

Characteristic Analysis of Roll and Pitch Independently Controlled Hydraulically Interconnected Suspension

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ABSTRACT

This paper presents the modeling and characteristic analysis of roll-plane and pitch-plane combined Hydraulically Interconnected Suspension (HIS) system. Vehicle dynamic analysis is carried out with four different configurations for comparison. They are: 1) vehicle with spring-damper only, 2) vehicle with roll-plane HIS, 3) vehicle with pitch-plane HIS and 4) vehicle with roll and pitch combined HIS. The modal analysis shows the unique modes-decoupling property of HIS system. The roll-plane HIS increases roll stiffness only without affecting other modes, and similarly pitch-plane HIS increases the pitch stiffness only with minimum influence on other modes. When roll and pitch plane HIS are integrated, the vehicle ride comfort and handling stability can be improved simultaneously without compromise. A detailed analysis and discussion of the results are provided to conclude the paper.

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INTRODUCTION

For a road vehicle, the suspension system serves mainly two purposes. One is to support the occupants or cargo and isolate from severe levels of shock and vibration induced by the road surface. The other is to enable the wheels to maintain contact with the ground, assuring traction, stability and control of the vehicle. The vehicle motion-modes need to be looked into when studying vehicle dynamics. There are seven modes for a seven degree-of-freedom (DOF) full vehicle model, that is body-dominated bounce, pitch and roll mode, and wheel-dominated bounce, pitch, roll and warp mode. Warp mode is also known as articulation which diagonally-opposed wheels move in phase relative to the vehicle body while body motion is negligible. This constitutes a non-planar mode, allowing the vehicle to travel on spatial surfaces. Different modes of four-wheel vehicle don't share the same preference with reference to suspension stiffness and damping. For instance, a soft bounce mode is desired for ride comfort, whilst stiff roll and pitch modes are beneficial for inhibiting vehicle attitude during steering, braking, and acceleration. Warp mode should be as soft as possible in order to enhance vehicle road holding performance. The conflicting requirement between ride and handling is the main challenge and the trade-off has to be made for conventional suspension.

Active controlled suspension has received much attention in research and industry field, but passive suspension systems still remain the dominant type in current vehicles, mainly due to their cost/energy effectiveness and reliability. Interconnected suspension could easily decouple different vehicle vibration

modes in a passive manner. The interconnections can be realized by various means, such as mechanical, hydraulic and pneumatic [1]. Anti-roll bars are one of the typical examples of mechanical interconnection which it decouples the roll mode from bounce and pitch. But the drawback is that it is unable to decouple the warp mode from other modes. Namely, anti-roll bars can stiffen the roll mode without affecting bounce and pitch mode, but the warp mode is stiffened unfavorably. This affects vehicle handling/stability due to increased variation in vertical tyre-ground contact forces.

Roll-plane interconnected suspension systems have been studied theoretically and experimentally. Cao etc. [2, 3] investigated the roll-plane interconnection with proposed twin-gas-chamber hydro-pneumatic suspension strut, and compared the roll property of different connections. Zhang & Smith etc. [4, 5] investigated the modeling and dynamics of vehicles fitted with roll resistant hydraulically interconnected suspension (HIS) system with a focus on the roll stability and fluid circuit dynamics. In [6], ride comfort of the vehicle with roll-plane HIS system was analyzed. Experimental studies also confirmed the performance of HIS systems in [7, 8]. Road holding performance of HIS system especially at warp mode has been investigated and compared with anti-roll bars in [9, 10], it shows that anti-roll bar stiffened the warp mode which increased warp natural frequency, while with roll-plane HIS suspension, the warp natural frequency remains unchanged. The tyre normal force response at warp mode also illustrates that dynamic tyre force variation of vehicle fitted with anti-roll bar increased 50% more than that of vehicle fitted with

roll-plane HIS. Tenneco's Kinetic H2/CES suspension is one of the commercialized HIS suspension developed to improve the lateral stability and road holding performance [12].

With the unique modes-decoupling of the HIS system, the ride quality can be maintained when improving lateral stability. But when the suspension design is biased towards ride comfort, the longitudinal stability should also be considered during acceleration or braking. In this paper, both roll and pitch resistant HIS is proposed and modelled. Additional set of 4 cylinders are added to the roll-plane HIS with fluid connection to against the pitch motion. So the roll and pitch stiffness can be tuned independently. Bounce and warp stiffness which relies on the coil springs are maintained, or can be further reduced to improve the ride comfort without affecting the handling. Characteristic analysis is performed to show the advantages and drawbacks of the proposed system.

VEHICLE MODELLING

Conventional Suspension

A vehicle can be modelled as a system with seven DOFs as shown in Figure 1. It consists of a rigid sprung mass supported by four independent suspensions, which includes wheel assemblies as four unsprung masses.

The sprung mass has inertial properties of mass (M_s), roll moment of inertia I_{xx} , and pitch moment of inertia I_{yy} . The roll and pitch rotations are restricted only to the sprung mass. The seven DOFs of the vehicle model includes: sprung mass vertical center-of-mass movement Z_{v0} , roll angle ϕ_v , pitch angle θ_v and four vertical center-of-mass movements of unsprung masses: Z_{w1} Z_{w2} Z_{w3} Z_{w4} . The origin of coordinates is fixed in the center of gravity (CG) of sprung mass. One assumption in this study is the linear behavior of the suspension springs and shock absorbers.

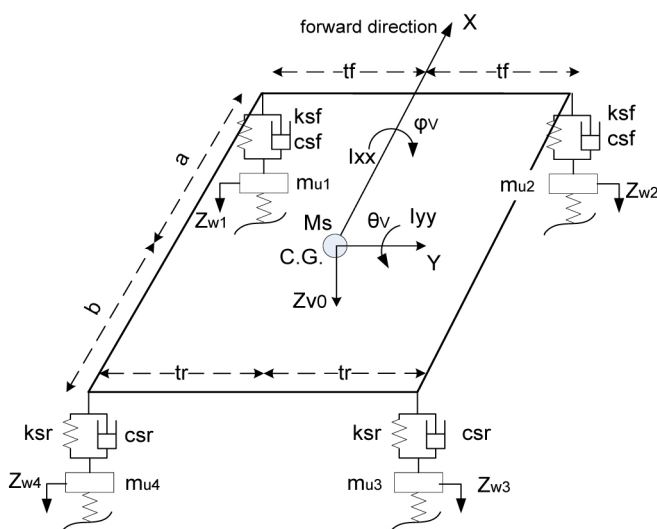


Figure 1. Seven-DOF vehicle model

Using free body diagram approach and applying Newton's second law, the full vehicle suspension equations of motion is derived and written in the matrix form:

$$M \ddot{Z} + C \dot{Z} + KZ = F \quad (1)$$

Where M , C , $K \in R^{7 \times 7}$ are the mass, damping and stiffness matrices; Z is the displacement vector at continuous time t ; F is the road excitation force input matrix.

Equation (1) can be converted to a continuous-time state space model as:

$$T \dot{X} = SX + F_{ex}, \quad \text{or} \quad \begin{bmatrix} I_7 & 0 \\ 0 & M \end{bmatrix} \begin{bmatrix} \dot{Z} \\ \ddot{Z} \end{bmatrix} = \begin{bmatrix} 0 & I_7 \\ -K & -C \end{bmatrix} \begin{bmatrix} Z \\ \dot{Z} \end{bmatrix} + \begin{bmatrix} 0 \\ F_{ex} \end{bmatrix} \quad (2)$$

The state vector describing the motion of the sprung and unsprung lumped suspension system is defined as

$$X_M = \begin{bmatrix} Z & \dot{Z} \end{bmatrix}^T \quad (3)$$

where displacement vector is $Z = [Z_{w1} \ Z_{w2} \ Z_{w3} \ Z_{w4} \ Z_{v0} \ \phi_v \ \theta_v]$ and velocity vector is

$$\dot{Z} = [\dot{Z}_{w1} \ \dot{Z}_{w2} \ \dot{Z}_{w3} \ \dot{Z}_{w4} \ \dot{Z}_{v0} \ \dot{\phi}_v \ \dot{\theta}_v]$$

Vehicle Fitted with Roll and Pitch Plane HIS

Description

The model of vehicle fitted with HIS system contains two subsystems: one mechanical and one hydraulic.

The hydraulic layout is shown in Figure 2. The inside 4 cylinders and fluid communication circuits A and B forms the roll-plane HIS. Additional set of 4 cylinders at outside and fluid communication circuits C and D formed the pitch-plane HIS. Totally, it includes four nitrogen filled diaphragm type accumulators; interconnecting pipelines; and 8 double-acting hydraulic cylinders, adding to the original suspension. Cylinders are designed to make the upper and lower chamber cross-section areas are as close as possible. The cylinder bodies are mounted to the car chassis with the piston rods being fixed to the wheel assemblies.

In the roll motion, pressures in the chamber of roll-plane cylinders (inside) changes due to gas volume change in the accumulators, and the generated cylinder forces are against the roll motion to provide additional roll stiffness. The circuit connection of pitch-plane cylinders only enables the fluid flows from upper chamber of one cylinder to lower chamber of another cylinder, so just minimum pressure changes and influence on the roll-plane suspension. Similarly at pitch motion, pitch-plane HIS provides

additional pitch stiffness while the roll-plane HIS has negligible impact on it. This property of the proposed interconnected suspension enables it to decouple the four vehicle modes and can be independently tuned without compromise.

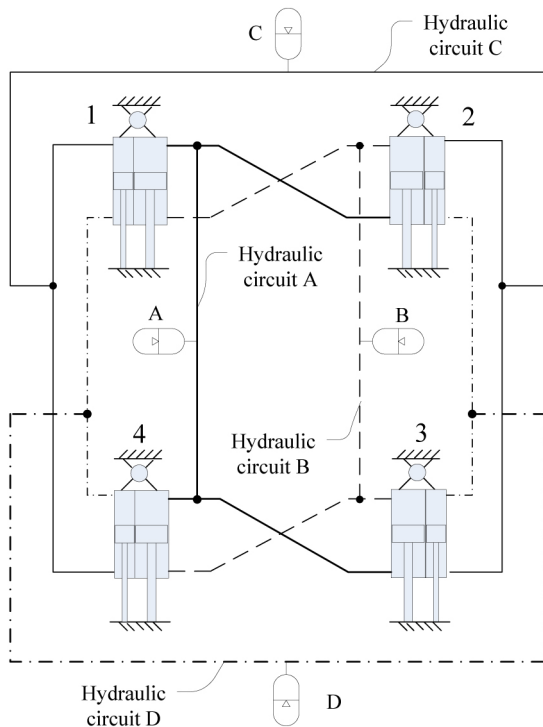


Figure 2. Schematic diagram of the roll-plane and pitch-plane combined HIS system

Mechanical System Equations

The equation of motion for the vehicle of integrated mechanical-hydraulic system can be written in the form:

$$M\ddot{Z} + C\dot{Z} + KZ + D_1 \cdot (A_R \cdot P_R + A_P \cdot P_P) = F_{ext}(t) \quad (4)$$

Where the displacement vector Z , mass matrix M , damping matrix C , stiffness matrix K and excitation forces F are as defined in the previous section; $D_1 \cdot (A_R \cdot P_R + A_P \cdot P_P)$ describes the forces of the cylinders due to the hydraulic pressure change; D_1 is linear transformation matrices. The area matrix A_R , A_P and pressure vector P_R , P_P related to the corresponding cylinder chambers (T-top; B- bottom), are defined as:

$$A_R = \text{diag} \begin{bmatrix} A_{RT}^1 & A_{RB}^1 & A_{RT}^2 & A_{RB}^2 & A_{RT}^3 & A_{RB}^3 & A_{RT}^4 & A_{RB}^4 \end{bmatrix};$$

$$A_P = \text{diag} \begin{bmatrix} A_{PT}^1 & A_{PB}^1 & A_{PT}^2 & A_{PB}^2 & A_{PT}^3 & A_{PB}^3 & A_{PT}^4 & A_{PB}^4 \end{bmatrix};$$

$$P_R = \begin{bmatrix} P_{RT}^1 & P_{RB}^1 & P_{RT}^2 & P_{RB}^2 & P_{RT}^3 & P_{RB}^3 & P_{RT}^4 & P_{RB}^4 \end{bmatrix}^T;$$

$$P_P = \begin{bmatrix} P_{PT}^1 & P_{PB}^1 & P_{PT}^2 & P_{PB}^2 & P_{PT}^3 & P_{PB}^3 & P_{PT}^4 & P_{PB}^4 \end{bmatrix}^T$$

Fluidic Equations

Assumptions are made in this paper as: (a) piston friction is not considered. (b) The fluid is incompressible compared with nitrogen gas in the accumulator. (c) The oil density is constant. (d) Pipeline is simplified with no viscous resistance, namely pressure and flow rate inside the pipelines are equally throughout. A more complicated model with consideration of fluid viscous resistance has been developed to study the damping effects of the fluid system in reference [5].

The cylinder chambers form the boundary between the mechanical and hydraulic subsystems. The compressibility of the hydraulic fluid in each of the cylinder chambers is shown as:

$$Q_{comp} = \frac{V}{\beta} \frac{dP}{dt} \quad (5)$$

Where V and β are the volume and effective bulk modulus of the cylinder chamber.

Thus the fluid compressibility in cylinder chamber is given by

$$Q_{comp} = \dot{Z}_{s-u}(t) A - Q(t) = \frac{V_0 - Z_{s-u}(t) A}{\beta} \dot{P}(t) \quad (6)$$

where $Q(t)$ denote the volume flow rate where the pipeline meets the cylinder chamber; $\dot{P}$ are the rates of change of in-chamber pressure; V_0 are the initial volume of the cylinder chambers; and Z_{s-u} is the relative displacement between the unsprung mass and the point of strut contact at the corner of the sprung mass.

When applied to all chambers in the fluid system in Fig. 2, Equation (6) can be re-written as:

$$Q_R = A_R \cdot D_4 \cdot \dot{Z}(t) + V(t) \cdot \dot{P}(t), \quad Q_P = A_P \cdot D_4 \cdot \dot{Z}(t) + V(t) \cdot \dot{P}(t) \quad (7)$$

In which Q_R , Q_P are the flow vector as

$$Q_R = \begin{bmatrix} Q_{RT}^1 & Q_{RB}^1 & Q_{RT}^2 & Q_{RB}^2 & Q_{RT}^3 & Q_{RB}^3 & Q_{RT}^4 & Q_{RB}^4 \end{bmatrix}^T,$$

$$Q_P = \begin{bmatrix} Q_{PT}^1 & Q_{PB}^1 & Q_{PT}^2 & Q_{PB}^2 & Q_{PT}^3 & Q_{PB}^3 & Q_{PT}^4 & Q_{PB}^4 \end{bmatrix}^T,$$

$V(t)$ is a time-variant matrix of cylinder volume and bulk modulus terms, and D_4 is a constant linear transformation matrix.

The accumulators are modeled by assuming an adiabatic process. The pressure and volume at any time in the accumulator P_a and V_a are related to the pre-charged values, P_p and V_p , as follows: