pipe length from accumulators to cylinders	1.0 m
Pipe cross area	3.96×5 m ²

Half-Car Model

Compared to the traditional independent suspensions, this novel suspension is equipped with an extra interconnected hydraulic system. The differential equations for the heave and roll motions of the vehicle body, and the heave motions of the two wheels are given as follows,

$$\begin{aligned}
 m_s \ddot{z}_s &= -k_{sl}(z_s - \phi B/2 - z_{ul}) - c_{sl}(\dot{z}_s - \dot{\phi} B/2 - \dot{z}_{ul}) \\
 &\quad - k_{sr}(z_s + \phi B/2 - z_{ur}) - c_{sr}(\dot{z}_s + \dot{\phi} B/2 - \dot{z}_{ur}) + F_l + F_r \\
 I_0 \ddot{\phi} &= k_{sl}(z_s - \phi B/2 - z_{ul})B/2 + c_{sl}(\dot{z}_s - \dot{\phi} B/2 - \dot{z}_{ul})B/2 \\
 &\quad - k_{sr}(z_s + \phi B/2 - z_{ur})B/2 - c_{sr}(\dot{z}_s + \dot{\phi} B/2 - \dot{z}_{ur})B/2 \\
 &\quad + m_s v^2 h_e / R - F_l B/2 + F_r B/2 \\
 m_{ul} \ddot{z}_{ul} &= k_{sl}(z_s - \phi B/2 - z_{ul}) + c_{sl}(\dot{z}_s - \dot{\phi} B/2 - \dot{z}_{ul}) \\
 &\quad - k_{tl}(z_{ul} - z_{gl}) - F_l \\
 m_{ur} \ddot{z}_{ur} &= k_{sr}(z_s + \phi B/2 - z_{ur}) + c_{sr}(\dot{z}_s + \dot{\phi} B/2 - \dot{z}_{ur}) \\
 &\quad - k_{tr}(z_{ur} - z_{gr}) - F_r.
 \end{aligned} \tag{1}$$

Assuming that the roll angle ϕ is small enough, the following linear approximations are applied

$$\begin{aligned}
 z_{sl} &= -z_s + \phi B/2 \\
 z_{sr} &= -z_s - \phi B/2,
 \end{aligned} \tag{2}$$

where z_{sl} , z_{sr} represent the vertical displacement of the sprung mass at two suspension contact points, namely, left (sl) and right (sr). Equation (2) can be summarized as

$$x_s = L^T x_m, \tag{3}$$

where

$$x_m = [z_s \quad \phi]^T, \quad x_s = [z_{sl} \quad z_{sr}]^T \quad \text{and} \quad L = \begin{bmatrix} -1 & -1 \\ B/2 & -B/2 \end{bmatrix}.$$

Unsprung mass state vector is $x_u = [z_{ul} \quad z_{ur}]^T$; ground input vector is $x_g = [z_{gl} \quad z_{gr}]^T$. By defining

$$\begin{aligned}
 Z &= [x_m \quad x_u]^T, \quad M_s = \begin{bmatrix} m_s & 0 \\ 0 & I_0 \end{bmatrix}, \quad M_u = \begin{bmatrix} m_{ul} & 0 \\ 0 & m_{ur} \end{bmatrix}, \\
 K_s &= \begin{bmatrix} k_{sl} & 0 \\ 0 & k_{sr} \end{bmatrix}, \quad K_t = \begin{bmatrix} k_{tl} & 0 \\ 0 & k_{tr} \end{bmatrix} \quad \text{and} \quad C_s = \begin{bmatrix} c_{sl} & 0 \\ 0 & c_{sr} \end{bmatrix},
 \end{aligned}$$

Equation (1) then can be further written as

$$M \ddot{Z} + C \dot{Z} + KZ = F, \tag{4}$$

where

$$M = \begin{bmatrix} M_s & 0 \\ 0 & M_u \end{bmatrix}, \quad C = \begin{bmatrix} LC_s L^T & -LC_s \\ -C_s L^T & C_s \end{bmatrix},$$

$$K = \begin{bmatrix} LK_s L^T & -LK_s \\ -K_s L^T & K_s + K_t \end{bmatrix} \text{ and}$$

$$F = \begin{bmatrix} 0 \\ K_t \end{bmatrix} x_g + \begin{bmatrix} -L \\ -I \end{bmatrix} \begin{bmatrix} F_l \\ F_r \end{bmatrix} + [0 \quad m_s a h_e \quad 0 \quad 0]^T.$$

Hydraulic System

The HIS system consists of four double-acting hydraulic cylinders, two nitrogen-filled diaphragm accumulators, hydraulic lines and hoses [3, 4, 5]. The following model is obtained as shown in Figure 2.

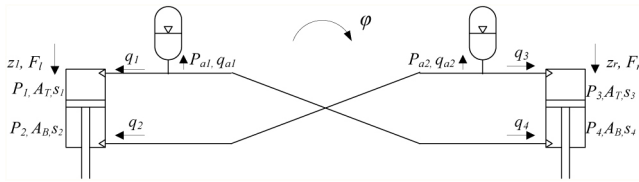


Figure 2. Simplified model of the half-car hydraulically interconnected suspension system.

The oil inside the cylinder chambers is considered compressible. By using the Bulk modulus, we obtain the following equations:

$$\dot{p}_1 = \frac{E}{s_1 A_T} (\dot{z}_l A_T + q_1), \quad \dot{p}_2 = \frac{E}{s_2 A_B} (-\dot{z}_l A_B + q_2),$$

$$\dot{p}_3 = \frac{E}{s_3 A_T} (\dot{z}_r A_T + q_3), \quad \dot{p}_4 = \frac{E}{s_4 A_B} (-\dot{z}_r A_B + q_4),$$

(5)

where E is the bulk modulus of the hydraulic oil; A_T and A_B represent the cross area of the hydraulic cylinder; z_l, z_r represent the vertical displacement of the hydraulic cylinder; $s_i (i=1,2,...,4)$ represent the height of the hydraulic cylinder, which are given by

$$z_l = z_s - \phi B / 2 - z_{ul}, \quad z_r = z_s + \phi B / 2 - z_{ur},$$

$$\dot{z}_l = \dot{z}_s - \dot{\phi} B / 2 - \dot{z}_{ul}, \quad \dot{z}_r = \dot{z}_s + \dot{\phi} B / 2 - \dot{z}_{ur},$$

and

$$s_1 = s_0 - z_l, \quad s_2 = s_0 + z_l,$$

$$s_3 = s_0 - z_r, \quad s_4 = s_0 + z_r.$$

(6)

The oil inside the pipe is considered incompressible. By using the Newton's second law $m\ddot{x} = F$, we obtain the following equation:

$$\dot{q}_1 \frac{\rho L_p}{A_p} = p_{a1} - p_1, \quad \dot{q}_2 \frac{\rho L_p}{A_p} = p_{a2} - p_2,$$

$$\dot{q}_3 \frac{\rho L_p}{A_p} = p_{a2} - p_3, \quad \dot{q}_4 \frac{\rho L_p}{A_p} = p_{a1} - p_4,$$

(7)

where ρ is the fluid density; L_p and A_p are the length and the pipe area, respectively.

The pressure and volume at any time in the accumulator- p_{ai} and V_{ai} ($i=1, 2$), respectively-are related to the pre-charge values, p_p and V_p , as follows:

$$p_p v_p^\gamma = p_{a1} v_{a1}^\gamma = p_{a2} v_{a2}^\gamma = \text{constant},$$

(8)

where γ is the ratio of specific heat for the gas. When the car is in stable driving conditions, the accumulator can be considered working in the isothermal conditions, then $\gamma = 0$; otherwise $\gamma = 1.4$. Then we can obtain the following equation:

$$\dot{p}_{a1} = \frac{\gamma q_{a1} p_{a1}}{V_p} \left(\frac{p_{a1}}{p_p} \right)^{1/\gamma} \quad \text{and} \quad \dot{p}_{a2} = \frac{\gamma q_{a2} p_{a2}}{V_p} \left(\frac{p_{a2}}{p_p} \right)^{1/\gamma}$$

(9)

where

$$q_{a1} = -q_1 - q_4 \quad \text{and} \quad q_{a2} = -q_2 - q_3.$$

Thus, the force supplied by the HIS system to the vehicle can be written as

$$F_l = -p_1 A_T + p_2 A_B \quad \text{and} \quad F_r = -p_3 A_T + p_4 A_B.$$

(10)

Integrated Model with HIS

Now we can assemble the HIS to the half-car model. By defining,

$$Y = \begin{bmatrix} z_s & \phi & z_{ul} & z_{ur} & \dot{z}_s & \dot{\phi} & \dot{z}_{ul} & \dot{z}_{ur} & p_1 & p_2 & p_3 & p_4 & \dots \\ p_{a1} & p_{a2} & q_1 & q_2 & q_3 & q_4 \end{bmatrix}^T$$

Equation (4, 5, 7, 9) can then be written in the matrix state-space form,

$$\dot{Y} = \tilde{A}Y + \tilde{F},$$

(11)