

H_{∞} Control of a Novel Low-Cost Roll-Plane Active Hydraulically Interconnected Suspension: An Experimental Investigation of Roll Control under Ground Excitation

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ABSTRACT

In order to make the active suspension more affordable, a novel low-cost active hydraulically interconnected suspension is developed, assembled and tested onto a sport utility vehicle. H_{∞} roll control strategy is employed to control vehicle body's roll motion. The hydraulic suspension model used for deriving the H_{∞} controller is estimated experimentally from the testing data. The active suspension model is then combined with the half-car model through their mechanical-hydraulic interface in the cylinders. The weighting function design of the H_{∞} control is provided. On a 4-post-test rig, the active suspension with H_{∞} control is validated with several road excitations. The test rig and experimental setup are explained and the obtained results are compared. The effectiveness of the designed H_{∞} controller is verified by the test data, with a considerable roll angle reduction in the three tests presented.

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1. INTRODUCTION

The vehicle suspension connecting a vehicle's wheels to its body is pivotal to the isolation of ground excitations, prevention of vehicle roll, and maintaining the tire/ground contact. The parameters of passive suspensions are fixed and generally nonadjustable after production. Vehicle handling and ride comfort commonly have conflicting requirements to suspension system design, and therefore compromise has to be made for passive suspensions. For example, a stiffer suspension produces better handling and a softer suspension possess better ride comfort[1].

Active suspensions have the ability of solving the conflicting requirements between ride and handling and hence have attracted wide attention in the past several decades. However, although there are some active suspension systems that are available on the market[2, 3], their applications are strictly limited to luxury vehicles. This is mainly due to two reasons: high manufacturing cost and excessive energy consumption.

This paper presents a safety-orientated, low-cost hydraulic active suspension system, containing four hydraulic double

directing cylinders, interconnected by two circuits in roll-plane. This way, the whole active suspension can be controlled by only one servo valve, which brings about substantial cost reduction compared to conventional active suspensions which use four valves for individual cylinder control. The active hydraulically interconnected suspension (HIS), evolved from passive HIS systems, directly supplies restoring forces to the vehicle chassis. It differs itself from semi-active suspension, which can only consume the vehicle vibration energy passively by adjusting the viscous damping coefficient of the shock absorbers. In the previous work, the modeling and dynamic analysis of roll-plane passive HIS system, has been investigated in[4,5,6], and experimental studies are included in [7, 8].

To counteract a vehicle body's roll motion induced by cornering, the active suspension needs to provide sufficient forces to the vehicle body, in order to negate the roll moment generated from the centrifugal force. From a mechanical point of view, the control to minimize roll angle during cornering (e.g, roll-free cornering) does not consume much energy, because only vertical forces vary while suspension

deflection remains unchanged [9] and not much work has been done during this process.

Stiff passive anti-roll systems, such as anti-roll bars or the passive anti-roll HIS system, are also adequate in reducing vehicle roll angle during cornering; however, due to the increased roll stiffness, vehicle's ride comfort is slightly compromised under ground excitations, in particular, the coherence of the excitations along the left and right tracks. In contrast, the active HIS is able to compensate for the road unevenness and minimize the vehicle body-roll angle which is superior to passive HIS. For vehicle body leveling control under ground excitation, since the active HIS needs to do work to expand or compress actuators to achieve the desired suspension deflection, there will be energy consumption from its operation.

From the above discussion of the two control scenarios, the active suspension system can be very energy efficient to control handling and safety related objectives, but to achieve better ride performance (body leveling control), a certain amount of energy is required. So the overall energy management strategy for active suspension depends on how to balance ride comfort and handling related control objectives.

Various control approaches have been applied to improve the active suspension performance in recent decades, such as root locus [10], linear and fuzzy control [11], neural networks [12], LQG and LQR [13], H_∞ control [14], learning automata [15], nonlinear control [16], and preview control [17]. Recognized as one of the most robust and effective optimal control strategies, H_∞ has been employed in active suspension control for many years, but majority of the research works are quarter-car model based [18, 19]. Some active H_∞ control researches those are based on half-car [20] or full-car models [1, 21] show promising results, but require four independent actuators on each wheel station to realize. The aforementioned researches are predominately conducted in the computer-based simulation environment [14, 22]. The experimental H_∞ control implementation on a full-car level with a practical active suspension system is of great interests to extend the research on active suspensions into engineering practice.

The effectiveness and performance of the derived H_∞ controller are largely determined by the mathematical modeling of the physical plant, thus an appropriate model and correct parameters of the active suspension (mainly the pressure control unit) are essential for this model-based optimal controller. The modeling has to be simple enough for the controller design and also captures the system dynamic characteristics. An empirical approach is proposed to derive the model and estimate the necessary parameter values. The derived active suspension model is then combined with a half-car model, and the combined dynamic model is then used to derive the H_∞ controller for vehicle body roll angle control.

Vehicle untripped rollover accidents are dangerous and vehicles with high center of gravity (CG) are particularly

vulnerable to these accidents. In this paper, a typical sports utility vehicle (SUV) is selected as the test vehicle and assembled with the developed active HIS system. Based on a real-time communication board, an H_∞ controller developed in the Simulink environment is applied through the proposed active suspension to control the test vehicle's body roll motion. The test is performed on a full-scale 4-post test rig which provides the ground excitation to represent the uneven road surface. Ground roll excitation is described as two out-of-phase inputs acting on the left and right wheels to stimulate the roll motion of the vehicle body. In this paper, three sets of different ground roll excitations are used to stimulate the roll motion of the vehicle body.

The paper is outlined as follows. The Active HIS is introduced in Section 2. The hydraulic model of the pressure control unit is derived in Section 3, assembled with the half-car model in Section 4. In Section 5, the theory of the H_∞ control is presented and the weighting function selections are proposed. In Section 6, the experimental results are presented. In Section 7 the results are discussed and compared. The conclusions are included in Section 8.

2. ACTIVE HYDRAULICALLY INTERCONNECTED SUSPENSIONS

The schematic drawing of this active HIS is provided in Figure 1, where we can see the active HIS is made up of four hydraulic actuators which are interconnected to form two hydraulic circuits, denoted as A and B. The two circuits are then controlled by a pressure control unit which consists of a tank, a pressure pump, an accumulator and a servo-valve. The CAD model and a prototype of this power unit are shown in Figure 2 (a-b) respectively. The cylinders are mounted between wheels and chassis, and the front and rear mounting are shown in Figure 3.

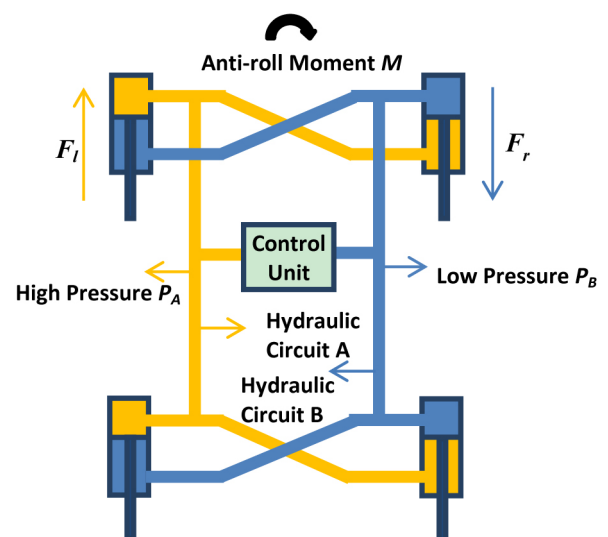
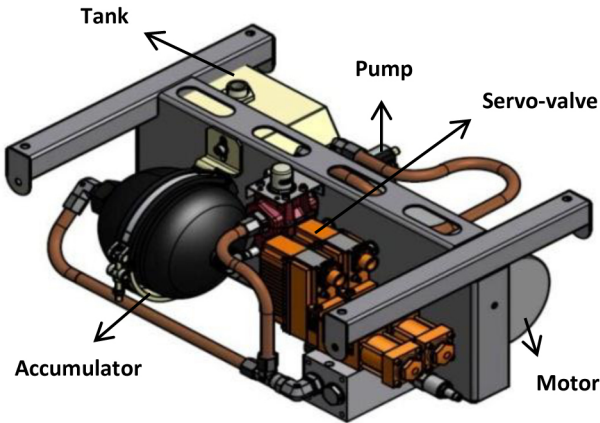
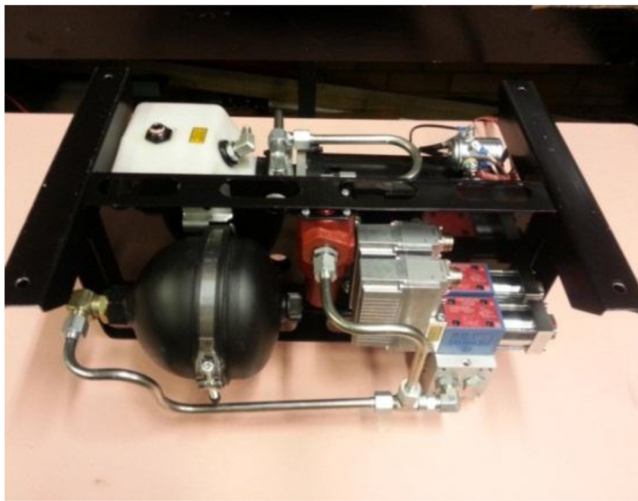


Figure 1. Schematic diagram of active HIS



a. CAD model



b. Assembly

Figure 2. Designed and assembled pressure control unit of the active HIS suspension



a. Front

Figure 3.



b. Rear

Figure 3. Photos of actuators' mounting

The four cylinders are interconnected into two enclosed hydraulic circuits in roll-plane, and executed by the pressure control unit. The controller acquires the vehicle body roll angle measurement and provides an according pressure command to the active HIS. The mechanism of the active hydraulic suspension is described as follows.

To be specific, in the ground excited roll motion control, one side (e.g., left) of the suspension needs to extend while another side (e.g., right) needs to compress in order to maintain vehicle body's leveling. In this circumstance, the pressure control unit will pump oil into the circuit A to increase its pressure, while reduce the pressure (if not tank pressure) in the circuit B, as shown in Figure 1. The pressure difference of circuit A and B forms an anti-roll moment to the vehicle body through four cylinders. The control forces and moment are presented in Equations (1,2,3).

$$F_l = 2a(P_A - P_B) \quad (1)$$

$$F_r = 2a(P_A - P_B) \quad (2)$$

$$M = F_l l / 2 + F_r l / 2 \quad (3)$$

where, F_l and F_r are the forces effected on the vehicle body by the left and right actuators respectively; P_A and P_B are the pressures inside the two hydraulic circuits; a is the piston area; M is the anti-roll moment and l is the distance between left and right actuators.

3. MODEL ESTIMATION OF PRESSURE CONTROL UNIT

As discussed above, the pressure control unit includes a motor-pump set, an accumulator and a pressure control servo-valve, as well as a tank, a filter and other hydraulic fittings.

The modeling and parameter estimation of the active system are vital to the control performance of the model-based controllers, e.g., H_2 , H_∞ in our case. The way of individually modeling each element could be theoretically accurate and ideal, but nevertheless, the parameter values for this ideal model are very hard to be derived and the estimation method for these physical parameters is very much limited by available testing methods. Without correct parameters, a complex model with larger degree of freedom (DoF) makes less sense. Hence, an empirical modeling method [23] is proposed to estimate the hydraulic model and corresponding parameter values by testing the assembled active pressure control unit separately from the vehicle system.

The setup for the testing is illustrated in Figure 4 and described as follows. The output ports from the pressure control unit are connected to a double acting cylinder which is locked in a middle position. A pressure transducer is located at the port to the upper cylinder chamber. The pressure command signal sent to the servo valve is a sinusoidal signal at different frequencies, and the actual pressure measured at the cylinder port is acquired for comparison.

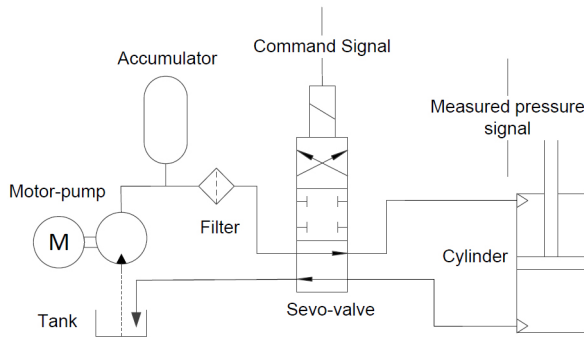


Figure 4. Schematic drawing of the testing setup

Both command and measured signal are collected by a National Instruments data acquisition board. 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 5.

From the plotted Bode graph in Figure 5, a second order system is considered sufficient to cover the main dynamic characteristics of the system, shown in Equation (4):

$$\frac{P(s)}{I(s)} = \frac{1}{1 + (2\xi / w_n)s + s^2 / w_n^2} \quad (4)$$

$P(s)$ 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 5.

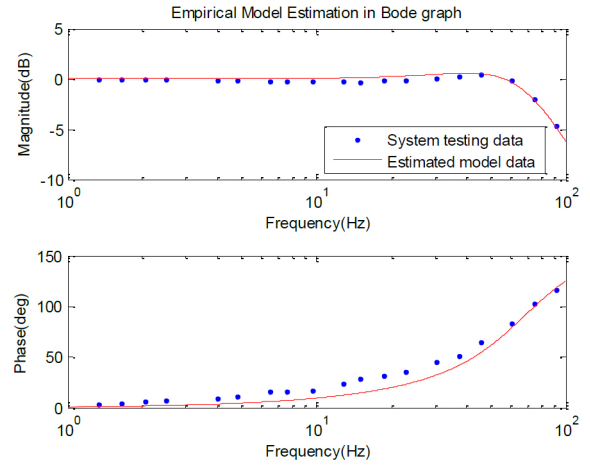


Figure 5. Empirical model estimation in Bode graph

To be consistent with the vehicle modeling from previous work [24], 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

$$\dot{X}_p = A_p X_p + B_p inp$$

$$Y_p = C_p X_p + D_p inp$$

(5)

where

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

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

(6)

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

4. MODEL INTEGRATION

A model of smaller dimension is preferable in the designing of advanced controllers, so the obtained hydraulic model is then combined with a 4-dof half-car model shown in Figure 6.