FULL VEHICLE ACTIVE SUSPENSION: SENSOR FAULT DIAGNOSIS AND FAULT TOLERANCE

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Abstract: In this paper, sensor fault detection, identification and fault tolerance model-based approach is designed for a linear full vehicle's active suspension system. The approach uses a bank of reduced order sliding mode observers to generate residuals. Residuals are defined in such a way to isolate the faulty sensor after detecting the occurrence of the fault. Once detected and isolated, the sensor fault is accommodated by replacing the faulty measurement by its estimation. In this study, sensor drift, bias and complete breakdown are considered. Simulation is made to illustrate the proposed strategy.

Keywords: Active vehicle suspension, Sensor failures, Fault diagnosis, Fault tolerant systems.

1. INTRODUCTION

Unlike the passive suspension system, in the active suspension, a force actuator is able to both add and dissipate energy from the system. This results in the capability of the suspension system to control the vehicle dynamics, to reduce the effects of braking and the vehicle roll during cornering maneuvers in addition to increasing the ride comfort and vehicle road handling.

The active suspension control problem has been studied by many researchers: state and output feedback scheduled controller (Köse and Jabbari, 2003), a modular adaptive robust control technique (Chantranuwathana and Peng, 2004), a fuzzy logic (Stríbrský *et al.*, 2003), a stochastic optimal (Marzbanrad *et al.*, 2004), a mixed H_2/H_∞ (Gáspár *et al.*, 2000), a combined filtered feedback and an input decoupling transformation (Ikenaga *et al.*, 2000) controllers and many other controllers were designed and applied on vehicle active suspension system.

However, the behavior of the active suspension system depends on the information provided by the sensors. Thus any incorrect information caused by a faulty sensor can lead the system to undesirable or even dangerous behavior.

Few are the researches which treat the active suspension fault diagnosis problem: statistical methodologies are applied to perform fault detection in nonlinear two degrees of freedom quarter-car model and complete vehicle models (Metallidis *et al.*, 2003). Model based fault detection and identification methods are developed in (Fischer

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et al., 2003), (Börner *et al.*, 2002), and analytical redundancy techniques are used for fault detection for active heavy vehicle suspension (Jeppesen and Cebon, 2002). Fewer treat the suspension fault tolerance problem: a design of a bank of Kalman filters, one for each possible sensor failure configuration, providing an estimation of the system state when a sensor fault occurs, is carried out for a quarter car test rig (Silani *et al.*, 2004). A methodology for the comparison among different alternative fault tolerant architectures, based on risk evaluation has been applied to a full active suspension control system (Borodani *et al.*, 1996).

In this paper a model-based sensor fault detection, identification and tolerance strategy is designed for a linear full vehicle's active suspension system, which is not treated before. To detect and identify sensor failure, a bank of reduced order sliding mode observers is used to generate residuals. Residuals are designed in such a way that every sensor fault has a specific pattern and thus it can be easily identified. To accommodate for the sensor fault, the faulty measurement is replaced by its estimation. This study treats the sensor bias, drift and the sensor breakdown.

This paper is organized as follows: in section 2 the active suspension system is presented and the system model is given. The control strategy of the system is briefly presented in section 3. The used sensors are presented in section 4. In section 5 and 6, the strategy of fault detection, identification and tolerance is explained. Simulation is done in section 7 to illustrate the proposed strategy. Finally, the conclusions and future works are given.

2. ACTIVE SUSPENSION SYSTEM

2.1 System description

The full vehicle model consists of the car body (sprung mass) connected by the suspension systems to four wheels (unsprung masses). This system is illustrated in Figure 1. Each active suspension system is modeled as a linear viscous damper, a linear spring and a force actuator. Each wheel is modeled as a linear spring. The car body is free to heave, pitch and roll, and the wheels are free to bounce vertically with respect to the car body.

2.2 System model

The system model can be written as a linear state space representation, as following (Ikenaga *et al.*, 2000):

$$\begin{cases} \dot{x} = Ax + Bu + Fd \\ y = Cx \end{cases} \quad (1)$$

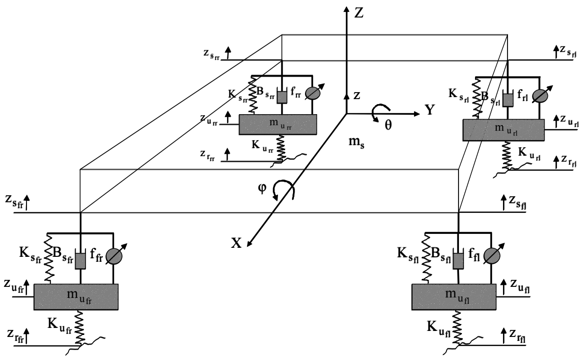

Fig. 1. Full vehicle model

Where $x \in \Re^{14}$ is the state vector which are:

x_1 : heave position of the sprung mass (z),
x_2 : heave velocity of the sprung mass ($\dot{z}$),
x_3 : pitch angle (θ),
x_4 : pitch angular velocity ($\dot{\theta}$),
x_5 : roll angle (ϕ),
x_6 : roll angular velocity ($\dot{\phi}$),
x_7 : front-left unsprung mass height ($z_{u_{fl}}$),
x_8 : front-left unsprung mass velocity ($\dot{z}_{u_{fl}}$),
x_9 : front-right unsprung mass height ($z_{u_{fr}}$),
x_{10} : front-right unsprung mass velocity ($\dot{z}_{u_{fr}}$),
x_{11} : rear-left unsprung mass height ($z_{u_{rl}}$),
x_{12} : rear-left unsprung mass velocity ($\dot{z}_{u_{rl}}$),
x_{13} : rear-right unsprung mass height ($z_{u_{rr}}$),
x_{14} : rear-right unsprung mass velocity ($\dot{z}_{u_{rr}}$).

$u = (f_{fl}, f_{fr}, f_{rl}, f_{rr})^T \in \Re^4$ is the vector of the system inputs which are the four actuator forces, and $d = (z_{r_{fl}}, z_{r_{fr}}, z_{r_{rl}}, z_{r_{rr}})^T \in \Re^4$ is the vector of the unknown inputs which are the road irregularities. The output y is discussed later in the paper. A, B and F are constant matrices with appropriate dimensions.

Because of the lack of space, neither the system equation nor the parameters definition are given here. For more details concerning the physical model of the system, the reader can refer to (Ikenaga *et al.*, 2000).

3. SYSTEM CONTROL STRATEGY

The main objective of the system control is to reduce the effect of the road irregularities on the passengers and to insure the system safety during the maneuvers of the vehicle. The adopted control strategy is the one designed by Ikenaga, *et al* (2000) which is shown in Figure 2.

Where:

$P^d = (z^d, \theta^d, \phi^d)^T \in \Re^3$ is the vector of the desired heave position, pitch and roll angles,
$P = (z, \theta, \phi)^T \in \Re^3$ is the vector of the actual heave position, pitch and roll angles,
$\dot{P} = (\dot{z}, \dot{\theta}, \dot{\phi})^T \in \Re^3$ is the vector of the actual heave velocity, pitch and roll angular velocities,
$z_s - z_u = (z_{s_{fl}} - z_{u_{fl}}, z_{s_{fr}} - z_{u_{fr}}, z_{s_{rl}} - z_{u_{rl}}, z_{s_{rr}} -$