

Development of A New Model for Roll-Plane Active Hydraulically Interconnected Suspension

Sangzhi Zhu

University of Technology Sydney

Haiping Du

University of Wollongong

Nong Zhang and Lifu Wang

University of Technology Sydney

ABSTRACT

In this paper, a more sophisticated mathematical linear model for a roll-plane active hydraulically interconnected suspension (HIS) system was developed. Model parameters tuning were then carried out, which resulted in a model that is capable of producing rather accurate estimation of the system, with significant improvements over models built previously. For the verification of the new model, two simulations and corresponding experiments are conducted. Data comparisons between the simulations and experiments show high consistent responses of the model and the real system, which validated the robustness and accuracy of the new mathematical model. In this process, the characteristics of the pressure response and the rise time inside the actuators have been revealed due to the presence of the flow.

CITATION: Zhu, S., Du, H., Zhang, N., and Wang, L., "Development of A New Model for Roll-Plane Active Hydraulically Interconnected Suspension," *SAE Int. J. Passeng. Cars - Mech. Syst.* 7(2):2014, doi:10.4271/2014-01-0053.

1. INTRODUCTION

Fatal crashes caused by vehicle unstable motion, especially rollover, have threatened public health and safety. Even only about 3% of all crashes are rollovers, but they account for 33% of crash related death [1]. The vehicles, such as Sport Utility Vehicles (SUVs), off-road vehicles, trucks, vans, pickups, are more vulnerable in rollover accident because of their lower rollover threshold due to higher center of gravity (CG). Compared with other vehicle types, SUVs experienced the highest rollover rate: 36% in fatal crashes [2].

Modern suspension systems consist of a combination of restoring and damping media [3]. Generally, the vehicle roll stability can be improved by increasing the suspension roll stiffness and damping. A vehicle with a relatively stiff suspension is likely to possess good handling stability but poor ride comfort, and vice versa [4]. Therefore a compromise is required between handling and ride comfort in vehicle design.

The advent and advancement of active suspension provides another option for customers of luxury vehicles. Active suspensions directly supply desired control forces to the

vehicle chassis according to the road condition and vehicle motions, therefore achieving superior performance. But the active suspensions generally adopt independently controlled actuators to control vehicle motions that are in different frequency range and coupled together. This kind of unfocused control is too general to be efficient, and brings high design cost and energy consumption, and because of this, their applications are still limited to best-in-class vehicles. The main obstacle for commercialization of such systems is the significant power requirement [5].

An active HIS named Demand Dependent Active Suspension System (DDAS) was proposed in 2009 [6] to improve the above mentioned issues for active suspensions. Then an early version of this active HIS system was investigated and a simple test rig were designed in [7]. The objective of the proposed active HIS is to actively control the roll motion of the vehicle body before it reaches rollover threshold. To achieve this objective, the active HIS system employs four hydraulically interconnected actuators into each wheel station and a control unit comprised of a motor, a bump, a servo valve, a tank and an accumulator etc. is also designed to control these actuators. This design enables the active HIS to exert desired control

forces to stabilize the roll angle of vehicle body under certain circumstances. A mathematical model of the actuator based on the bulk modulus equation of the fluid mass inside the chamber of the actuator has been built in previous work [8]. And the responses of the system under road excitation have also been investigated in experiments with the H^∞ control [9].

In the initial development of this active HIS, noticeable progress has been made. However, due to the non-linear nature of the hydraulic systems used in the HIS system, further performance improvements were unreachable. By looking at the root of the problem, it was found that the mathematical model of the system, which was also the pivot of the model based control system, was not sophisticated enough to handle various factors in the physical system itself, such as fluid flow, pipe friction, etc. In summary, the previous mathematical model of the system was too ideal and therefore, yielded inaccurate estimations of the system's states.

In this paper, further development of the system was explored using an enhanced version of the system's mathematical model which includes the flow. It is followed by the model parameter tuning, in which the results of a locked actuator response in previous test for building the old model have been adopted to be tuned again. In addition, in order to validate the new model, two experiments were conducted with a purpose of comparing with the simulation implemented of the new model using the same inputs. The experiments were conducted on a SUV, which is a typical and venerable vehicle in roll motion.

Moreover, the response of the old mathematical model was compared with the new model to demonstrate the improvement of the new model.

The paper is organized as follows. [Section 2](#) presents the new modeling process. The old modeling was also introduced. In [section 3](#), model verification and active HIS system response are presented. In [section 4](#), the model development has been discussed and the aforementioned two control algorithms have been compared. [Section 5](#) provides the concluding statements.

2. MODELING

This section briefly introduces the old model of the active HIS system and provides details of the proposed new model. Based on the built physical prototype, various parameters of the system obtained from previous tests were used.

2.1 Half-Car Model

To simplify the problem of analyzing the dynamics of the vehicle body roll motion, a half-car model was utilized as shown in [Figure 1](#).

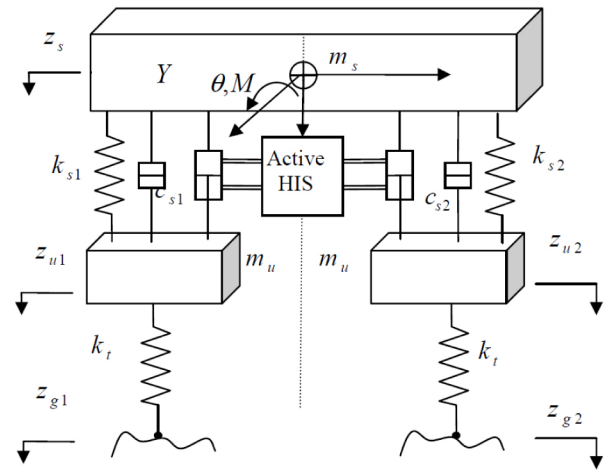


Figure 1. Half-car model integrated with active HIS

To capture the dynamics of the half-car model under actuation of the active HIS system, motion equations have been derived by applying Newton's second law:

$$\begin{aligned}
 (mh^2 + I)\ddot{\theta} &= -2l^2 k_s \theta + \\
 &\quad (Z_{u1} - Z_{u2})k_s - 2l^2 C_s \dot{\theta} + \left(\dot{Z}_{u1} - \dot{Z}_{u2} \right) l C_s + F_1 l - F_2 l \\
 m_s \ddot{Z}_s &= -k_s (Z_s + \theta l - Z_{u1}) - \\
 &\quad C_s \left(\dot{Z}_s + \dot{\theta} l - \dot{Z}_{u1} \right) - k_s (Z_s - \theta l - Z_{u2}) - C_s \left(\dot{Z}_s + \dot{\theta} l - \dot{Z}_{u2} \right) - F_1 - F_2 \\
 m_u \ddot{Z}_{u1} &= -k_t Z_{u1} + k_s (Z_s + \theta l - Z_{u1}) + C_s \left(\dot{Z}_s + \dot{\theta} l - \dot{Z}_{u1} \right) + F_1 + k_t Z_{g1} \\
 m_u \ddot{Z}_{u2} &= -k_t Z_{u2} + k_s (Z_s - \theta l - Z_{u2}) + C_s \left(\dot{Z}_s + \dot{\theta} l - \dot{Z}_{u2} \right) + F_2 + k_t Z_{g2}
 \end{aligned} \tag{1}$$

where

Z_{g1} - road surface input at left wheel

Z_{g2} - road surface input at right wheel

F_1 - force exerted on the car body by the left cylinder

F_2 - force exerted on the car body by the right cylinder

2.2 Active HIS System Model

2.2.1 Old Model Development

The old model was developed by the method of model estimation. The hydraulic actuator was locked in a position so it cannot retract or expand as shown in [Figure 2](#). A sinusoidal pressure control signal with different frequencies was sent to the servo valve and the actuator was charged with working fluid. The schematic of the testing set up is shown in [Figure 3](#). The pressure responses in the chambers of the cylinder were recorded by a pressure transducer and used to develop the old model and to be tuned against for the new model. Tuning the

model against the output of the test will be a major step towards producing a linear model that can sufficiently address the non-linear properties of the active HIS system.

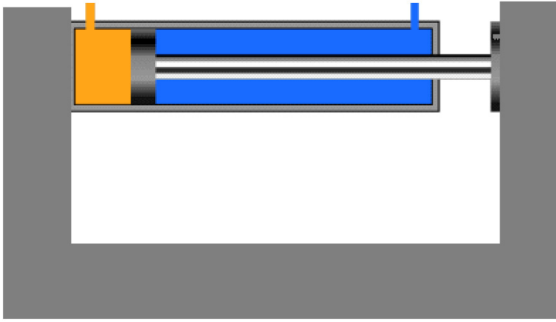


Figure 2. Locked actuator in the experiment

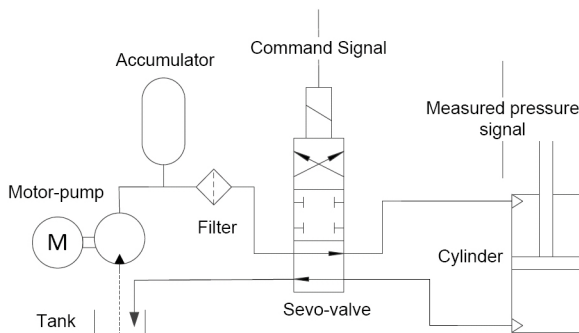


Figure 3. Schematic drawing of the testing setup

Both the command and measured signal are collected by the National Instrument data acquisition board and LabVIEW. The differences of these signals in terms of magnitude and phase delay are calculated from the obtained results and plotted against the frequency, shown in Figure 4.

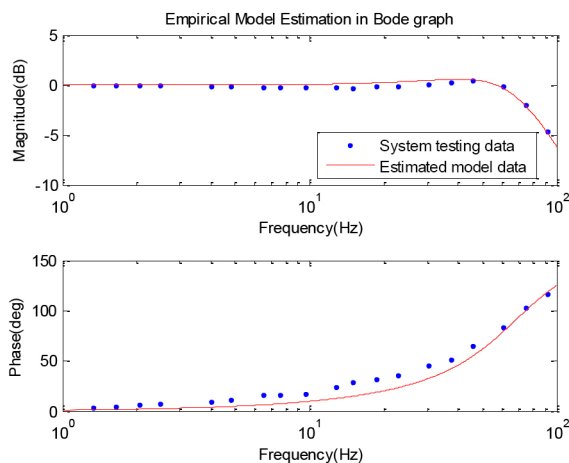


Figure 4. Empirical model estimation in Bode graph

From the plotted Bode graph in Figure 4, a second order system is considered to be sufficient to cover the main dynamic characteristics of the system, shown in Equation (2):

$$\frac{P(s)}{I(s)} = \frac{1}{1 + (2\xi / w_n)s + s^2 / w_n^2} \quad (2)$$

$P(s)$ represents the measured pressure signal, and $I(s)$ is the input command signal. w_n is the natural frequency, ξ is the damping ratio; by curve fitting, the optimized value for w_n and ξ are estimated as 68Hz and 0.57. The estimated system frequency response is also shown in the solid line in Figure 4.

To be consistent with the vehicle modeling from previous work [8], it is necessary to convert the hydraulic system's transfer function into state-space form. The resulting state-space equation and system matrices are presented

$$\begin{aligned} \dot{X}_p &= A_p X_p + B_p \text{inp} \\ Y_p &= C_p X_p + D_p \text{inp} \end{aligned} \quad (3)$$

where

$$A_p = \begin{bmatrix} -487.1 & -356.5 \\ 512 & 0 \end{bmatrix}, \quad B_p = \begin{bmatrix} 16 \\ 0 \end{bmatrix},$$

$$C_p = [0 \quad 22.28], \quad D_p = [0]$$

(4)

inp is the control signal sent to the pressure control unit, and Y_p is the actual pressure in the actuator.

2.2.2 New Model Development

The developed new mathematical model is remaining a linear model so that advanced linear controllers can be designed for the active HIS system.

The new version of the system model will ignore the dynamics of the accumulator and pump to simplify the problem. Further studies of effects of these two components on the system's performance can be carried out by replacing the two pressures P_2 and P_4 in Figure 5 with the appropriate governing equations.

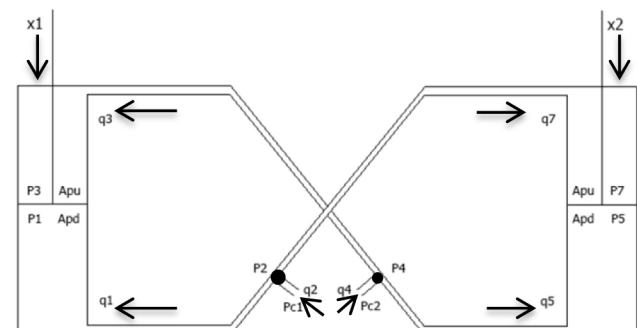


Figure 5. Diagram of the interconnected hydraulic suspension system with flow and input sign convention

As show in Figure 5, $P_1, P_3, P_5, P_7, P_2, P_4$ are pressures in different position of the system; $q_1, q_3, q_5, q_7, q_2, q_4$ are flows; and A_{pu} and A_{pd} are the upside and downside areas of the piston respectively. Pressures at the servo valve inlet of the active HIS system P_2 and P_4 are given by the two equations shown below (note that P_{c1} and P_{c2} are the main control inputs through which the active HIS system will be controlled).

$$\begin{aligned}\dot{P}_2 &= \frac{E}{LA_L}(q_2 - q_1 - q_7) = \frac{E}{LA_L}(GP_{c1} - GP_2 - q_1 - q_7) \\ \dot{P}_4 &= \frac{E}{LA_L}(q_4 - q_3 - q_5) = \frac{E}{LA_L}(GP_{c2} - GP_4 - q_3 - q_5)\end{aligned}\quad (5)$$

where

G - proportional coefficient of servo valve controller

P_{c1} - pressure control signal (reference pressure value of servo valve 1's controller)

P_{c2} - pressure control signal (reference pressure value of servo valve 2's controller)

Flow of working fluid from the accumulator and pump q_2 and q_4 are assumed to be controlled solely by the servo valve. As mentioned earlier, effects of accumulator and pump are ignored.

Pressures inside the cylinder chamber P_1 are given by

$$\dot{P}_1 = \frac{E}{sA_{pd}} \left(A_{pd} \dot{x}_1 + q_1 \right) = \frac{E}{sA_{pd}} \left(A_{pd} \left(-l\dot{\theta} + \dot{Z}_s - \dot{Z}_{u1} \right) + q_1 \right)\quad (6)$$

Note that $\dot{x} = -l\dot{\theta} + \dot{Z}_s - \dot{Z}_{u1}$ represents the relationship of the piston's movement in relation to the car body's movements or in other words, coupling of effects of car body onto the active HIS system.

Similarly, the pressures in other chambers of the actuators are given by the following equations:

$$\begin{aligned}\dot{P}_3 &= \frac{E}{sA_{pu}} \left(A_{pu} \left(l\dot{\theta} - \dot{Z}_s + \dot{Z}_{u1} \right) + q_3 \right) \\ \dot{P}_5 &= \frac{E}{sA_{pd}} \left(A_{pd} \left(l\dot{\theta} + \dot{Z}_s - \dot{Z}_{u1} \right) + q_5 \right) \\ \dot{P}_7 &= \frac{E}{sA_{pu}} \left(A_{pu} \left(-l\dot{\theta} - \dot{Z}_s + \dot{Z}_{u1} \right) + q_7 \right)\end{aligned}\quad (7)$$

where

A_{pu} - cross-section area of the top of the piston

A_{pd} - cross-section area of the bottom of the piston (please refer to Nomenclature for explanation of other symbols)

The flow into the piston chamber is given by

$$\begin{aligned}F - Cq_1 - Re\dot{q}_1 &= ma \\ \therefore (P_2 - P_1)A_L - Cq_1 - Re\dot{q}_1 &= LA_L\rho\dot{q}_1 \\ \therefore \dot{q}_1(LA_L\rho + Re) &= -P_1A_L + P_2A_L - Cq_1 \\ \therefore \dot{q}_1 &= -\frac{P_1}{(LA_L\rho + Re)} + \frac{P_2}{(LA_L\rho + Re)} - \frac{C}{(LA_L\rho + Re)}q_1\end{aligned}\quad (8)$$

where

Re - pipe friction parameter

C - pipe damping parameter

Similarly, the flows injected into other piston chambers are recorded as:

$$\begin{aligned}\dot{q}_3 &= -\frac{P_3}{(LA_L\rho + Re)} + \frac{P_4}{(LA_L\rho + Re)} - \frac{C}{(LA_L\rho + Re)}q_3 \\ \dot{q}_5 &= -\frac{P_5}{(LA_L\rho + Re)} + \frac{P_4}{(LA_L\rho + Re)} - \frac{C}{(LA_L\rho + Re)}q_5 \\ \dot{q}_7 &= -\frac{P_7}{(LA_L\rho + Re)} + \frac{P_2}{(LA_L\rho + Re)} - \frac{C}{(LA_L\rho + Re)}q_7\end{aligned}\quad (9)$$

2.3 Model Integration

By looking at the configuration of the active HIS system in Figure 1 and Figure 5, the forces produced can be expressed as:

$$\begin{aligned}F_1 &= P_1A_{pd} - P_3A_{pu} \\ F_2 &= P_5A_{pd} - P_7A_{pu}\end{aligned}\quad (10)$$

Substituting F_1 and F_2 into the half-car model equations, the relations between various half-car dynamic parameters and the pressure inside the cylinders of the active HIS system can be obtained.

Having recognized the coupling interactions between the half-car model and the active HIS model, the active HIS system's new model can be assembled by combining these two sets of equations and expressing them in matrix form.