

Active Electromagnetic Suspension System for Improved Vehicle Dynamics

Bart L. J. Gysen, *Member, IEEE*, Johannes J. H. Paulides, *Member, IEEE*,
Jeroen L. G. Janssen, *Member, IEEE*, and Elena A. Lomonova, *Fellow, IEEE*

Abstract—This paper offers motivations for an electromagnetic active suspension system that provides both additional stability and maneuverability by performing active roll and pitch control during cornering and braking, as well as eliminating road irregularities, hence increasing both vehicle and passenger safety and drive comfort. Various technologies are compared with the proposed electromagnetic suspension system that uses a tubular permanent-magnet actuator (TPMA) with a passive spring. Based on on-road measurements and results from the literature, several specifications for the design of an electromagnetic suspension system are derived. The measured on-road movement of the passive suspension system is reproduced by electromagnetic actuation on a quarter car setup, proving the dynamic capabilities of an electromagnetic suspension system.

Index Terms—Active suspension, permanent magnet (PM), tubular actuator.

I. INTRODUCTION

ADVANCED electromechanical and electronic systems are used to influence the dynamic performance of the vehicle, for example, antilock braking systems, electronic brake force distribution, electronic stability program, etc. These systems improve vehicle handling and passenger safety, since this becomes an ever-increasing demand for the automotive industry, particularly when cars tend to become smaller (SMART) and incorporate a higher center of gravity (sport utility vehicle) and a reduced footprint. For instance, the Transportation Research Board [1] reported that 51% of serious car accidents are caused by rollover. Another trend in the automotive industry is the “more electric car,” e.g., the Toyota Prius. These hybrid vehicles combine the efficiency of an electric motor and an internal combustion engine. Due to the global increase in oil prices and the importance of environmental sustainability, the “full electric car” is also gaining attention. Recently, a Dutch energy company (Essent), together with Electric Car Europe [2], has launched a commercial electric car. These cars have a totally different weight distribution since the combustion engine is replaced by an electric motor and a battery pack that weighs around 400 kg. As such, the optimal electrical drive train efficiency is reached when completely incorporated in the wheel

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The authors are with the Department of Electrical Engineering, Eindhoven University of Technology, 5600 Eindhoven, The Netherlands (e-mail: b.l.j.gysen@tue.nl).

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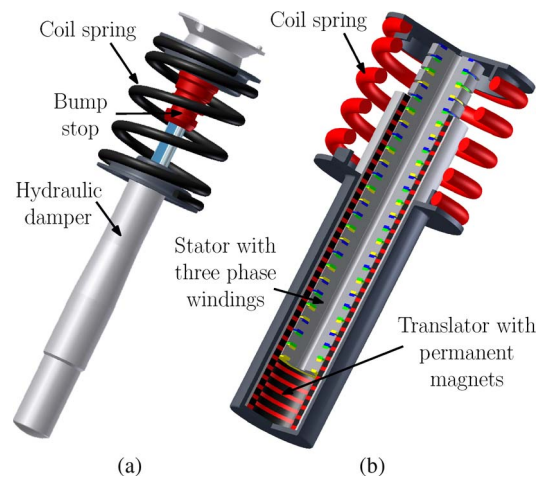


Fig. 1. (a) Conventional passive suspension system. (b) Electromagnetic suspension system.

ciency is reached when completely incorporated in the wheel [3], [4]. However, as a result, the unsprung mass of the vehicle increases, which is a disadvantage with regard to passenger comfort and handling. These trends clearly show the need of active suspension to be incorporated into vehicles. These systems allow for greater suspension articulation when driving under low-yaw circumstances (driving relatively straight) to absorb road irregularities and have a much more rigid response when the car is driven through turns, which improves vehicle dynamics. To facilitate the ideal suspension with regard to comfort, handling, and safety, various commercial technologies are used to improve or replace the conventional passive suspension system shown in Fig. 1(a). This paper discusses an active electromagnetic suspension system incorporating a brushless tubular permanent-magnet actuator (TPMA) in parallel with a mechanical spring [5]–[7], as illustrated in Fig. 1(b). This topology has been chosen since the tubular structure exhibits high efficiency and excellent servo characteristics. Furthermore, the mechanical spring supports the sprung mass; hence, no continuous power is needed. The main advantage of this system is that it simultaneously allows for both the elimination of the road disturbances and active roll and pitch control.

In Section II, the ideal suspension with regard to passenger comfort is discussed. Section III gives an overview of different technologies for active suspension, which are compared with the proposed electromagnetic suspension system. Based on the quarter car model, the influence of changing the sprung and unsprung masses is shown in Section IV. Various on- and

off-road measurements are performed on the current passive suspension system, and specifications with regard to force, power, and stroke are derived for the active suspension system in Section V. Electromagnetic actuation is performed on a quarter car test setup, which mimics the on-road measurements, and conclusions are drawn in Section VI.

II. IDEAL SUSPENSION

The ideal automotive suspension would rapidly independently absorb road shocks and would slowly return to its normal position while maintaining optimal tire-to-road contact. However, this is difficult to passively achieve, where a soft spring allows for too much movement and a hard spring causes passenger discomfort due to road irregularities. Passenger comfort (combined with handling and safety) is an ever-increasing demand, where everybody expects ever-improving comfort and handling from the automotive industry. Even though a dearth of publications exists with regard to passenger comfort, it was assumed that general comfort is improved when the following conditions are minimized:

- 1) motion sickness: ~ 1 Hz [8];
- 2) head toss: ~ 2 –8 Hz [9].

A. Motion Sickness

Motion sickness, particularly when reading, is a common by-product of exposure to optical depictions of inertial motion [12]. This phenomenon, which is called visually induced motion sickness, has been reported in a variety of virtual environments, such as fixed-base flight and automobile simulators [13]–[15]. Furthermore, Gahlinger [16] discussed that motion sickness most commonly occurs with acceleration in a direction perpendicular to the longitudinal axis of the body, which is why head movements away from the direction of motion are very provocative. He further mentioned that vertical oscillatory motion (appropriately called heave) at a frequency of 0.2 Hz is most likely to cause motion sickness, although the incidence of motion sickness quite rapidly falls at higher frequencies. This results in the design criteria for active systems wherein frequencies lower than 1 Hz need to be eliminated. This is underlined by surveys documenting that motion sickness occurs in 58% of children [8].

B. Head Toss

Head toss happens when a car makes a sudden roll motion, e.g., occurring when one tire drives through a deep hole. This is not due to optical depictions; rather, it happens because the receptor mechanisms of the three orthogonally oriented canals in each inner ear are activated by angular acceleration of the head [16]. This particularly occurs when a suspension with coupled left and right wheels is used, as is the case with passive antiroll bars. At frequencies below 1–2 Hz, the head moved with the body, but in the frequency range of 2–8 Hz, the amplitude of head acceleration is augmented, indicating that oscillation about a center of rotation low in the body may induce large angular movements in this frequency range because of the linear component of acceleration delivered at

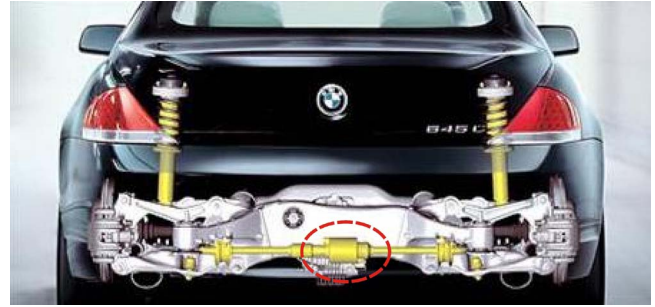


Fig. 2. ARC system of BMW using a hydraulic rotary actuator [10].

the cervical vertebrae. At higher frequencies, the acceleration at the head was attenuated with an associated increase in phase lag, probably due to the absorption of input acceleration by the upper torso [17].

To quantify the acceptable acceleration levels, the ISO 2631 standard is often used as a reference set of root-mean-square (RMS) accelerations, which produce equal fatigue-decreased proficiency [18]. Hence, the ideal suspension system should minimize the frequency response of the sprung mass accelerations to the road disturbances in the band between 0.2 and 10 Hz while maintaining a stiff ride during cornering. However, one of the main problems of an active suspension system is the absence of a fixed reference position, and hence, only relative displacements can be measured. Next to this, it is difficult to distinguish the situation of roll during cornering and the condition where the right wheel experiences a different bump followed by the left wheel. Therefore, different sensor inputs, e.g., position, speed, acceleration, force, and roll angle, are preferred, where based on these measurements, the exact state of the vehicle can be estimated to control the active suspension system [19], [20].

III. ACTIVE SUSPENSIONS

A. Hydraulic Systems

Due to the high force density, ease of design, maturity of technology, and commercial availability of the various parts, hydraulic systems are commonly used in body control systems. As an example, BMW has recently developed an antiroll control (BMW-ARC) system by placing a hydraulic rotary actuator in the center of the antiroll bar at the rear of the vehicle [10], as shown in Fig. 2. Another example is given by the active body control system of Mercedes [21], which uses high-pressure hydraulics to prestress the spring, hence generating antiroll forces without coupling the left and right wheels (as in the case of an antiroll bar). All commercial body control systems use hydraulics to provide the active suspension system to improve vehicle roll behavior and ride control, where the main advantages of the hydraulic system are as follows:

- 1) very high force density;
- 2) ease of control;
- 3) ease of design;
- 4) commercial availability of the various parts;
- 5) reliability;
- 6) commercial maturity.

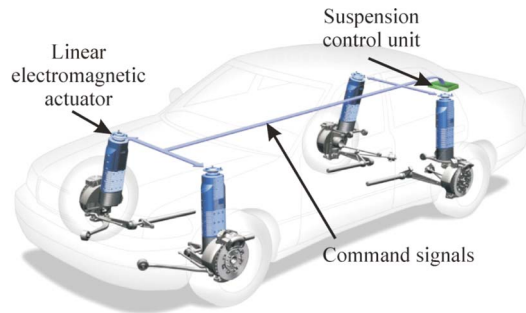


Fig. 3. Bose suspension system [11].

The main disadvantages of the hydraulic system are as follows:

- 1) considered inefficient due to the required continuously pressurized system;
- 2) relatively high system time constant (pressure loss and flexible hoses);
- 3) environmental pollution due to hose leaks and ruptures, where hydraulic fluids are toxic;
- 4) mass and intractable space requirements of the total system, including supply system, even though it mainly contributes to the sprung mass.

Hydraulic systems already proved their potential in commercial systems with regard to active roll control (ARC) since the bandwidth requirement is very small (order of hertz); however, concerning reduction of road vibrations, the performance of the hydraulic system is insufficient.

B. Electromagnetic Systems

An electromagnetic suspension system could counter the disadvantages of a hydraulic system due to the relatively high bandwidth (tens of hertz), and there is no need for continuous power, ease of control, and absence of fluids. Linear motion can be achieved by an electric rotary motor with a ball screw or other transducers to transform rotary motion to linear translation. However, the mechanism required to make this conversion introduces significant complications to the system. These complications include backlash and increased mass of the moving part due to connecting transducers or gears that convert rotary motion to linear motion (enabling active suspension). More important, they also introduce infinite inertia, and therefore, a series suspension, e.g., where electromagnetic actuation is represented by a rotary motor connected to a ball screw bearing, is preferable. These direct-drive electromagnetic systems are more suited to a parallel suspension, where the inertia of the actuator is minimized.

Recently, a system has been presented, namely, the Bose suspension system [11], as shown in Fig. 3, which includes a linear electromagnetic motor and a power amplifier at each wheel, and a set of control algorithms. In this system, the high-bandwidth linear electromagnetic motor is installed at each wheel. This linear electromagnetic motor responds quickly enough to counter the effects of bumps and potholes while maintaining a comfortable ride. Additionally, the motor has been designed for maximum strength in a small package, allowing it to put out

enough force to prevent the car from rolling and pitching during aggressive driving maneuvers. Electrical power is delivered to the motor by a power amplifier in response to signals from the control algorithms. The bidirectional power amplifier allows power to flow into the linear electromagnetic motor and allows power to be returned from the motor. For example, when the suspension encounters a pothole, power is used to extend the motor and isolate the vehicle's occupants from the disturbance. On the far side of the pothole, the motor operates as a generator and returns the power back through the amplifier. It is attained that this suspension system requires less than a third of the power of a typical vehicle's air-conditioning system, i.e., hundreds of watts. Compared with hydraulic actuators, the main advantages of electromagnetic actuators are as follows:

- 1) increased efficiency;
- 2) improved dynamic behavior;
- 3) stability improvement;
- 4) accurate force control;
- 5) dual operation of the actuator.

The disadvantages are as follows:

- 1) increased volume of the suspension, since the force density of the active part of hydraulics is higher than for electromagnetic actuation, i.e., system mass and volume could be less;
- 2) relatively high current for a 12- to 14-V system;
- 3) conventional designs that need excitation to provide a continuous force;
- 4) higher system costs.

Although numerous linear motor topologies exist, the permanent-magnet (PM) synchronous linear actuator is investigated since it offers a high permissible power density at an ever-decreasing cost penalty. More specifically, a tubular PM synchronous actuator, as shown in Fig. 1(b), is preferred since this actuator inhibits the highest force density, where various different topologies are

- a) ironless (no attraction forces);
- b) ironless with back-iron, higher force density compared with (a);
- c) slotted with soft magnetic powder composite materials (low saturation level);
- d) laminated (difficult to achieve but higher dynamic capability).

The actuator topology achieving the highest force density is (d); however, (b) is more preferred with regard to manufacturing. Various magnetization patterns are possible, such as

- 1) radially magnetized north and south poles;
- 2) axially magnetized north and south poles with iron poles (no back-iron);
- 3) Halbach array (no back-iron).

In [7], a slotless tubular actuator is optimized for the mean output force for all these magnetization patterns and for interior (moving magnet) and exterior (moving coil) magnet topologies. It has been shown that exterior Halbach magnetization offers the highest output force within the volume constraints given by the BMW 530i. Equivalent conclusions were drawn for the slotted topology in [22]; however, a higher force density is obtained than in [7].

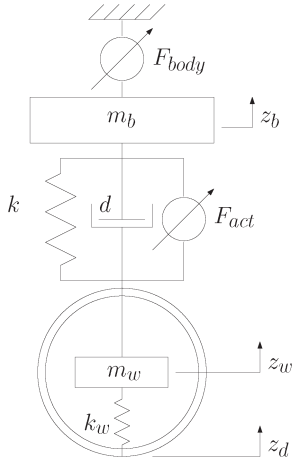


Fig. 4. Quarter car model, including body disturbances and active suspension.

 TABLE I
NOMINAL PARAMETERS OF THE QUARTER CAR MODEL

Parameter	Value	Description
k	30 kN/m	Passive spring stiffness
k_w	160 kN/m	Tire stiffness
d	1200 Ns/m	Passive damping constant
m_b	450 kg	Quarter sprung mass
m_w	40 kg	Unsprung mass

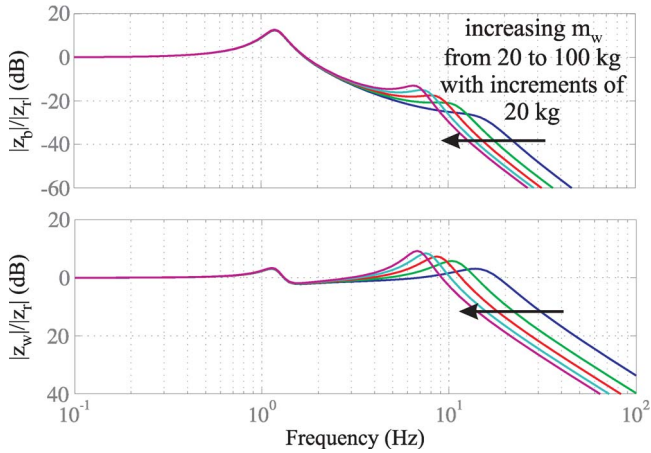


Fig. 5. Bode diagrams of the sprung and unsprung mass responses to road disturbances for increasing unsprung mass.

IV. SYSTEM MODELING

In general, a full car model [23] is preferred to model the dynamic behavior of the suspension system; however, roll and pitch behavior can also be modeled as an equivalent disturbance force acting on the body mass, i.e., F_{body} . Hence, for the scope of this paper, a quarter car model, as shown in Fig. 4, is used with the parameters shown in Table I. This model allows, for example, the investigation of the increase or decrease in the respective sprung and unsprung masses on the response of body height z_b and wheel height z_w to road disturbances z_r . From these Bode diagrams, it can be observed that increasing the unsprung mass, as shown in Fig. 5 (for a wheel motor design [3], [4]), and decreasing the sprung mass, as shown in Fig. 6 (the more electrical car), or increasing the unsprung-to-sprung mass ratio, leads to an increase in response of the frequency range

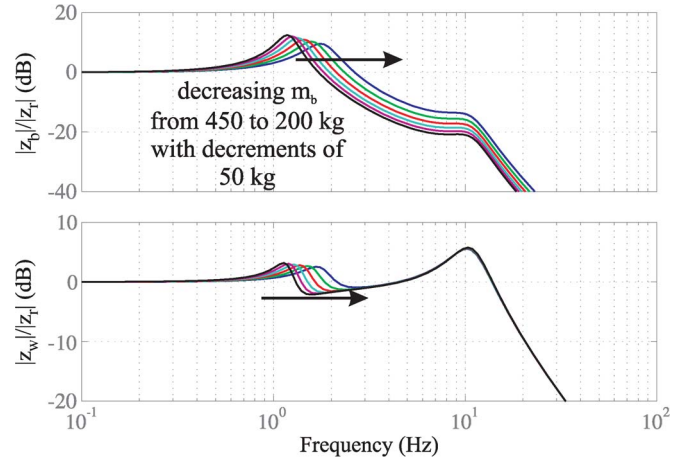


Fig. 6. Bode diagrams of the sprung and unsprung mass responses to road disturbances for decreasing sprung mass.

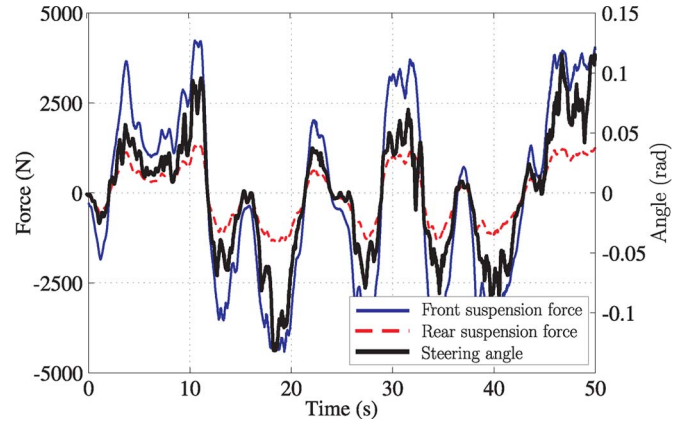


Fig. 7. Time interval of the derived suspension forces from acceleration measurements on the Nürburgring, together with the steering angle.

between 2 and 10 Hz, which decreases the isolation of road disturbances and comfort, as explained in Section II. Hence, the suspension system should be designed for maximum sprung-to-unsprung mass ratio while minimizing total mass.

V. SYSTEM SPECIFICATIONS

A. Roll and Pitch Behavior

During high-speed cornering, braking, and accelerating, roll and pitch forces tend to turn the body around the roll and pitch axes. As a result, the total weight is not evenly distributed along the four wheels, which increases the instability and could lead to tip over of the vehicle during cornering [24]. To have an indication of the particular roll forces during high-speed cornering, a test drive with a BMW 530i is performed on the Nürburgring in Germany. The vertical acceleration of the sprung mass is measured, and the resulting roll forces are calculated; an extensive analysis is given in [7]. A time interval of the calculated forces deducted from the measurements, together with the steering angle, is shown in Fig. 7. During calculation, a front-to-total force ratio of 0.7 is taken into account to design for safer understeer behavior. It can be observed that a peak force of 4 kN is necessary for the front actuators. Furthermore,

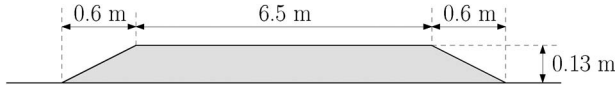
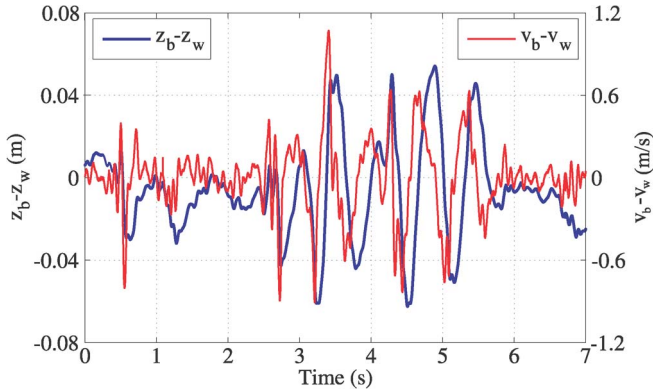


Fig. 8. Sizes of the bump on the test track.

TABLE II
STROKE AND SPEED MEASUREMENT RESULTS

	Max. bound	Max. rebound	Mean bound	Mean rebound
Stroke	80 mm	58 mm	4.5 mm	3.4 mm
Speed	1.28 m/s	2.25 m/s	38.5 mm/s	38.4 mm/s

Fig. 9. Position ($z_b - z_w$) and speed ($v_b - v_w$) measurements while driving on the bump of Fig. 8.

a mean force of 2 kN is measured; however, to determine the mean force specification, a duty cycle has to be taken into account since the Nürburgring does not represent normal driving and road conditions.

B. Road Disturbances

A further test drive is performed on a more common road with bumps and potholes. During this measurement, the relative vertical position between the sprung and the unsprung mass is measured with an optical sensor. This sensor is aligned with the passive spring and damper; hence, the stroke is directly measured. The speed is then derived, where a small time interval is shown in Fig. 9, where, at that moment (2.5 s), the bump, as shown in Fig. 8, is hit at 35 km/h. From these measurements, the stroke, speed, and force requirements of the suspension system can be derived, which are given in Table II. However, first, the characteristics of the passive spring, including the bump stop and damper, need to be measured, e.g., using a standard Verband Der Automobilindustrie (VDA) test. This measurement is supplemented with additional points, as the maximum velocity in the VDA test is limited to 1.05 m/s. Although this limit is sufficient for normal road behavior, when very steep bumps are hit, as shown in Fig. 9, the velocity increases beyond this point. The measured spring and damper characteristics are shown in Figs. 10 and 11, respectively. Using the on-road speed measurement and the off-road measured damping characteristic, the absorbed power of the hydraulic damper can be calculated. An instantaneous peak damping power of 2 kW is necessary when driving onto the bump shown in Fig. 8; however, when taking the average value of the total driving cycle, only a power level of 16 W per damper is necessary, which is

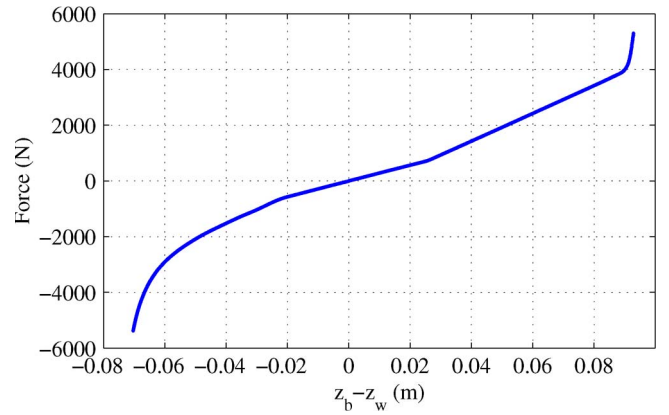


Fig. 10. Measured spring characteristic of the passive suspension.

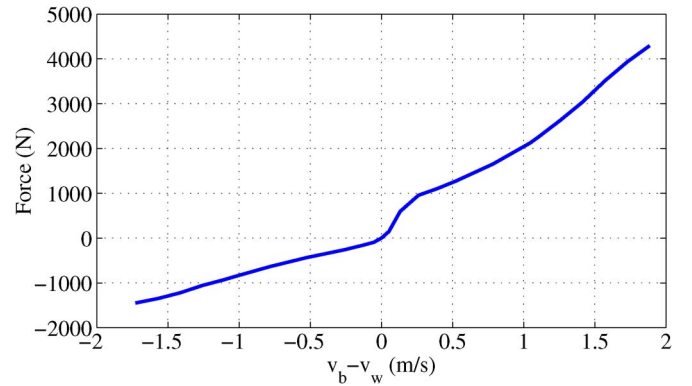


Fig. 11. Measured damper characteristic of the passive suspension.

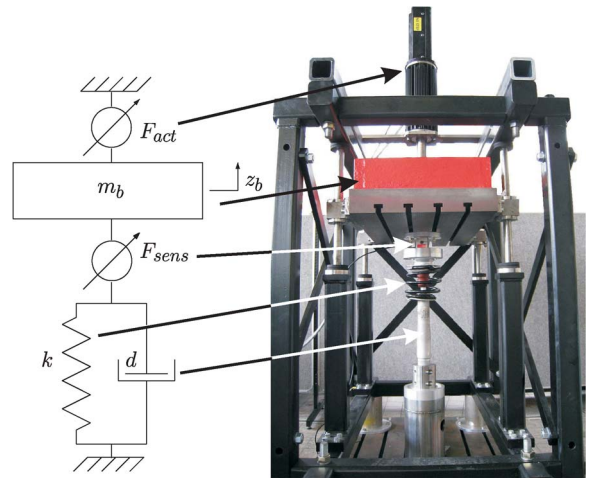


Fig. 12. Quarter car test setup.

comparable with the results obtained in [25] for normal city driving.

VI. QUARTER CAR SETUP

In this section, the on-road measurements will be reproduced by means of electromagnetic actuation on a quarter car test setup shown in Fig. 12. The setup consists of a single moving mass (hence, wheel dynamics are neglected), together with the passive suspension of a BMW 530i, and a three-phase brushless TPMA in parallel (on top of the quarter car setup).

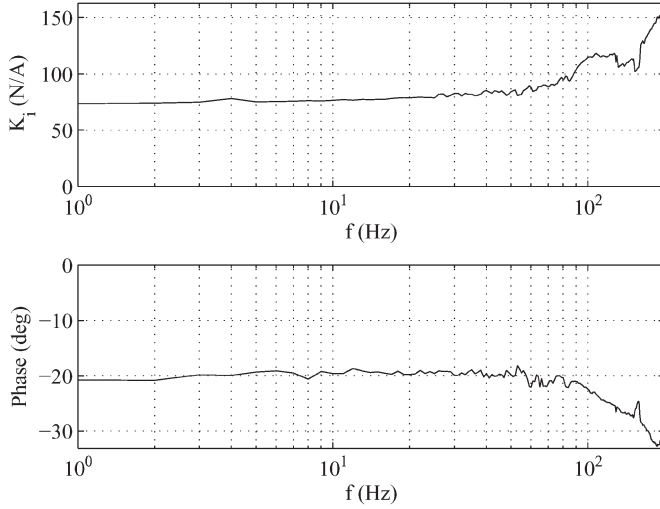


Fig. 13. Bode diagram of the motor constant of the TPMA.

A. TPMA

One of the advantages of an electromagnetic actuator compared with a hydraulic actuator is the increased bandwidth. In Fig. 13, the Bode diagram of the motor constant of the TPMA is shown; this test is performed while applying a sinusoidal nominal current (corresponding to a nominal force of 1 kN and a motor constant of 75 N/A), with increasing frequency to the locked TPMA. It can be observed that the bandwidth of the actuator is higher than 50 Hz, proving the improved dynamic force capability compared with a hydraulic system. The measurement results for higher frequencies (> 100 Hz) are inaccurate since resonance frequencies of the total setup are becoming dominant. The phase shift of -20° at lower frequencies is caused by nonlinearities of the setup and hysteresis of the force sensor.

B. Controller

The position of the body mass of 450 kg, i.e., m_b , is measured with an incremental encoder and controlled with a reference position equal to the on-road measurement shown in Fig. 9. A feedforward controller, which uses the measured spring and damper characteristics of Fig. 10 and 11, respectively, is employed. Furthermore, a feedback controller ensures correct tracking of the reference and compensates the friction and cogging forces of the actuator and the setup, as shown in the block scheme of Fig. 14. The feedback controller, with an open-loop bandwidth of 15 Hz, is designed using a model-based approach [26] on a measured frequency response function of the quarter car setup given in Fig. 15. It consists of a notch filter at 138 Hz, a low-pass filter (50 Hz), and a lead filter (zero at 5 Hz and pole at 45 Hz) given by

$$C_{fb} = K_p C_{notch} C_{LP} C_{lead} \quad (1)$$

$$K_p = 20\,000 \quad (2)$$

$$C_{notch} = \frac{1.33e^{-6}s^2 + 230.7e^{-6}s + 1}{1.33e^{-6}s^2 + 1.153e^{-3}s + 1} \quad (3)$$

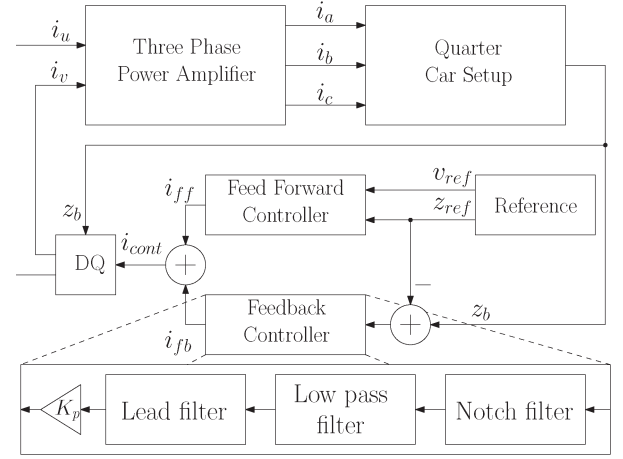


Fig. 14. Block diagram of the measurement setup.

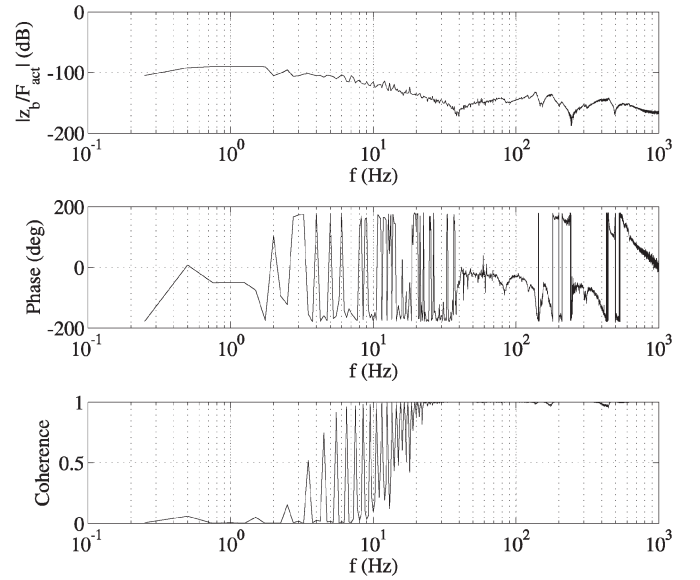


Fig. 15. System identification of the quarter car test setup.

$$C_{LP} = \frac{1}{3.183e^{-3}s + 1} \quad (4)$$

$$C_{lead} = \frac{31.83e^{-3}s + 1}{3.537e^{-3}s + 1} \quad (5)$$

If correct tracking is obtained, the actuator should apply equal force levels, as compared with the passive suspension during the test drive. The tracking of the actuator, as shown in Fig. 16, gives an indication of the dynamic possibilities of electromagnetic actuation. The electromagnetic actuator has the possibility of applying forces equivalent to the passive suspension system within a very small response time.

VII. CONCLUSION

Due to the change in vehicle concepts to the more electric car, the suspension system becomes ever more important due to changes in the sprung and unsprung masses. Active electromagnetic suspension systems can maintain the required stability and comfort due to the ability of adaptation in correspondence with

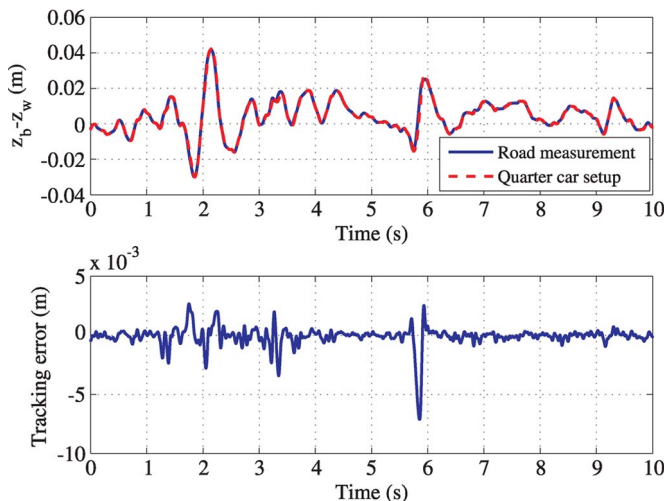


Fig. 16. Time interval of the on-road measurements and off-road electromagnetic actuation on the quarter car test setup.

the state of the vehicle. Specifications are drawn from on- and off-road measurements on a passive suspension system, and it can be concluded that, for ARC, a peak force of 4 kN and an RMS force of 2 kN (duty cycle of 100%) are necessary for the front actuators. Furthermore, the necessary peak damping power is around 2 kW; however, the RMS damping power is only 16 W during normal city driving. The maximum bound and rebound strokes are 80 and 58 mm, respectively. The on-road measurements, which are mimicked on a quarter car setup by means of electromagnetic actuation, a good tracking response, and measurement of the frequency response of the tubular actuator, prove the dynamic performance of the electromagnetic suspension system.

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Bart L. J. Gysen (S'07–M'10) was born in Bilzen, Belgium, in 1984. He received the B.Sc. and M.Sc. degrees in electrical engineering from Eindhoven University of Technology, Eindhoven, The Netherlands, where he is currently working toward the Ph.D. degree with the Electromechanics and Power Electronics Group.

His research interest is electromagnetic active suspension systems for automotive applications.



Johannes J. H. Paulides (M'06) was born in Waalwijk, The Netherlands, in 1976. He received the B.Eng. degree from the Technische Hogeschool 's-Hertogenbosch, 's-Hertogenbosch, The Netherlands, in 1998 and the M.Phil. and Ph.D. degrees in electrical and electronical engineering from the University of Sheffield, Sheffield, U.K., in 2000 and 2005, respectively.

Since 2005, he has been a Research Associate with Eindhoven University of Technology, Eindhoven, The Netherlands, where he is simultaneously a Director of Paulides BV and Advanced Electromagnetics BV, which are small and medium enterprises based in The Netherlands, producing electrical machines and prototype electromagnetic devices. His research activities span all facets of electrical machines, particularly linear and rotating permanent-magnet excited machines for automotive and high-precision applications.



Jeroen L. G. Janssen (S'07–M'10) was born in Boxmeer, The Netherlands, in 1982. He received the B.Sc. and M.Sc. degrees in electrical engineering from Eindhoven University of Technology, Eindhoven, The Netherlands, where he is currently working toward the Ph.D. degree with the Electromechanics and Power Electronics Group.

His research activities are focused on permanent-magnet-based electromagnetic vibration isolation for lithographic applications.



Elena A. Lomonova (M'04–SM'07–F'10) was born in Moscow, Russia. She received the M.Sc. (*cum laude*) and Ph.D. (*cum laude*) degrees in electromechanical engineering from the Moscow State Aviation Institute in 1982 and 1993, respectively.

She is currently a Professor with Eindhoven University of Technology, Eindhoven, The Netherlands. She has worked on the electromechanical actuator design, as well as optimization and development of advanced mechatronics systems.