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Bose Automotive Suspension

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Abstract : This paper offers motivations for an electromagnetic active suspension system that provides both additional stability 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.
***Keywords*: - electromagnetic, Irregularities, eliminating.**

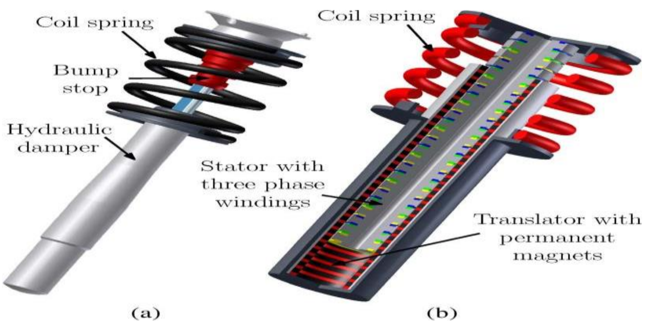
I. INTRODUCTION

Suspension system is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels. It is basically cushion for passengers protects the luggage or any cargo and also itself from damage and wear.

Sir **WILLIAM BRUSH** is the father of suspension in automobiles.

Role of Suspension system

1) It supports the weight of an automobile
2) Provides smoother ride for the driver and passenger
3) Protects your vehicle from damage and wear
4) It also place a critical role in maintaining self driving conditions
5) It also keeps the wheels pressed firmly to the ground for traction
6) It isolates the body from road shocks and vibrations which would otherwise transfer to the passengers and load



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

The efficiency of the electromagnetic suspension system is only reached when completely incorporated in the wheel. 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. A. This paper discusses an active electromagnetic suspension system incorporating a brushless tubular permanent-magnet actuator (TPMA).

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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 com-fort (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
2) head toss: ~2–8 Hz

1. Motion Sickness

Motion sickness, particularly when reading, is a common by-product of exposure to optical depictions of inertial mo-tion. This phenomenon, which is called visually induced motion sickness, has been reported in a variety of virtual envi-ronments, such as fixed-base flight and automobile simulators. Furthermore, Gahlinger 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.

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. 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 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.

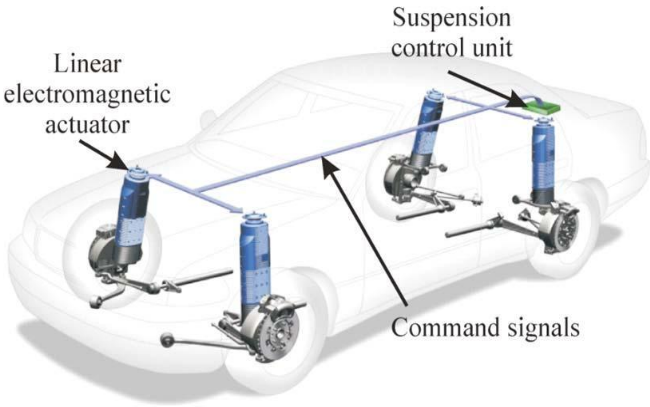
To quantify the acceptable acceleration levels, the ISO 2631 standard is often used as a reference set of root-mean-square (RMS) accelerations, which produce equal fatigue-decreased proficiency. Hence, the ideal suspension system should minimize the frequency response of the sprung mass accelerations 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

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, as shown in Fig Another example is given by the active body control system of Mercedes, which uses high-pressure hydraulics to pre stress 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.



Bose suspension System

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).

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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, as shown in Fig, 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 gen-erator 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 den-sity 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 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; however, a higher force density is obtained than in.

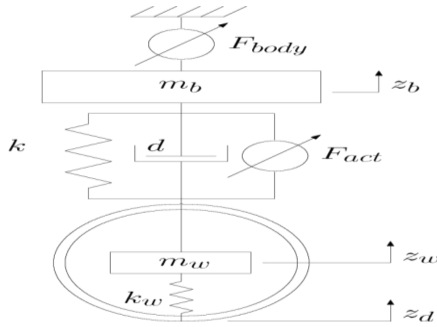


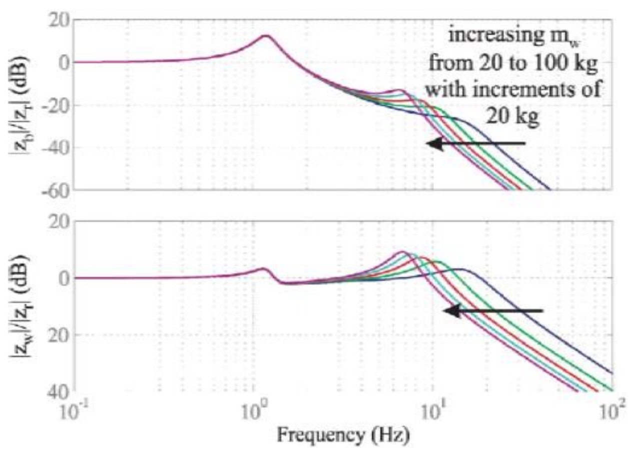
Fig. Quarter car model, including body disturbances and active suspension

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

IV. SYSTEM MODELING



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In general, a full car model 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., 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 (for a wheel motor design, and decreasing the sprung mass, as shown in Fig. (the more electrical car), or increasing the unsprung-to-sprung mass ratio, leads to an increase in response of the frequency range 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.The mass, m_b, represents the car chassis (body) and the mass, m_w, represents the wheel assembly. The spring,k_s, and damper, b_s, represent the passive spring and shock absorber placed between the car body and the wheel assembly. The spring, k_t, models the compressibility of the pneumatic tire. The variables x_b, x_w, and r are the car body travel, the wheel travel, and the road disturbance, respectively. The force, f_s, which is applied between the body and wheel assembly, is controlled by feedback. This force represents the active component of the suspension system.

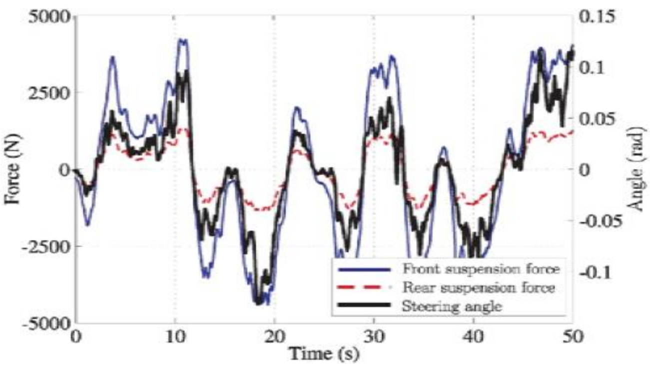

Fig. Time interval of the derived suspension forces from acceleration measurements on the Nürburgring, together with the steering angle

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.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 cal-culated; an extensive analysis is given in. A time interval of the calculated forces deducted from the measurements, together with the steering angle, is shown in Fig.. 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, 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.

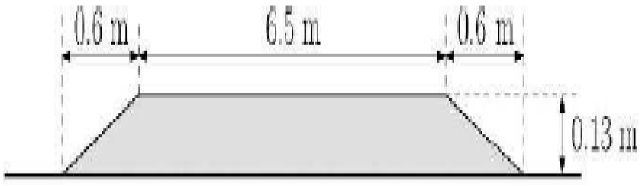

Fig. Sizes of the bump on the test track

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

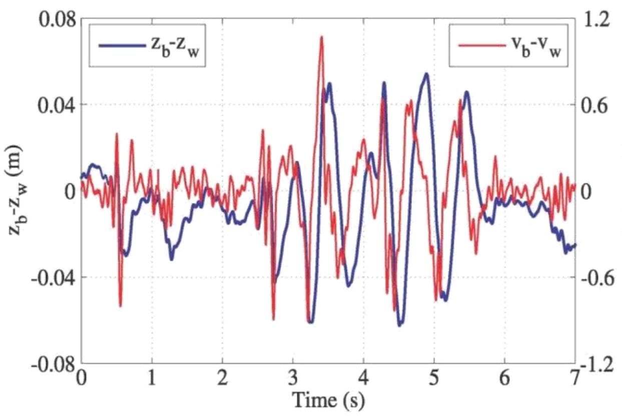

Fig. Position (zb − zw) and speed (vb − vw) measurements while driving on the bump of Fig

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 mea-sured. The speed is then derived, where a small time interval is shown in Fig, where, at that moment (2.5 s), the bump, as shown in Fig, is hit at 35 km/h. From these measurements, the stroke, speed, and force requirements of the suspension sys-tem 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 measure-ment 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, the velocity increases beyond this point.

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The measured spring and damper characteristics are shown in Figs, 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; how-ever, when taking the average value of the total driving cycle, only a power level of 16 W per damper is necessary, which is comparable with the results obtained in for normal city driving.

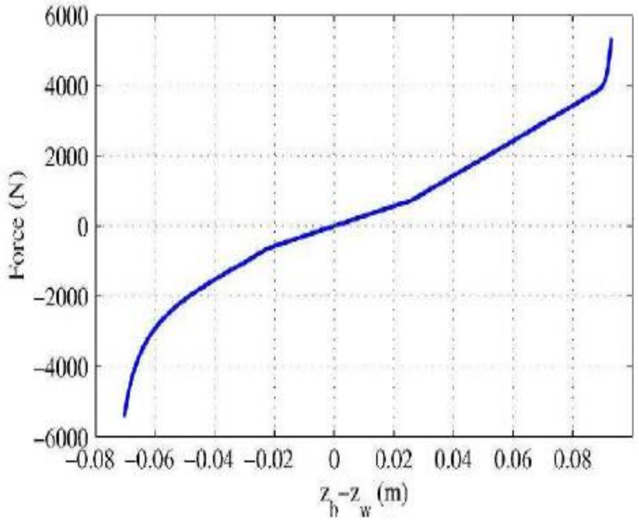
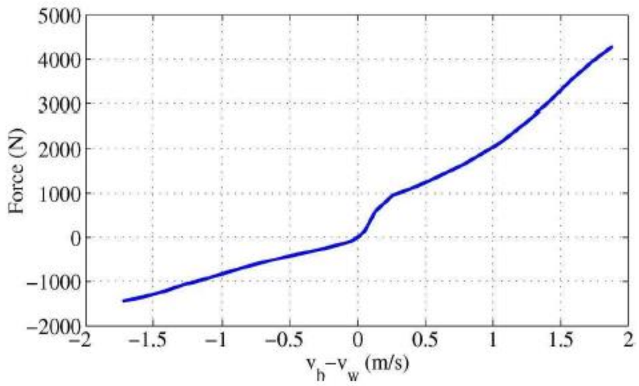


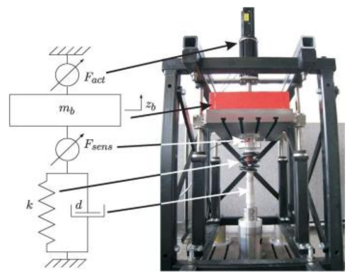
Fig. Measured spring characteristic of the passive suspension



Measured damper characteristic of the passive suspension

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. 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).



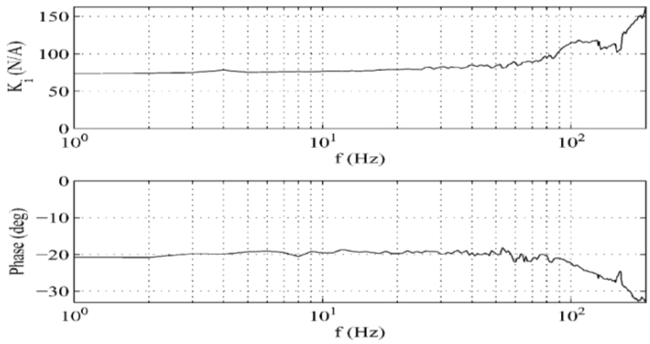


Fig. Bode diagram of the motor constant of the TPMA.

A. TPMA

One of the advantages of an electromagnetic actuator com-pared with a hydraulic actuator is the increased bandwidth. In Fig, 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 be-coming 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. A feedforward controller, which uses the measured spring and damper characteristics of Figs 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 The feedback controller, with an open-loop bandwidth of 15 Hz, is designed using a model-based approach on a measured frequency response function of the quarter car setup given in Fig. 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).

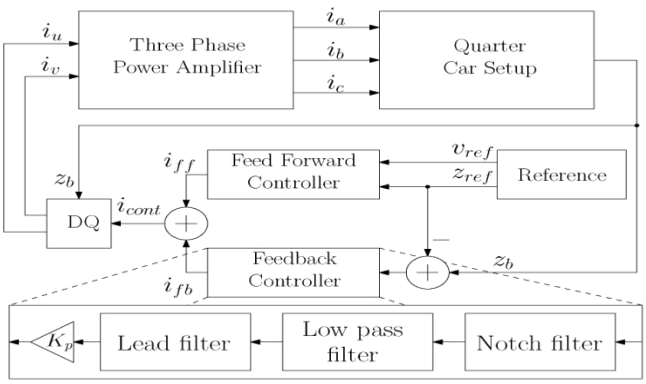


Fig. Block diagram of the measurement setup

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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, 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

Active suspension offers many benefits over conventional and semi-active suspension systems. Electromagnetic suspension is a high bandwidth and efficient solution for improving handling and comfort. Direct drive tubular permanent magnet actuator technology with integrated damping offers a high force density and fail-safe solution. In-lab and on-road experimental verification proved the performance and efficiency of the proposed solution.

Guidance:

Mr. Naga Prasad, Professor at KSIT, Bangalore, Mechanical Department.
Mr. Yogendra Mahajan, Lecturer at KSIT, Bangalore, Mechanical Department.

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