Journal of Zhejiang University-SCIENCE C (Computers & Electronics)
ISSN 1869-1951 (Print); ISSN 1869-196X (Online)
www.zju.edu.cn/jzus; www.springerlink.com
E-mail: jzus@zju.edu.cn

Review:

Advances in the control of mechatronic suspension systems*

Wajdi S. ABOUD[1,3], Sallehuddin Mohamed HARIS[1], Yuzita YAACOB[2]
([1]*Centre for Automotive Research, Universiti Kebangsaan Malaysia, UKM Bangi 43600, Malaysia*)
([2]*Faculty of Information Science and Technology, Universiti Kebangsaan Malaysia, UKM Bangi 43600, Malaysia*)
([3]*Institute of Technology-Baghdad, Foundation of Technical Education, Baghdad, Iraq*)
E-mail: wajdisadik@gmail.com; salleh@eng.ukm.my; yy@ftsm.ukm.my

Received Jan. 20, 2014; Revision accepted July 3, 2014; Crosschecked Sept. 17, 2014

Abstract: The suspension system is a key element in motor vehicles. Advancements in electronics and microprocessor technology have led to the realization of mechatronic suspensions. Since its introduction in some production motorcars in the 1980s, it has remained an area which sees active research and development, and this will likely continue for many years to come. With the aim of identifying current trends and future focus areas, this paper presents a review on the state-of-the-art of mechatronic suspensions. First, some commonly used classifications of mechatronic suspensions are presented. This is followed by a discussion on some of the actuating mechanisms used to provide control action. A survey is then reported on the many types of control approaches, including look-ahead preview, predictive, fuzzy logic, proportional–integral–derivative (PID), optimal, robust, adaptive, robust adaptive, and switching control. In conclusion, hydraulic actuators are most commonly used, but they impose high power requirements, limiting practical realizations of active suspensions. Electromagnetic actuators are seen to hold the promise of lower power requirements, and rigorous research and development should be conducted to make them commercially usable. Current focus on control methods that are robust to suspension parameter variations also seems to produce limited performance improvements, and future control approaches should be adaptive to the changeable driving conditions.

Key words: Mechatronics, Active suspensions, Semi-active suspensions, Multiple model adaptive control
doi:10.1631/jzus.C14a0027 **Document code:** A **CLC number:** TP273; TB535

1 Introduction

Statistics show that many people lose their lives or suffer from non-fatal injuries from road accidents yearly. These statistics can be reduced with the use of mechatronic suspensions. It is known that mechatronic suspensions could play a significant role in ensuring the safety of road users. This is made possible by the generation of forces by the suspension system that transmits onto the road surface to provide more comfortable and safer rides, providing improved handling capabilities. The suspension system allows the driver to always be in control of the vehicle in critical situations. Also, by providing a more comfortable ride, the driver would be less susceptible to physical fatigue. Suspensions also significantly influence perceptions of the comfort and safety features of vehicles, which are paramount concerns when purchasing a vehicle. Thus, these factors provide motivation for greater developments and improvements in the design of vehicle suspensions.

The main parts of a passive suspension system comprise wheels and tyres, the wheel carrier system, damper and spring elements, brakes, and the steering mechanism (Rajamani, 2011). The dynamic behavior of passive automotive suspensions is primarily

* Project supported by the Ministry of Education, Malaysia (No. ERGS/1/2012/TK01/UKM/02/4)

determined by the choice of the damper damping coefficient and the spring stiffness. Different aspects are taken into consideration in the selection of these parameters. One aspect is that the driver needs to always be in control of the vehicle to ensure safety. This requires a stiff, well-damped coupling between the vehicle and the road, especially for non-stationary driving manoeuvres, such as when driving along a rough road or during cornering. Another aspect is that, for the suspension to provide a truly comfortable ride, a soft spring and damper setup is required so as to isolate the chassis from road induced vibrations (Rajamani, 2011). However, these requirements for the provision of comfort and safety are in conflict with one another.

These conflicting requirements can be overcome with the use of mechatronic active suspensions. Such suspensions employ controlled force-generating actuators between the chassis and the wheels in the case of active suspension systems, or adjustable dampers in the case of semi-active suspensions, to improve the dynamic behavior. A mechatronic suspension system employs some pneumatic, magnetorheological, hydraulic, or electromagnetic actuators to generate the control action. Researchers like Appleyard and Wellstead (1995) and Hrovat (1997) have conducted studies on practical applications of active suspension systems since the middle of the 1980s, in tandem with the development of microprocessors and electronics. More recently, Gavriloski *et al.* (2007), Genger (2009), Kruczek *et al.* (2010), and Elmadany and Qarmoush (2011) conducted surveys and presented theories and applications of active suspension control systems. It is noted that mainly ride comfort, suspension deflection, and road handling are used as indicators to evaluate suspension performance. Ride comfort and sprung mass acceleration are interrelated, road handling depends on the contact forces between tyres and the road surface, and suspension deflection is associated with the displacement between the sprung and unsprung masses (Yamashita *et al.*, 1990; Lai and Liao, 2002). For the past two decades, a large amount of research has been carried out, all with the aim of improving the performance of vehicle suspension systems. Due to the inherent conflicting nature of the performance criteria (for instance, enhancing ride comfort requires a larger suspension stroke and a smaller damping of the wheel-hop mode), often a degradation in ride safety results as a consequence (Liu *et al.*, 2005). In other words, the option for a more effective solution to the problem is still open.

The most important benefits gained from employing mechatronic suspension systems come from the flexibility provided by the actuator components. If this flexibility is not fully exploited, the mechatronic suspension cannot be tuned to deliver optimal results for every driving state. Consequently, the potential of delivering the maximum performance cannot be realized as the safety limit requirements under different road surface conditions call for a departure from conservatism in the controller design. It is from this observation that the need to develop new control approaches is felt.

Studies have been made on the applications, performance potentials, and algorithms of suspension control (Sharp and Crolla, 1987; Nagai, 1993; Smith and Walker, 2000; Fischer and Isermann, 2004). Also, there have been studies on the general limitations of mechatronic suspension concepts due to the restrictions imposed on actuator placement, passivity constraints, and bandwidth and energy consumption (Karnopp, 1983; 1986).

Control methods play an important role in determining the performance of mechatronic suspensions. Various approaches have been proposed, including proportional–integral–derivative (PID), optimal, robust, predictive, sliding mode, and adaptive control methods. The motivation for this paper is to examine the state-of-the-art in mechatronic suspension control mechanisms and approaches, so as to provide an overview of current trends and an indication of future directions in research and development.

2 Mechatronic suspension classification

A state-of-the-art mechatronic suspension system is as shown in Fig. 1, equipped with continuously variable dampers and active anti-roll bars that enable the driver to select between differently tuned suspension settings. Such modern active suspension systems require an external power supply, and they obviate the need for the integration of high bandwidth actuators in today's highly efficient production vehicles.

Mechatronic suspension systems can be categorized based on characteristics of the actuators,

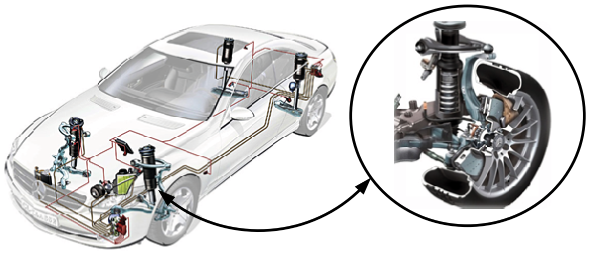


Fig. 1 Mercedes Benz mechatronic suspension system (Peter, 2012)

namely bandwidth, power demand, and the controllability range (Savaresi *et al.*, 2010). Savaresi *et al.* (2010) and Fijalkowski (2011) categorized five groups of mechatronic suspension systems. The following describes each of these groups:

1. Semi-active suspensions are defined as suspension systems with the capability of making quick adjustments to the damper and/or spring characteristics. One important characteristic of a semi-active system is that the forces produced by the semi-active element rely strongly on the direction of relative motion of that particular element. Semi-active dampers can change the level of energy dissipation, but they do not supply energy to the system. Note that these dampers have very low power consumption, approximately 20–40 W per damper. The bandwidth of semi-active dampers is up to approximately 40 Hz (Savaresi *et al.*, 2010). Semi-active dampers are integrated into the current versions of automobiles such as the BMW 7 series, Porsche 911, and Mercedes Benz E-class. Heiring and Ersoy (2011) cited an anti-roll bar with switchable additional springs as an example of a semi-active spring element.

2. Active suspension systems normally refer to slow active systems, often known as low bandwidth active systems. These types of system are characterized by an electrical linear motor or a hydraulic cylinder integrated into the system to generate forces independently, without relying on the relative motion between the body mass and the wheel. The bandwidth of slow active systems is approximately 5 Hz. Typically, low bandwidth active systems employ actuators which are merged in a series configuration to the primary spring. They tend to become stiff when their bandwidths are exceeded (Sharp and Crolla, 1987). The energy consumption of the system is in the range of 1–5 kW (Savaresi *et al.*, 2010).

3. Generally, fast active suspension systems are known as fully active or high bandwidth systems. The passive damping element can be substituted or replaced by an actuator with a bandwidth of 20 Hz or higher (Savaresi *et al.*, 2010). The actuators, which are fully active, are merged with the primary spring in a parallel configuration. In some applications, the passive damper is left out, although it can be taken into consideration to represent the effects of friction in a quarter-vehicle model. According to Heiring and Ersoy (2011), the main drawback of fully active suspension systems is high energy consumption, typically in the range of 4–20 kW.

4. Adaptive suspension systems are defined by slowly varying spring and damper characteristics, where the variation is dependent on vehicle velocity. Lowering the center of gravity of the vehicle ensures a higher sporty road holding ability. This type of suspension has been realized in the Porsche Panamera (2009 model) via the use of airsprings (Möller, 2009). In 1989, Citroën introduced the hydractive suspension which provides slow adjustment between different airspring characteristics and adjustable discrete settings for the damper characteristics (Altet *et al.*, 2003; Pyper *et al.*, 2003). The power consumption of such systems depends mainly on the energy required to vary the spring stiffness.

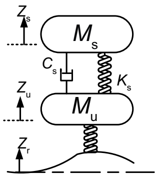
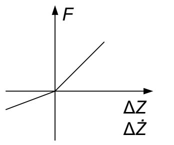
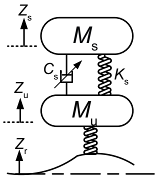
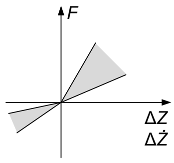
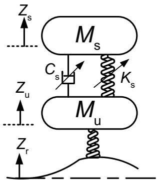
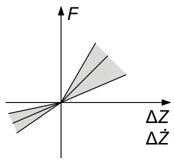
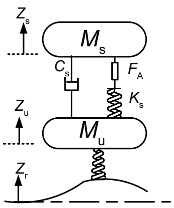
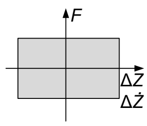
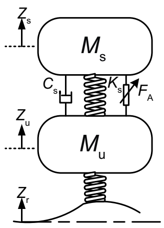
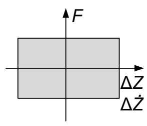
5. To compensate for various loading levels, automatic level control systems are used. These systems normally operate quasi-statically to maintain the distance between the chassis and the road at a constant distance (Heiring and Ersoy, 2011). A level control system functions through the use of airsprings and compressors. In this respect, a soft, comfort oriented suspension setup with adequate suspension travel distance can be considered to be a self-determining suspension, as it autonomously adjusts the chassis height according to the vehicle load level. The required power for automatic level control systems is 100–200 W (Savaresi *et al.*, 2010).

Table 1 shows the classification of suspension systems according to Fischer and Isermann (2004) and Koch (2011). This classification is not dissimilar from that of Savaresi *et al.* (2010), except that they do not include the automatic level control systems.

3 Suspension control mechanisms

In this section we describe the mechanisms that provide the control action in mechatronic suspension systems. For simplicity, we classify them into only

Table 1 Classification of suspension systems*

System type	Model	Force limitation	Operating range (Hz)	Energy demand
Passive			No actuator	0 W
Semi-active			0–40	Low
Adaptive			<1	Low
Slow active			0–5	Medium
Fully active			0–30	High

* Adopted from Fischer and Isermann (2004) and Koch (2011)

semi-active and active suspensions. Semi-active systems refer to suspensions where the control action is performed by varying the damping characteristics, while active systems make use of an actuator to provide an external compensating force.

3.1 Semi-active suspensions

The conflicting performance requirements of a suspension system can be improved by employing a damper with variable damping characteristics. Suspension performance is affected by road excitation. Hence, to improve performance, the damping is adjusted according to the road excitation.

Damping in a shock absorber is produced from the flow of hydraulic fluid through orifices. The slower the fluid flow, the larger the damping that can be generated. The damping coefficient can be controlled either by changing the fluid viscosity or by adjusting the orifice size. Today's semi-active suspension systems can rapidly adjust the shock absorber damper characteristics (Karnopp *et al.*, 1974). The following describes the physical principles of operation of three types of semi-active dampers (Chung and Shin, 2004; Heiring and Ersoy, 2011):

1. Magnetorheological dampers. In magnetorheological dampers, the viscosity of a magnetorheological fluid is changed by the application of a magnetic field, which causes the magnetic particles in the fluid to form chains (Gao and Yang, 2006). Delphi (2005) introduced a magnetorheological damping system known as MagneRide.

2. Electrorheological dampers. The operation of electrorheological dampers is based on varying the flow properties of the contained electrorheological fluids. An electrical field is used for the formation of particle chains in the fluid (Chung and Shin, 2004). An advantage of electrorheological dampers over magnetorheological ones is that the particles in the former are not abrasive, so the seals are more durable.

3. Hydraulic dampers. Hydraulic dampers dissipate energy by throttling hydraulic fluid between two or more chambers inside the damper. Technically, in a semi-active hydraulic damper, valves are used to vary the cross-section of the opening between the chambers. In turn, this causes variations in the level of hydraulic fluid dissipation (Guglielmino *et al.*, 2010). Codeca *et al.* (2008) and Xu and Guo (2010) presented exemplary applications for this widely used semi-active suspension damping principle.

The fact that a semi-active damper is a passive element, which cannot supply energy to the suspension system, is expressed by a passivity constraint (Giorgetti *et al.*, 2005; Savaresi *et al.*, 2010) for the velocity dependent damper force, $F_{\mathrm{A}}(t)$, which is given by

$$F_{\mathrm{A}}(t) = d_{\mathrm{c}}(\dot{x}_{\mathrm{c}}(t) - \dot{x}_{\mathrm{w}}(t)) \geq 0 \qquad \forall t,$$

where d_{c} is the damper coefficient, and $\dot{x}_{\mathrm{c}}(t)$ and $\dot{x}_{\mathrm{w}}(t)$ are the velocities of the sprung and unsprung masses, respectively. The damper spread is an important criterion for determining the achievable performance of semi-active suspensions, in the range of $d_{\mathrm{c,min}} \leq d_{\mathrm{c}} \leq d_{\mathrm{c,max}}$. The dynamics of the damper are determined by its electrical and fluid dynamical components. Semi-active damping systems are significantly more prominent in modern mechatronic vehicle suspensions than active systems due to their comparatively low energy consumption (only for the positioning of the electromagnetical valves and the generation of electrical or magnetic fields), low cost, and simple structure.

3.2 Active suspensions

Different from semi-active suspensions, an active suspension does not change the damper characteristics, but generally is implemented using an actuator that either replaces the suspension components or acts in parallel with them. The great virtue of an active suspension system is its ability to adapt to variable road conditions, and to employ the full suspension working space (allowable suspension stroke) to satisfy ride comfort and handling requirements. To understand apparently the subtle difference between semi-active and active suspensions, consider a hypothetical conflict with a known pothole. A semi-active system will make the suspension softer when hitting the pothole and stiff after the pothole. An active suspension could feasibly lift the wheel over the pothole, and thereby will improve both ride comfort and safety.

Sensors together with a microprocessor form the integral components of this type of suspension system. Clearly, this type of technology appears on very high-end vehicles. Nevertheless, the rapid advances in the science and technology of microprocessors, sensors, and actuators have brought a whole new range of features to the automobile industry.

High energy requirements have long been a prohibiting factor in employing an active suspension system in a production motorcar. Efatpenah *et al.* (2000), Graves *et al.* (2000), and Stribrsky *et al.* (2007) studied concepts related to energy management and recovery in suspension systems and showed that by using suitable electronic devices, energy consumption can be reduced to less than half of that used in conventional active suspension systems.

In the following, the types of actuators used in active suspension systems are discussed.

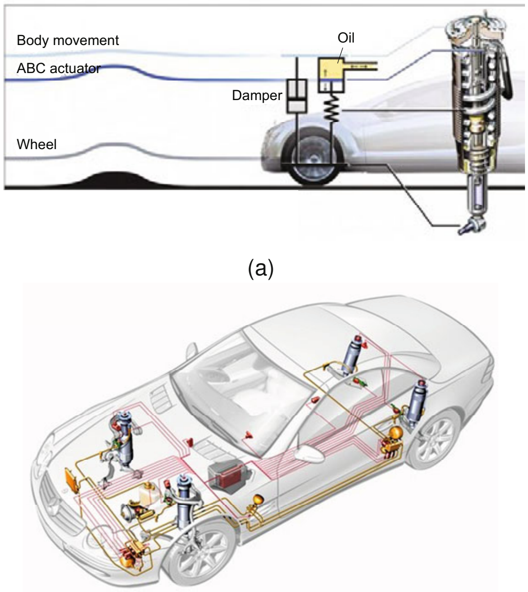
1. Oleo-pneumatic actuators. Williams and Best (1994) designed an oleo-pneumatic actuator with a low bandwidth active suspension. They also described the structure of the oleo-pneumatic actuator together with the control system. Paulides *et al.* (2006a) demonstrated a commercial low bandwidth active suspension using hydraulics, which is known as an active roll control system. Likewise, Martins *et al.* (2006) presented the hydraulic active suspension with its control schematic specifically designed for the quarter-car model.

2. Hydraulic actuators. Strassberger and Guldner (2004) described the active stabilizer bar system developed by BMW for the hydraulic active suspension. The system consists of the following components: a lateral accelerometer, an electronic control unit, a hydraulic pump and its oil reservoir, a hydraulic valve block, and two active stabilizer bars with rotating hydraulic actuators. The system plays

the following roles: (1) it can considerably decrease the roll angle during cornering; (2) it can remove the negative effects of passive stabilizer bars; and (3) it can regulate the dynamic characteristics of self-steering as a function of driving conditions and vehicle speed, leading to improved agility, handling and steering precision. Sam and Hudha (2006) introduced the modelling and force tracking control of a non-linear hydraulic actuator employed in a quarter-car hydraulic active suspension system.

Another commercial example of a low bandwidth active suspension system is found in some Mercedes Benz models. This low bandwidth active suspension system is known as the active body control (ABC) system. The system has been integrated into the S-class and the Coupe CL-class since 1999. It was also integrated into the SL Roadster in 2001 (Pyper *et al.*, 2003). Fig. 2 is an illustration of a Mercedes Benz car equipped with the ABC system (Peter, 2012). The ABC system comprises a hydraulic pump, a high-pressure accumulator, steel springs, hydraulic struts, dampers, sensors, and an electronic control unit. In this system, the suspension struts are placed between the body and wheels of the vehicle. An electronic unit controls the hydraulic system, in which the signals measured by the sensors are analyzed while the vehicle is in motion. The oil flow into the spring struts is controlled by the ABC system independently at each wheel. The movement of the hydraulic actuators compensates for road irregularities (roughness), and therefore the vehicle body movement is significantly reduced. To produce less drag and improve handling, the ABC system slowly lowers the vehicle height at highway speeds. The system actively damps the translator movement of the chassis mass, as well as the roll and pitch motions. It can also operate as a level control system (Pyper *et al.*, 2003). A passive damper, however, realizes the damping of the unsprung mass.

The controller structure of the Mercedes Benz ABC consists of four primary parts: skyhook control, feedforward control using lateral acceleration (to suppress roll movement) and longitudinal acceleration (to compensate for pitch movement), and the ABC algorithm (Becker *et al.*, 1996). The ABC algorithm suppresses pitch, roll, and heave motions of the chassis by feedback control. It also enables level control of the vehicle. The measured suspension deflections and chassis velocities of each suspension strut are used to calculate the modal quantities roll angle, pitch angle, and heave motion. Based on these quantities, the algorithm generates reference forces for the chassis motions and transforms them back as references for the local control forces acting at the struts (Becker *et al.*, 1996).



(a)

(b)

Fig. 2 Active body control (ABC) (BOSE, 2010; Peter, 2012): (a) hydraulic active suspension for the ABC system; (b) Mercedes Benz equipped with the ABC system

Heiring and Ersoy (2011) presented a new concept, known as ASCA (active suspension system with integrated body control and variable wheel damping). In this system, rotator actuators are employed to introduce forces between the chassis and wheel mass via the wishbone struts. The hydraulic actuator realizes passive damping through a throttle and a cam ring inside a pump. The configuration primarily acts as an integrated roll and damping control system. The advantage of this system is its comparably low power consumption due to the placement of the efficient actuators parallel to the airsprings (Heiring and Ersoy, 2011).

Other implementation studies on active suspensions were carried out by Williams and Best (1994) and Wu *et al.* (2011).

3. Magnetic and electromagnetic actuators. Jonasson and Roos (2008) presented a prototype of an electromechanical slow active suspension system which is integrated in series to the primary spring. As it is based on a spindle motor, it can lower energy

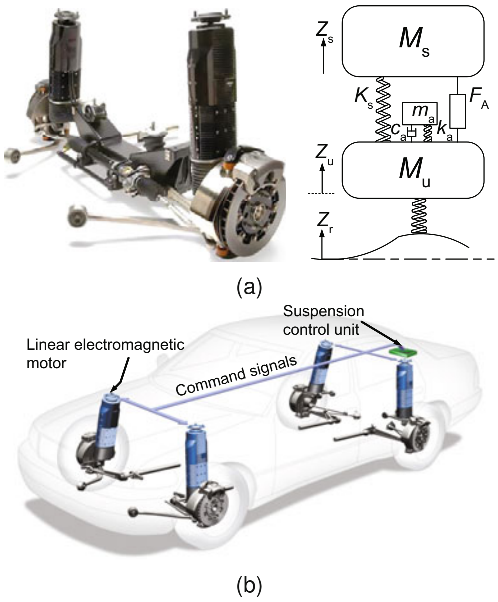


Fig. 3 High bandwidth active suspension concepts by BOSE (Jones, 2005; Liu and Wang, 2008; BOSE, 2010): (a) assembly of the front linear electromagnetic motor (left) and its quarter-car model (right); (b) full assembly of the linear electromagnetic motor

consumption compared to the hydraulic ABC system by up to 0.6 L/100 km.

BOSE has worked on a concept for a high bandwidth active suspension system since 1980. Fig. 3 shows the application of a high bandwidth system in vehicles. In this system, the conventional passive dampers are replaced by electrical linear motors, while the static load of the vehicle is suspended by torsion bars (Jones, 2005). A reaction mass absorber is attached to each wheel (Fig. 3a). This reduces the resonance peak at the unsprung mass natural frequency, inhibiting the transfer of reaction forces directly to the chassis (BOSE, 2010). Although the system is able to recover energy by driving the linear motors in generator mode, it is noted that power consumption of the system accounts for less than "one-third of the energy used by a car's air conditioner" (Jones, 2005). Martins *et al.* (1999; 2006) provided designs of electromagnetic actuators for active suspension systems, while Paulides *et al.* (2006b) provided designs for semi-active suspension systems.

The performances of electromagnetic active suspensions have been recorded for computational and analytical purposes. Gysen *et al.* (2009), being concerned with aspects in the design of electromagnetic active suspensions, presented a passive spring and a slotless brushless tubular permanent magnet actuator. It was found that the proposed actuator design yielded optimum results. Paulides *et al.* (2006a) presented and discussed the requirements for generating force and power of an electromagnetic active suspension. In this type of suspension, the permanent magnet linear actuator works mechanically parallel to the spring. Additionally, an algorithm was developed to optimize the design. Gysen *et al.* (2010) presented the dynamic capabilities of an electromagnetic suspension that required the merging of a tubular permanent magnet actuator with a spring on a quarter-car setup. Improved stability and manoeuvrability were provided by the implementation of active roll and pitch control during cornering and braking, and road irregularity effects were eliminated, hence increasing both vehicle and passenger safety, and ensuring a comfortable drive. Xue *et al.* (2011) described a model active suspension system made up of an electromagnetic actuator and a mechanical spring. They also examined the effects on performance when there were changes in crucial parameters, such as spring stiffness and actuator force.

4 Mechatronic suspension control

Today, suspension control is considered in a global chassis control framework as it ensures the comfort and safety of passengers in the vehicle. This framework fosters interaction and optimization of combined mechatronic subsystems, namely suspension, braking, and steering systems (Isermann, 2006). Its potential is based on the effects of possible synergy due to the interchange of data and interaction between mechatronic automotive subsystems. This section provides an overview of selected active suspension control methods, including practical suspension control applications in production vehicles and studies on simulated concepts in suspension control.

4.1 Constrained frequency band

Sun *et al.* (2012) presented the frequency band constraints and actuator input delay problems in the active suspension system. They developed a finite-frequency method to overcome the actuator time delay problem using the Kalman-Yakubovich-Popov lemma. This approach was compared to the traditional entire-frequency approach. It was found that this method produced better disturbance attenuation for the chosen frequency range, while adherence to constraints imposed by actual situations was guaranteed in the controller design. Simulations under several types of road disturbance excitations were carried out to verify this approach.

4.2 Look-ahead preview control

The controller for the new generation of Mercedes-Benz ABC systems is equipped with laser scanners to gather preview information on the road profile. The preview approach is able to significantly improve the performance of the suspension system (BOSE, 2010). More control approaches designed for suspensions using preview information including the road profile ahead of the vehicle have been demonstrated in Kim *et al.* (2002), Akbari and Lohmann (2010), and Ryu *et al.* (2011).

In 2008, the concept model Mercedes Benz F700, which features LiDAR-scanners in the vehicle's headlights to scan the road profile in front of the car, was equipped with an enhanced version of the ABC system. This preview information was used in the control algorithm for disturbance feedforward compensation for counteracting road induced vibrations in advance (Voelcker, 2008). The actuators were based on the hydraulic ABC actuators used in production vehicles.

4.3 Predictive control

Shoukry *et al.* (2010) employed generalized predictive control (GPC) to a class of automotive active suspension systems, whereby a digital model of an active suspension was used to tune the GPC controller. The outcome demonstrated ride comfort with an acceptable level of exerted energy.

4.4 Fuzzy logic control

Salem and Aly (2009) and Changizi and Rouhani (2011) used tunable fuzzy logic controllers to maximize passenger comfort in a quarter-car active suspension model. Chang (2007) proposed the use of fuzzy logic controllers for both serial and parallel active suspensions. Simulation results indicate that both types produce good control performance.

Lam *et al.* (2013) described the mechanism of an active hydraulically interconnected suspension (HIS) system which can be used in economy efficient vehicles. The vehicle body roll angle was controlled by implementing a fuzzy logic controller. Experimental implementations showed that the roll angle, with and without active control, had been reduced by approximately 40% and 30% respectively, in comparison to a passive suspension subjected to the same excitation. This study, however, did not address the HIS system energy consumption.

Li *et al.* (2013) implemented an adaptive sliding-mode control approach for a nonlinear active suspension. The Takagi-Sugeno fuzzy logic method was used to handle uncertainties related to the sprung and unsprung masses. The designed controller ensures reachability of the specified switching surface. Sufficient conditions to guarantee asymptotic stability of the specified switching surface have also been established. Simulation results demonstrated the usefulness of this control approach.

Kaleemullah *et al.* (2011) developed a linear control approach that does not require a model of the hydraulic actuator and studied the performance of this method with a fuzzy controller implementation.

4.5 PID control

Ab Talib and Mat Darns (2013) studied an active suspension system for a half-car model using a PID controller. A hydraulic actuator was adopted in this system and three different excitation sources were applied. To obtain the most appropriate PID parameter values, three different tuning methods, namely the Ziegler-Nichols method, heuristic tuning, and the iterative learning algorithm (ILA), were used. The transfer function of the hydraulic actuator was determined using system identification methods. The study revealed that the PID controller tuned by ILA performed better than the other two methods. They concluded that the intelligent ILA is a very good optimization tool to optimize PID controller gains.

Nurhadi (2010) and Sun *et al.* (2010) used PID controllers with different controller tuning methods, revealing that these active suspensions can significantly improve ride performance for quarter-car models.

Sam *et al.* (2005) used proportional integral sliding mode control for a half-car model. The effectiveness and robustness of this approach were proven, and its performance was compared to those of both the linear quadratic regulator and the passive suspension systems.

4.6 Optimal control

Corriga *et al.* (1991) described the optimization of the damper characteristic coefficient and the spring elastic constant for a hydraulic active suspension, such that minimum power is required.

Kumar and Vijayarangan (2006) used linear quadratic optimal control theory to design an active linear quadratic regulator (LQR) system controller. They used two different approaches, the conventional approach and the acceleration dependent approach. Abdalla *et al.* (2008) presented a linear matrix inequality optimized control approach for active suspension systems. Also, Kaleemullah *et al.* (2011) studied an LQR controller used in conjunction with their linear control approach that does not require a physical model of the shock absorber.

4.7 Robust control

A considerable amount of literature has been published on the robustness of mechatronic suspension systems. In one of these studies, Kruczek *et al.* (2010) used an H_∞ controller and compared its performance to those of different controllers for quarter-, half-, and full-car models. The results were verified on a quarter-car test bed. Meanwhile, Jamshidi and Shaabany (2011) introduced H_2 and H_∞ control for active suspension systems. In their research, an order-reduced model of the plant was used in the controller design, but the stability and performance of the nominal closed-loop system were maintained.

Multiobjective H_∞ or mixed H_2/H_∞ control approaches have been used for the operation of frequency dependent filters that shape the frequency response of the controlled system and to achieve robustness against parameter variations (Gao *et al.*, 2006; Zin *et al.*, 2008; Chang *et al.*, 2010; Crews *et al.*, 2011). Kaleemullah *et al.* (2011) used an H_∞ controller in conjunction with their linear control approach.

4.8 Adaptive control

Adaptive control approaches can quickly schedule the controller parametrization in mechatronic suspension systems according to road excitation or the driving state. Karnopp and Margolis (1984), Hać (1987), and Sharp and Crolla (1987) highlighted the advantages of suspension systems that adaptively adjust parameters in this manner. Hać (1987) stated that the controllers that adapt to the driving state have the potential to improve performance more significantly compared to the controllers that adapt to varying plant parameters (as found in classical adaptive control methods).

Venhovens (1993; 1994) introduced a control adaptation approach where the wheel load alters the damping constant in relation to the skyhook (C_{sky}) and a passive damping configuration (C_{S}) adapts to ride safety. This adaptation structure has been used as a basis for adaptation logic in several studies. Venhovens (1993) also considered the concept for adaptive control of an active quarter-car model but did not conduct stability analysis of the switched system. However, Venhovens (1994) could not reproduce the simulation results on a test vehicle mounted on a hydraulic test rig.

Rajamani and Hedrick (1995) developed an adaptive observer designed for the identification of observer-based parameters in a hydraulic active suspension system.

Lin and Ioannis (1997a; 1997b) presented an adaptive nonlinear controller for active suspension systems based on a backstepping design to compensate for nonlinearities of the hydraulic actuator. The regulated output variable is defined as the difference between the chassis displacement and the filtered wheel displacement. The bandwidth of this nonlinear filter is proportional to the suspension deflection. This means that the active suspension setting stiffens if the suspension deflection is stretched to its limits. Otherwise, it focuses on ride comfort. Lin and Ioannis (1997b) extended this method further by an approach that adjusts the shape of the filter nonlinearity. The adjustment depends on the history of the suspension deflection with the aim of providing smoother transitions between the different controller settings. To preserve stability, filter adaptation is realized slowly. This takes place only when the system trajectory is in regions of the state space where nonlinearity is inactive. According to Lin and Ioannis (1997a), the performance improvement in terms of chassis acceleration reduction can reach up to 70% for peak values in singular disturbance events. This is based on the assumption that the suspension components are linear except for the hydraulic actuator which does not receive any measurement feedback.

4.9 Robust adaptive control

Chantranuwathana and Huei (1999) implemented a modified adaptive robust control technique to improve closed-loop stability and performance in the absence of a feedback force sensor. Zhang and Alleyne (2005) considered the dynamic behavior of a hydraulic actuator for model reference adaptive and H_∞ control approaches of an active suspension system. Ramsbottom and Crolla (1999) presented a robust adaptive control approach that involves estimation of the chassis mass and the tyre stiffness parameters. Leite and Peres (2005) examined a pole assignment controller which also enables self-tuning of controller parametrization. Sun *et al.* (2013) proposed a saturated adaptive robust control (ARC) strategy to reduce the effects of uncertainties and possible actuator saturation in an active suspension system. The control strategy was adjusted by adding an antiwindup block. The approach was implemented on a half-car system, taking into consideration a nonlinear primary spring and a piecewise linear damper.

4.10 Switching control

Tran and Hrovat (1993) advocated an adaptive control system which is founded on switching between state feedback controllers depending on dynamic wheel load and suspension deflection. They proposed a heuristic scheduling approach with a brief discussion on a method to guarantee stability of the switched system by a dwell-time formulation with limitations on the switching frequency. Hać (1987) presented a similar approach for the adjustment of controller weights in the event that the suspension deflection nears its limits.

Alleyne and Hedrick (1995) presented a skyhook control approach that involves online estimation of parameters, including the nonlinear and time-varying dynamics of the hydraulic active suspension actuator. A formerly constructed nonlinear sliding control law was used in this system.

Fialho and Balas (2000) presented a linear parameter varying (LPV) control approach to schedule between differently tuned controllers depending on suspension deflection. The scheduling variable, which coordinates controller parameter adaptation, is inferred from the measured suspension deflection signals by means of a static look-up table. Controller adaptation would cause the suspension to turn stiff when suspension deflection becomes critical. In Fialho and Balas (2002), this approach was extended by introducing a second scheduling variable which quantifies road quality, although how this could be obtained from the measurement data was not specified. The controller performance was analyzed through simulations on a linear time-invariant quarter-car model and the nonlinear dynamics of a hydraulic actuator were addressed by a backstepping approach similar to the one described in Lin and Ioannis (1997a).

Zin *et al.* (2008) applied an H_∞ based LPV control methodology to establish robustness against parameter variations in component characteristics and suspension nonlinearities. The scheduling signal is received from a higher level global chassis controller to adapt the suspension to the driving state.

The implementation of multiple model adaptive control (MMAC) on active suspension systems is a relatively unexplored area and only a few studies can be found in the literature. Zhong *et al.* (2010) presented a dynamic-reliable MMAC approach to overcome the challenges in vehicle suspensions and address suspension reliability issues at the same time. In their work, a bank of Kalman filters was used to generate residuals, which were then used by a posterior probability evaluator (PPE) to estimate an uncertain parameter of the suspension within specific intervals. Koch (2011) presented the use of MMAC with LQR controllers to improve ride and handling properties. Stability was ensured by the existence of a common quadratic Lyapunov function. The adaptive approach was based on an adaptation logic derived from the suspension performance requirements. The performance potential of the adaptive switching control structure for a fully active suspension system was analyzed. Koch (2011) considered variations of the root-mean-square (RMS) values of the dynamic wheel load as well as the rapid singular wheel load. It was found that an increase in the rapid singular wheel load is critical as it affects ride safety. A Kalman filter was used with a linear quarter-car model to estimate the dynamic wheel load for adaptation purposes.

5 Conclusions

This paper has provided much insight into advancements and current trends in the research and development of mechatronic suspensions. Active suspension systems provide much flexibility in improving ride and handling characteristics, but high power requirements have limited its practical implementation. Because of this, semi-active systems have been seen as a more pragmatically realizable approach. However, if the power requirements of active suspensions can be significantly reduced, significantly improved performances can be expected.

Hydraulic actuators have thus far been the preferred choice to provide the control action in active suspensions. However, more recently, electromagnetic actuators have started to emerge as a potentially superior alternative, showing improved power consumption and higher bandwidth capabilities.

Active suspension systems that adapt themselves to the driving conditions have been recognized as capable of providing greater improvements in ride and handling performance, in contrast to making the system robust to vehicle parameter variations. For this, look-ahead preview control is being actively developed by motorcar manufacturers. The multiple model adaptive control approach is a control method that is potentially suited to this type of suspension adaptation.

References

Abdalla, M.O., Al Shabatat, N., Al Qaisi, M., 2008. Linear matrix inequality based control of vehicle active suspension system. *Veh. Syst. Dynam.*, **47**(1):121-134.

Ab Talib, M.H., Mat Darns, I.Z., 2013. Self-tuning PID controller for active suspension system with hydraulic actuator. IEEE Symp. on Computers & Informatics, p.86-91. [doi:10.1109/ISCI.2013.6612381]

Akbari, A., Lohmann, B., 2010. Multi-objective H_∞/GH_2 preview control of active vehicle suspensions. *Veh. Syst. Dynam.*, **48**(12):1475-1494. [doi:10.1080/00423110903509327]

Alleyne, A., Hedrick, J.K., 1995. Nonlinear adaptive control of active suspensions. *IEEE Trans. Contr. Syst. Technol.*, **3**(1):94-101. [doi:10.1109/87.370714]

Altet, O., Moreau, X., Oustaloup, A., *et al.*, 2003. The hydractive CRONE suspension: Operation principle and stability study. Proc. Design Engineering Technical Conf. and Computers and Information in Engineering Conf., p.677-683.

Appleyard, M., Wellstead, P.E., 1995. Active suspension: some background. *IEE Proc.-Contr. Theory Appl.*, **142**(2):123-128. [doi:10.1049/ip-cta:19951735]

Becker, M., Jaker, K.P., Fruhauf, F., *et al.*, 1996. Development of an active suspension system for a Mercedes-Benz coach (o404). Proc. IEEE Int. Symp. on Computer-Aided Control System Design, p.146-151.

BOSE, 2010. Resolving the Conflict Between Comfort and Control. Available from http://www.boseindia.com/the-bose-suspension-system/

Chang, J.C., 2007. Analysis of series type and parallel type active suspension systems. Proc. Ta Hwa Institute of Technology Int. Conf., p.177-182.

Chang, Y.C., Kuo, L.W., Wu, J.L., 2010. Reliable multi-objective decentralized controller design. Int. Conf. on System Science and Engineering, p.227-232. [doi:10.1109/ICSSE.2010.5551749]

Changizi, N., Rouhani, M., 2011. Comparing PID and fuzzy logic control a quarter car suspension system. *J. Math. Comput. Sci.*, **2**(3):559-564.

Chantranuwathana, S., Huei, P., 1999. Adaptive robust force control for vehicle active suspension. Proc. IEEE Int. Conf. on Control Applications, p.442-447.

Chung, S., Shin, H., 2004. High-voltage power supply for semi-active suspension system with ER-fluid damper. *IEEE Trans. Veh. Technol.*, **53**(1):206-214. [doi:10.1109/TVT.2003.819811]

Codeca, F., Savaresi, S.M., Spelta, C., *et al.*, 2008. Identification of an electro-hydraulic controllable shock absorber using black-block non-linear models. Proc. IEEE Int. Conf. on Control Applications, p.462-467.

Corriga, G., Sanna, S., Usai, G., 1991. An optimal tandem active-passive suspension system for road vehicles with minimum power consumption. *IEEE Trans. Ind. Electron.*, **38**(3):210-216. [doi:10.1109/41.87589]

Crews, J.H., Mattson, M.G., Buckner, G.D., 2011. Multi-objective control optimization for semi-active vehicle suspensions. *J. Sound Vibr.*, **330**(23):5502-5516. [doi:10.1016/j.jsv.2011.05.036]

Delphi, 2005. Delphi Magneride. Available from http://www.motor-talk.de/forum/aktion/attachment.html?attachmentid=488981 [Accessed on May 22, 2011].

Efatpenah, K., Beno, J.H., Nichols, S.P., 2000. Energy requirements of a passive and an electromechanical active suspension system. *Veh. Syst. Dynam.*, **34**(6):437-458. [doi:10.1076/vesd.34.6.437.2050]

Elmadany, M.M., Qarmoush, A.O., 2011. Dynamic analysis of a slow-active suspension system based on a full car model. *J. Vibr. Contr.*, **17**(1):39-53. [doi:10.1177/1077546309352828]

Fialho, I.J., Balas, G.J., 2000. Design of nonlinear controllers for active vehicle suspensions using parameter-varying control synthesis. *Veh. Syst. Dynam.*, **33**(5):351-370. [doi:10.1076/0042-3114(200005)33:5;1-Q;FT351]

Fialho, I.J., Balas, G.J., 2002. Road adaptive active suspension design using linear parameter-varying gain-scheduling. *IEEE Trans. Contr. Syst. Technol.*, **10**(1):43-54. [doi:10.1109/87.974337]

Fijalkowski, B.T., 2011. Automotive Mechatronics: Operational and Practical. Springer, London.

Fischer, D., Isermann, R., 2004. Mechatronic semi-active and active vehicle suspensions. *Contr. Eng. Pract.*, **12**(11):1353-1367. [doi:10.1016/j.conengprac.2003.08.003]

Gao, G.S., Yang, S.P., 2006. Semi-active control performance of railway vehicle suspension featuring magnetorheological dampers. Proc. 1st IEEE Conf. on Industrial Electronics and Applications, p.1-5.

Gao, H., Lam, J., Wang, C., 2006. Multi-objective control of vehicle active suspension systems via load-dependent controllers. *J. Sound Vibr.*, **290**(3-5):654-675. [doi:10.1016/j.jsv.2005.04.007]

Gavriloski, V., Danev, D., Angushev, K., 2007. Mechatronic approach in vehicle suspension system design. *World Congr.*, **12**:45-58.

Genger, C., 2009. Active and Semi-active Suspension Control for Specific Point Isolation of Vehicles. MS Thesis, University of California, Davis, USA.

Giorgetti, N., Bemporad, A., Tseng, H.E., *et al.*, 2005. Hybrid model predictive control application towards optimal semi-active suspension. Proc. IEEE Int. Symp. on Industrial Electronics, p.391-398.

Graves, K.E., Iovenitti, P.G., Toncich, D., 2000. Electromagnetic regenerative damping in vehicle suspension systems. *Int. J. Veh. Des.*, **24**(2/3):182-197. [doi:10.1504/IJVD.2000.005181]

Guglielmino, E., Sireteanu, T., Stammers, C.W., *et al.*, 2010. Semi-active Suspension Control: Improved Vehicle Ride and Road Friendliness. Springer, London.

Gysen, B.L.J., Janssen, J.L.G., Paulides, J.J.H., *et al.*, 2009. Design aspects of an active electromagnetic suspension system for automotive applications. *IEEE Trans. Ind. Appl.*, **45**(5):1589-1597. [doi:10.1109/TIA.2009.2027097]

Gysen, B.L.J., Paulides, J.J.H., Janssen, J.L.G., *et al.*, 2010. Active electromagnetic suspension system for improved vehicle dynamics. *IEEE Trans. Veh. Technol.*, **59**(3):1156-1163. [doi:10.1109/TVT.2009.2038706]

Hać, A., 1987. Adaptive control of vehicle suspension. *Veh. Syst. Dynam.*, **16**(2):57-74. [doi:10.1080/00423118708968870]

Heiring, B., Ersoy, M., 2011. Chassis Handbook: Fundamental, Driving Dynamics, Mechatronics, Perspectives. Springer, p.590.

Hrovat, D., 1997. Survey of advanced suspension developments and related optimal control applications. *Automatica*, **33**(10):1781-1817. [doi:10.1016/S0005-1098(97)00101-5]

Isermann, R., 2006. Automotive mechatronic systems—general developments and examples. *Automatisierungstechnik*, **54**(9):419-429.

Jamshidi, F., Shaabany, A., 2011. Robust control of an active suspension system using H_2 & H_∞ control methods. *J. Am. Sci.*, **7**(5):1-5.

Jonasson, M., Roos, F., 2008. Design and evaluation of an active electromechanical wheel suspension system. *Mechatronics*, **18**(4):218-230. [doi:10.1016/j.mechatronics.2007.11.003]

Jones, W.D., 2005. Easy ride: Bose Corp. uses speaker technology to give cars adaptive suspension. *IEEE Spectrum*, **42**(5):12-14.

Kaleemullah, M., Faris, W.F., Hasbullah, F., 2011. Design of robust H_∞, fuzzy and LQR controller for active suspension of a quarter car model. 4th Int. Conf. on Mechatronics, p.1-6.

Karnopp, D., 1983. Active damping in road vehicle suspension systems. *Veh. Syst. Dynam.*, **12**(6):291-311. [doi:10.1080/00423118308968758]

Karnopp, D., 1986. Theoretical limitations in active vehicle suspensions. *Veh. Syst. Dynam.*, **15**(1):41-54. [doi:10.1080/00423118608968839]

Karnopp, D., Margolis, D., 1984. Adaptive suspension concepts for road vehicles. *Veh. Syst. Dynam.*, **13**(3):145-160. [doi:10.1080/00423118408968772]

Karnopp, D.C., Crosby, M.J., Harwood, R.A., 1974. Vibration control using semiactive force generators. *ASME J. Eng. Ind.*, **96**(2):619-626. [doi:10.1115/1.3438373]

Kim, H., Hyun, S.Y., Park, Y., 2002. Improving the vehicle performance with active suspension using road-sensing algorithm. *Comput. Struct.*, **80**(18-19):1569-1577. [doi:10.1016/S0045-7949(02)00110-4]

Koch, G.P.A., 2011. Adaptive Control of Mechatronic Vehicle Suspension. PhD Thesis, Technical University of Munich, Munich, Germany.

Kruczek, A., Stribrsky, A., Honc, J., *et al.*, 2010. Controller choice for car active suspension. *Int. J. Mech.*, **3**(4):61-68.

Kumar, M.S., Vijayarangan, S., 2006. Design of LQR controller for active suspension system. *Ind. J. Eng. Mater. Sci.*, **13**:173-179.

Lai, C.Y., Liao, W.H., 2002. Vibration control of a suspension system via a magnetorheological fluid damper. *J. Vibr. Contr.*, **8(4)**:527-547. [doi:10.1177/107754602023712]

Lam, Q., Wang, L., Zhang, N., 2013. Experimental implimentation of a fuzzy controller for an active hydraulically interconnected suspension on a sport utility vehicle. IEEE Intelligent Vehicles Symp., p.383-390. [doi:10.1109/IVS.2013.6629499]

Leite, V.J.S., Peres, P.L.D., 2005. Pole location control design of an active suspension system with uncertain parameters. *Veh. Syst. Dynam.*, **43**(8):561-579. [doi:10.1080/00423110420002667O2]

Li, H., Yu, J., Hilton, C., *et al.*, 2013. Adaptive sliding-mode control for nonlinear active suspension vehicle systems using T-S fuzzy approach. *IEEE Trans. Ind. Electron.*, **60**(8):3328-3338. [doi:10.1109/TIE.2012.2202354]

Lin, J.S., Ioannis, K., 1997a. Nonlinear design of active suspensions. *IEEE Contr. Syst.*, **17**(3):45-59. [doi:10.1109/37.588129]

Lin, J.S., Ioannis, K., 1997b. Road-adaptive nonlinear design of active suspensions. Proc. American Control Conf., p.714-718.

Liu, H.M., Mok, T.K., Li, Y., *et al.*, 2005. Comparison between GA and gradient descent algorithm in parameter optimization of UPFC fuzzy damping controller. *Electr. Power Autom. Equipment*, **25**(11):5-10.

Liu, L., Wang, B., 2008. Multi objective robust active vibration control for flexure jointed struts of stewart platforms via H_∞ and μ synthesis. *Chin. J. Aeronaut.*, **21**(2):125-133. [doi:10.1016/S1000-9361(08)60016-3]

Martins, I., Esteves, M., da Silva, F.P., *et al.*, 1999. Electromagnetic hybrid active-passive vehicle suspension system. Proc. IEEE 49th Vehicular Technology Conf., p.125-133.

Martins, I., Esteves, J., Marques, G.D., *et al.*, 2006. Permanent-magnets linear actuators applicability in automobile active suspensions. *IEEE Trans. Veh. Technol.*, **55**(1):86-94. [doi:10.1109/TVT.2005.861167]

Möller, N., 2009. Porsche Engineering Magazine. Porsche Engineering, Germany, p.28.

Nagai, M., 1993. Recent researches on active suspensions for ground vehicles. *JSME Int. J. Ser. C*, **36**(2):161-170.

Nurhadi, H., 2010. Study on Control of Bus Suspension System. Department of Mechanical Engineering, Institut Teknologi Sepuluh Nopember (ITS).

Paulides, J.J.H., Encica, L., Lomonova, E.A., *et al.*, 2006a. Active roll compensation for automotive applications using a brushless direct-drive linear permanent magnet actuator. 37th IEEE Power Electronics Specialists Conf., p.1-6. [doi:10.1109/PESC.2006.1712068]

Paulides, J.J.H., Encica, L., Lomonova, E.A., *et al.*, 2006b. Design considerations for a semi-active electromagnetic suspension system. *IEEE Trans. Magnet.*, **42**(10):3446-3448. [doi:10.1109/TMAG.2006.879963]

Peter, H., 2012. ABC Active Body Control / MBC Magic Body Control. Available from http://500sec.com/abc-active-body-control-mbc-magic-body-control/ [Accessed on Apr. 12, 2011].

Pyper, M., Schiffer, W., Schneider, W., 2003. ABC—Active Body Control. Landsberg/Lech: Verl. Moderne Industrie.

Rajamani, R., 2011. Vehicle Dynamics and Control. Springer, p.325-355.

Rajamani, R., Hedrick, J.K., 1995. Adaptive observers for active automotive suspensions: theory and experiment. *IEEE Trans. Contr. Syst. Technol.*, **3**(1):86-93. [doi:10.1109/87.370713]

Ramsbottom, M., Crolla, D.A., 1999. Simulation of an adaptive controller for a limited bandwidth active suspension. *Int. J. Veh. Des.*, **21**(4/5):355-371. [doi:10.1504/IJVD.1999.005590]

Ryu, S., Park, Y., Suh, M., 2011. Ride quality analysis of a tracked vehicle suspension with a preview control. *J. Terramech.*, **48**(6):409-417. [doi:10.1016/j.jterra.2011.09.002]

Salem, M., Aly, A.A., 2009. Fuzzy control of a quarter-car suspension system. *World Acad. Sci.*, **5**:258-263.

Sam, Y.M., Hudha, K., 2006. Modelling and force tracking control of hydraulic actuator for an active suspension system. 1st IEEE Conf. on Industrial Electronics and Applications, p.1-6.

Sam, Y.M., Shah, J.H., Osman, B., 2005. Modeling and control of the active suspension system using proportional integral sliding mode approach. *Asian J. Contr.*, **7**(2):91-98. [doi:10.1111/j.1934-6093.2005.tb00378.x]

Savaresi, S., Poussot-Vassal, C., Spelta, C., *et al.*, 2010. Semi-active Suspension Control Design for Vehicles. Butterworth-Heinemann, Amsterdam, the Netherlands.

Sharp, R.S., Crolla, D.A., 1987. Road vehicle system design—a review. *Veh. Syst. Dynam.*, **16**(3):167-192. [doi:10.1080/00423118708968877]

Shoukry, Y., El-Shafie, M., Hammad, S., 2010. Networked embedded generalized predictive controller for an active suspension system. Proc. American Control Conf., p.4570-4575.

Smith, M.C., Walker, G.W., 2000. Performance limitations and constraints for active and passive suspensions: a mechanical multi-port approach. *Veh. Syst. Dynam.*, **33**(3):137-168. [doi:10.1076/0042-3114(200003)33:3;1-Y;FT137]

Strassberger, M., Guldner, J., 2004. BMW's dynamic drive: an active stabilizer bar system. *IEEE Contr. Syst.*, **24**(4):28-29, 107. [doi:10.1109/MCS.2004.1316650]

Stribrsky, A., Hyniova, K., Honcu, J., *et al.*, 2007. Energy recuperation in automotive active suspension systems with linear electric motor. Mediterranean Conf. on Control and Automation, p.1-5. [doi:10.1109/MED.2007.4433870]

Sun, J., Wang, Y., Liang, H., 2010. Comparative study on vibration control of engineering vehicle suspension system. Int. Conf. on Intelligent Computation Technology and Automation, p.989-992. [doi:10.1109/ICICTA.2010.669]

Sun, W., Zhao, Y., Li, J., *et al.*, 2012. Active suspension control with frequency band constraints and actuator input delay. *IEEE Trans. Ind. Electron.*, **59**(1):530-537. [doi:10.1109/TIE.2011.2134057]

Sun, W., Zhao, Z., Gao, H., 2013. Saturated adaptive robust control for active suspension systems. *IEEE Trans. Ind. Electron.*, **60**(9):3889-3896. [doi:10.1109/TIE.2012.2206340]

Tran, M.N., Hrovat, D., 1993. Application of gain scheduling to design of active suspensions. Proc. 32nd IEEE Decision and Control Conf., p.1030-1035.

Venhovens, P.J.T., 1993. Optimal Control of Vehicle Suspensions. PhD Thesis, Delft University of Technology, Delft, the Netherlands.

Venhovens, P.J.T., 1994. The development and implementation of adaptive semi-active suspension control. *Veh. Syst. Dynam.*, **23**(1):211-235. [doi:10.1080/00423119408969057]

Voelcker, J., 2008. The soul of a new Mercedes. *IEEE Spectrum*, **45**(12):36-41. [doi:10.1109/MSPEC.2008.4687367]

Williams, R.A., Best, A., 1994. Control of a low frequency active suspension. Int. Conf. on Control, p.338-343. [doi:10.1049/cp:19940155]

Wu, W., Ma, L., Yang, Q., 2011. Active optimal control for multi-dim vibration damping device based on parallel mechanism. 2nd Int. Conf. on Mechanic Automation and Control Engineering, p.5531-5534.

Xu, L., Guo, X., 2010. Hydraulic transmission electromagnetic energy-regenerative active suspension and its working principle. 2nd Int. Workshop on Intelligent Systems and Applications, p.1-5.

Xue, X.D., Cheng, K.W.E., Zhang, Z., *et al.*, 2011. Investigation on parameters of automotive electromagnetic active suspensions. 4th Int. Conf. on Power Electronics Systems and Applications, p.1-5.

Yamashita, M., Fujimori, K., Uhlik, C., *et al.*, 1990. H_∞ control of an automotive active suspension. Proc. 29th IEEE Conf. on Decision and Control, p.2244-2250. [doi:10.1109/CDC.1990.204024]

Zhang, Y., Alleyne, A., 2005. A practical and effective approach to active suspension control. *Veh. Syst. Dynam.*, **43**(5):305-330. [doi:10.1080/00423110412331282869]

Zhong, X., Ichchou, M., Gillot, F., *et al.*, 2010. A dynamic-reliable multiple model adaptive controller for active vehicle suspension under uncertainties. *Smart Mater. Struct.*, **19**(4):045007. [doi:10.1088/0964-1726/19/4/045007]

Zin, A., Sename, O., Gaspar, P., *et al.*, 2008. Robust LPV-H_∞ control for active suspensions with performance adaptation in view of global chassis control. *Veh. Syst. Dynam.*, **46**(10):889-912. [doi:10.1080/00423110701684587]