Design and Control of a Linear Electromagnetic Actuation System for Active Vehicle Suspensions

Simon Tuplin, Matthew C Best (Loughborough University, UK)
Jiabin Wang (Sheffield University, UK)

Department of Aeronautical and Automotive Engineering, Stewart Miller Building
West Park, Loughborough University, Loughborough, Leics, LE11 3TU, UK
Phone: +44 (0)1509 227263
Fax: +44 (0)1509 227275
E-mail S.Tuplin@lboro.ac.uk

Traditionally, automotive suspension designs have been a compromise between the three conflicting criteria of road holding, load carrying and passenger comfort. The Linear Electromagnetic Actuation System (LEA) design presented here offers an active solution with the potential to meet the requirements of all three conditions. Using a tubular permanent magnet brushless AC machine with rare earth magnets, thrust densities of over 6 x 10^5 N/m^3 can be achieved with a power requirement of around 50W RMS, much less than equivalent hydraulic systems. The paper examines the performance of the system for both the quarter car and full vehicle simulation, considering high level control of vehicle ride and chassis roll, with the vehicle model being parameterized for a target Jaguar XJ test vehicle. Results demonstrate the ability for 100% roll cancellation with significant improvements in ride quality over the passive Jaguar system.

Topics/Suspension System & Steering System, Suspension Control, Sensors and Actuators

1. INTRODUCTION

Traditionally, automotive suspension designs have been a compromise between the three conflicting criteria of road holding, load carrying and passenger comfort. The suspension system must support the vehicle, provide directional control during handling manoeuvres and provide effective isolation of passengers/payload from road disturbances. Good ride comfort requires a soft suspension, whereas insensitivity to applied loads requires a stiff suspension. Good handling, on the other hand, requires a suspension setting somewhere between the two. Active suspension systems allow this compromise to be overcome by not being constrained by fixed spring and damper rates, employing actuators that are capable of supplying as well as dissipating energy. Active system demonstrators, most notably those using hydraulic actuators, either alone or in tandem with passive components, were developed in the late 1980s by Lotus and Jaguar. These showed significant ride and roll control benefits, but were costly to manufacture and demanding in terms of power consumption; consumers would not pay significantly more at the petrol pump for approximately a 15% improvement in comfort.

More recently the trend has been toward electro-rheological dampers offering a semi-active solution. These modify the damping rate by controlling the alignment of metallic particles within the damping fluid. They offer improvements in ride and handling, but are limited by thermal and peak force capabilities, and are unable to supply energy to the system.

In the light of the very significant advances which have been made in Linear Electromagnetic Actuators (LEAs), achieving thrust force densities greater than 6 x 10^5 N/m^3 [1], an active system based on LEAs has the potential to fulfil the specification for this highly challenging application. LEA based systems have superior controllability and bandwidth, providing isolation between vehicle chassis and wheel due to the absence of any mechanical transmission, and therefore offer potentially outstanding performance [2]. They also have the ability to recover energy that is otherwise dissipated through the shock absorber in passive systems [3], which results in a much more energy efficient system.

The most appropriate LEA technology for meeting the targets is a tubular permanent magnet brushless AC machine equipped with rare-earth magnets providing a high force density, a high force/inertia and a high overload capability (typically 2.5 x rated force). This is particularly relevant since the required active suspension power consumption has to be low, yet the system needs to respond appropriately to impulsive disturbances.

This paper discusses the design and simulation of an LEA strut, focusing on design for implementation. The design incorporates an integral coil spring to support the vehicle weight offering a package of similar dimensions to that of air spring / damper units currently used on high-end production vehicles. The solution has the obvious advantage of no redesign of the chassis for the vehicle manufacturer as the load paths will be identical. The simulation serves dual purpose; it is used to define the necessary performance and characteristics of the LEA and to assess the likely benefits of the drive experience.

The Bose ® Suspension System is the only direct competitor utilising LEA technology. It exhibits all of the usual characteristics associated with active suspension systems; zero body roll, zero body movement over speed humps, zero squat and dive during acceleration and deceleration manoeuvres respectively, demonstrated by video footage on their website [4]. There is no published data to verify the footage, and to the authors knowledge no production vehicle has accepted the technology; this may be due to the use of a longitudinal torsion bar to support the vehicle weight redefining the suspension load paths.

NOMENCLATURE

Table 1 Vehicle Parameters

Description	Notation	Value	Units
Vehicle Mass	M	1838	Kg
Mass Distribution	-	52:48	-
Mass Body	M_b	1665	Kg
Wheel Mass	M_w	43.13	Kg
Roll Inertia	I_{xx}	734	Kgm^2
Pitch Inertia	I_{yy}	3983	Kgm^2
Yaw Inertia	I_{zz}	4240	Kgm^2
Product of Inertia	I_{xz}	-15.6	Kgm^2
Wheel Base	l	3.03	m
C.G. Height	h_q	0.54	m
F/R Track	t_f, t_r	1.56, 1.56	m
F/R Roll Centre	h_f, h_r	0.08, 0.12	m
F/R Damping	b_{sf}, b_{sr}	1.00, 1.13	$Nmsmm^{-1}$
F/R Sus Stiffness	k_{sf}, k_{sr}	20.1, 22.7	Nmm^{-1}
Tyre Stiffness / Damping	k_t b_t	$1.8x10^5$, 0	Nmm^{-1}, $Nmsmm^{-1}$
F/R Anti-Roll Bar Stiffness	k_{rf}, k_{rr}	14.4, 3.11	Nmm^{-1}
Gravitational Constant	g	9.81	ms^{-2}

Table 2 Notation

Description	Notation	Units
Displacements	x,y,z	m
Forward/Lateral/Vertical Velocity	u,v,w	ms^{-1}
Forward/Lateral/Vertical Acceleration	a_x,a_y,a_z	ms^{-2}
Roll/Pitch/Yaw Angle	ϕ,θ,ψ	rad
Roll/Pitch/Yaw Rate	p,q,r	$rad.sec^{-1}$
Roll Sensitivity	R_ϕ	$rad.s^2m^{-1}$
F/R Roll Stiffness	k_ϕ	Nmm^{-1}
Tyre Deflection	x_1	m
Suspension Deflection	x_2	m
Wheel Velocity	x_3	ms^{-1}
Body Velocity	x_4	ms^{-1}
Control Force (Ride/Roll)	$F_{C,rd/rl}$	N
Lateral/Sus/Tyre Force	F_y,F_s,F_t	N
Road Vertical Velocity	V_{ROAD}	ms^{-1}
LQR Gain Matrix	k	-
LQR Cost Coefficients	α,β	-

2. LEA AND SUSPENSION STRUT DESIGN

2.1 Overview of the design of the Linear Electric Motor

There are various linear machine technologies and numerous topologies which might be employed. The main technologies are induction machines, permanent magnet machines and switched reluctance machines [5]. Of the possible topologies, tubular configurations are compatible with the packaging/integration requirements since they have zero net radial force between the armature and stator, no end-windings, and they are volumetrically efficient, Figure 1. However, due to their significantly higher efficiency, only permanent magnet excited machines are deemed to be appropriate for the proposed active suspension applications. These may be classified as moving-coil, moving magnet or moving-iron.

Moving-magnet motor topologies have been shown to have the required high force capability. These may be radially, axially, or Halbach magnetized, and either slotted or slot-less. Although a Halbach magnetization distribution yields desirable features, it remains relatively difficult to manufacture magnets with an ideal Halbach magnetization. A simpler form, referred to as a quasi-Halbach magnetization [6], is, therefore, often employed. Further, a three-phase tubular PM machine can be wound to facilitate either brushless DC or brushless AC operation. To overcome known AC and DC winding limitations, a modular stator winding configuration is employed [7] in which each phase is comprised of a number of concentric coils which are adjacent to each other. This results in a high fundamental winding factor and a small number of stator slots for a given number of poles. This is not only conducive to a lower manufacturing cost, but also results in a fractional number of slots per pole. Thus, the cogging force due to stator slotting can be very small without employing skew. This machine topology is, therefore, considered as the most suitable candidate for this application.

Figure 1 shows a 9-slot, 10-pole quasi Halbach magnetised tubular machine with modular winding. Each pole of the magnetic armature comprises of one radially and one axially magnetised ring magnet. The salient feature of the quasi-Halbach magnetisation is that the axially magnetized magnets essentially provide a return path for the radial air-gap flux, and hence, the flux in the inner bore is relatively small. As a result, the use of a very thin ferromagnetic tube or even a non-magnetic tube on which to mount the magnets will not significantly compromise the thrust force capability. This is conducive to reducing un-sprung mass and hence enhancing the dynamic capability of the active suspension.

Figure 2b, shows the variation of thrust force with the peak current. As can be seen, there is essentially a linear relationship between thrust force and motor current, and the motor has a peak force capability up to approximately 5000N, and a target RMS force capability of 1500 - 2000N.

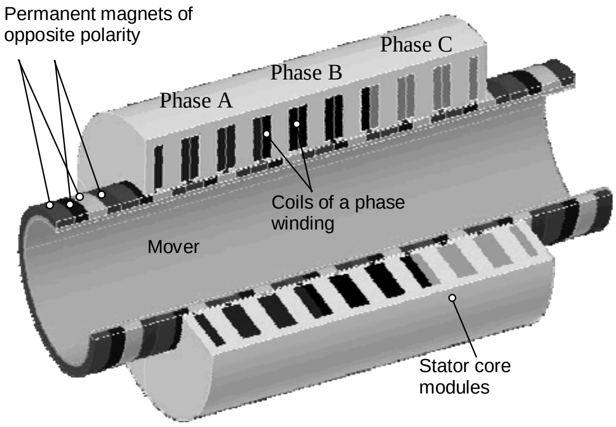

Figure 1 Schematic of 9-slot, 10-active pole tubular PM machine with quasi-Halbach magnetization

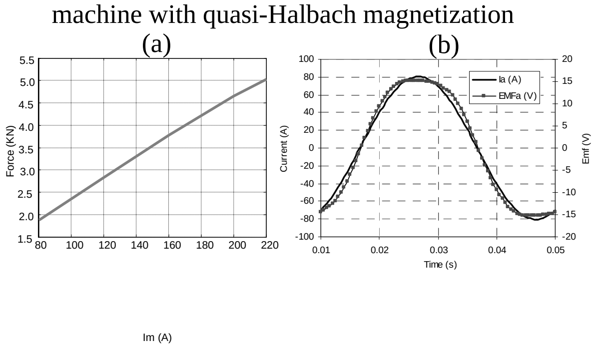

Figure 2 (a): Variation of thrust force with peak phase current (b): Phase A emf and current waveforms under rated motoring operation,

2.2 Integration of LEA into feasible strut design

A compact strut design has been finalised, that incorporates the coil spring, offering an LEA package of similar dimensions to that of air spring / damper units currently used on production vehicles. The solution has the obvious advantage of no redesign of the chassis for the vehicle manufacturer as the load paths will be identical, reduced load through the LEA as the coil spring will take the static load and a self contained unit with no dependency on other vehicle systems (e.g. cooling). The air-spring strut from the Jaguar XJ was used as a design guide line for the principle dimensions.

Figure 3 shows the final assembly drawing, with the lower clevis and vehicle chassis top mount omitted. Thick wall plastic bush bearings {1} have been utilised to minimise eddy current losses from the mover. Longitudinal slots are cut through the bearing housing to capture any eddy current circulation. The plastic of choice is PTFE with 25% 'glass fibre' infill. This has a very low dynamic coefficient of friction (~0.07) and several orders of magnitude increase in wear life over virgin PTFE as well as improved dimensional stability. The apparent over-sizing of the bush bearings is a conservative design to prevent locking.

The coil spring {2} is incorporated inside the mover. The one disadvantage of this choice of spring is that self levelling of the vehicle is not possible. The original intention was to use a rolling lobe (air-spring), but expected peak temperatures have prohibited this. The spring top mount is connected to the vehicle chassis through a 14mm diameter stainless steel 'damper tube' {3} which is rated to 20KN in bump.

A guide bush {4} is mounted in the top-cap of the mover to support the tube, but minimal bending force is anticipated. The mover top-cap also has four holes to prevent air damping as the system moves. With edge clearance this provides 710mm^2 of through flow, ample at ambient pressures.

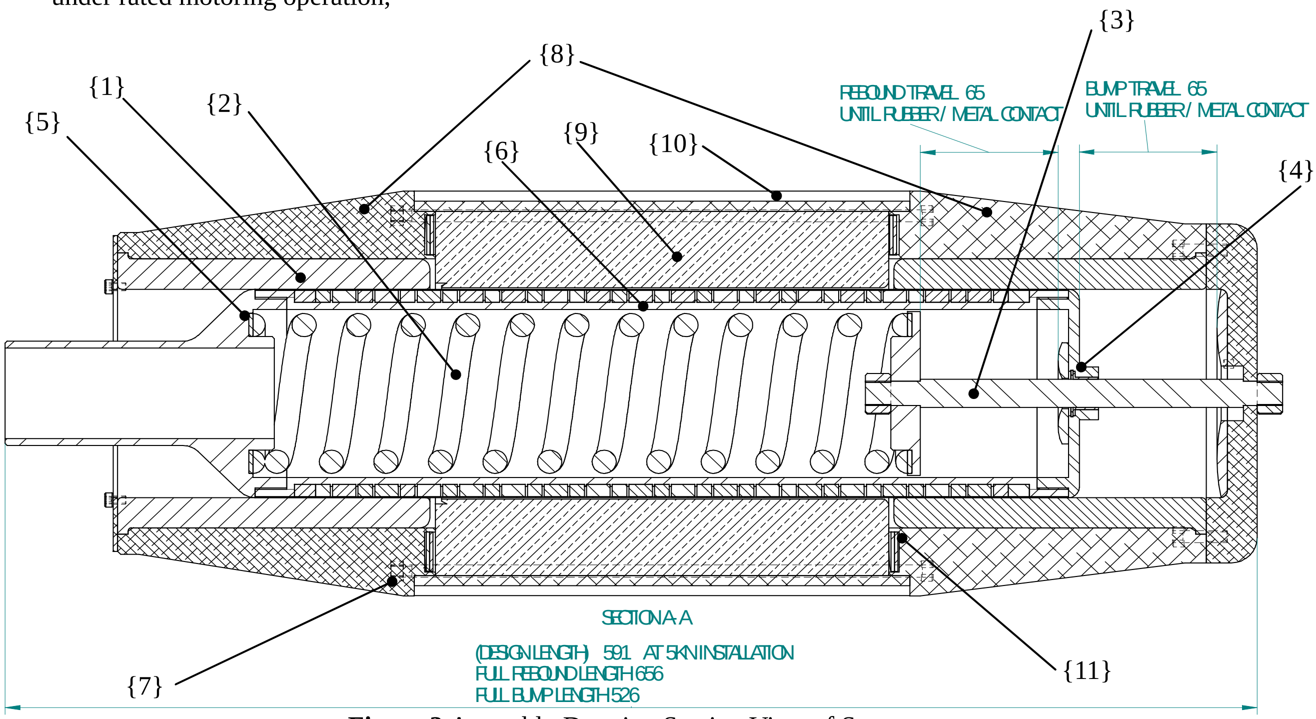

Figure 3 Assembly Drawing Section View of Strut

The coil spring is mounted either end on PTFE coated thrust bearings {5} to allow rotation during actuation. This is to minimise bowing of the spring during compression, preventing contact between the spring and the mover inner wall {6}. Eight stress bolts {7} are used to clamp the bearing housings {8} to the motor, providing pre-loading to the motor {9} and shell casing {10} through paired wave spring washers {11}. The wave spring washers have an approximate rate of 200 Nmm^{-1}, thus with a compression of 10mm will yield 8000N of preload when used in two pair, twice that of the peak force.

3. QUARTER AND FULL VEHICLE MODELS

The paper examines the performance of an LEA active suspension system in both quarter vehicle and full vehicle simulations, using real world road and vehicle information. The vehicle used to populate the models and for the data acquisition is a Jaguar XJ 3.5 (MY2002) with an Oxford Technical Solutions 3200 inertial GPS system installed. Additionally wheel speed and hand wheel angle information was acquired. The vertical road input used is a local, rough ‘B’ class road, measured at 0.2m intervals. The profile is filtered for the simulation such that it has zero mean and low frequencies are removed, the high pass being set sufficiently low that body bounce response (~1Hz) is not improved. This is necessary because the model uses a global wheel and body velocity giving rise to low frequency drift in ride controllers.

3.1 Quarter Vehicle Suspension Model

The standard quarter vehicle (QV) model is utilised with parameters as given in table 1. The primary purposes of the QV simulation is to optimise / asses the cost of the coil spring rate. This aim is to meet two criteria; to support the vehicle load and hence reduce power requirement, and to allow the LEA to run in the optimum configuration for power regeneration. The motor is incorporated into the QV model using a 10ms delay with a 100Hz bandwidth. A secondary use is to provide initial control parameters for use in the full vehicle model.

3.2 Full Vehicle Model

The full vehicle model has 4 degrees of freedom, with an inclined roll axis. Anti roll/dive/squat suspension geometry is used and a combined slip Pacejka ‘magic formula’ lifting tyre model [8] provides vertical load dependency.

Equations (1) – (4) are the dynamic equations of motion for yaw, lateral, longitudinal and roll degrees of freedom (body), and the system states in equation (5):

$$I_{zz}\dot{r} + (\varepsilon I_{zz} - I_{xz})\dot{p} = a\sum_{i=1,2} F_{yi} - b\sum_{i=3,4} F_{yi} + t_f(F_{x1} - F_{x2}) + t_r(F_{x3} - F_{x4}) \tag{1}$$

$$M\dot{v} + Mh\dot{p} = \sum_{i=1,4} F_{yi} - Mur \tag{2}$$

$$M\dot{u} = \sum_{i=1,4} F_{xi} + Mrv + Mhrp \tag{3}$$

$$-I_{xz}\dot{r} + Mh\dot{v} + (I_{xx} - \varepsilon I_{xz})\dot{p} = -Mhur - (B_f + B_r)p + (Mgh - K_f - K_r)\phi + (h_{frc} - h_0)\sum_{i=1,2} F_{yi} + (h_{rrc} - h_0)\sum_{i=3,4} F_{yi} \tag{4}$$

$$x = \begin{bmatrix} u & v & w & p & q & r & \phi & \theta & \psi & z \end{bmatrix}^T \tag{5}$$

The suspension is based on having a quarter car model at each wheel station and the following load modifiers used for anti dive(+)/squat(-) geometry (6) and lateral load transfer and jacking (7).

$$F = \frac{\sum F_x h_g}{2l} \tag{6}$$

$$F = \frac{F_y h_{f/r}}{t_{f/r}/2} \tag{7}$$

The model is populated from table 1, with tyre and engine parameters tuned such that the longitudinal and lateral acceleration traces correlate with those recorded on the test vehicle. Test vehicle data inputs to the model are hand wheel angle and the four wheel speeds. The data was logged from a drive on a dry, local ‘B’ class road, with a mean speed of 22ms^{-1}. The road represented a good mix tight and open corners.

4. CONTROL STRATEGY

The control of the suspension system can be split into three distinct areas; Ride Control, Roll Control and Handling Control. Ride and roll control are considered here, with combined handling control to be dealt with in later work.

4.1 Ride Control

The premise for the ride control is to minimise $\dot{x}_4$ whilst maintaining x_1 & x_2 at passive levels. The control implemented in simulation is LQR four state feedback system:

$$F_{C/rd} = -k\underline{x} \tag{8}$$

the solution to k being such that the control force minimises the infinite time integral (9); thus the costing of body acceleration is conducted, with α & β tuned to maintain the passive system RMS response of the tyre deflection and suspension travel.

$$J = \int_0^\infty \alpha x_1^2 + \beta x_2^2 + \dot{x}_4^2 \partial t \tag{9}$$

The passive spring is included and 100% of the system damping ($b_{sf}, b_{sr} = 0$) is supplied by the LEA maximising the regenerative properties. The control is then applied in conjunction with the passive support components.

4.2 Roll Control

Roll control is applied using an open loop controller, allowing roll sensitivity to lateral acceleration to be defined. The target design for the system is to achieve 100% roll cancellation. However, research programs by Jaguar Cars have found that drivers prefer to experience some roll during cornering, therefore the implemented roll correction will be customer driven rather than being limited by actuator performance. A consequence of this is the possibility to use a smaller actuator, reducing cost, weight, power requirements and heat generation; all desirable.

The required roll control force is derived directly from the roll sensitivity of the vehicle, given by(10), with lateral acceleration being estimated from tyre force information. The roll control force (11) is calculated proportionally and separately front to rear and applied equally and oppositely left to right.

$$\phi = R_\phi a_y$$
$$R_\phi = \frac{M_b g h_{f/r}}{k_\phi - M_b g h_{f/r}}, a_y = \frac{\sum F_y}{M}$$
$$k_\phi = 2k_{sf/r}\left(\frac{t_{f/r}}{2}\right)^2 + k_{rf/r}$$
$$K_s = K_{passive} + k_{2:LQR} \tag{10}$$

$$F_{C\phi f} = \frac{R_{\phi f}}{R_{\phi f} + R_{\phi r}} K_{rc}\phi_f, F_{C\phi r} = \frac{R_{\phi r}}{R_{\phi f} + R_{\phi r}} K_{rc}\phi_r \tag{11}$$

5. RESULTS

5.1 Coil Spring Rate Selection

Initially, the spring damper unit in the passive quarter car system was replaced by a fully active unit, with unlimited performance and bandwidth. This was controlled using the LQR defined in section 3.1. The optimal spring rate can be found by least squares regression between the active control unit force and the suspension travel. The use of an optimal spring will result in a reduction of the power requirement of the actuator without affecting the performance.

However, the usable passive spring rate is ultimately constrained by physical limits. If for example, the actual optimum spring rate were used (1.8 N/mm) a spring of over 3m in length would be required. (2.77m compression to carry 5KN corner load, plus operating length). This is clearly not possible as the spring would either become coil bound before a useful installation length is achieved, or would have so few coils as to be laterally unstable.

To support the vehicle weight, keeping within the physical limits of the LEA, a compromise spring rate of 30 Nmm^{-1} is to be used. This has a free length of 475mm, , and an equal bump and rebound travel of 65mm. Table 3 highlights the power consumption of the two configurations and, for reference, the passive system.

Table 3 Coil Spring Rate Selection

Measure	Passive	With Optimal Spring	With Selected Spring
RMS x_1 [m]	0.0043	0.0046	0.0046
RMS x_2 [m]	0.0165	0.0157	0.0157
RMS Power[W]	339	365	410
Power Flow [W]	-339	-237	-236

Use of the compromise spring rate will have no effect on the primary performance measure, body acceleration, and the RMS power cost of increasing the spring rate away from the optimal is approximately 45W. The result is a negative term in $k_{2:LQR}$, working to compress the passive spring to maintain ride performance. Considering the regenerative properties of the LEA, results indicate that the spring rate has little influence on the overall power flow, mainly because the majority of the ride improvement is gained through the active damping portion of the control.

5.2 Quarter Car LQR Controller Performance

Using the selected spring rate, the feedback gain is; k = [-10700 -29700 925 -530]. The second term multiplying suspension deflection acts against the spring, indicating a lower rate would be desirable, but this is only the case for ride control. For roll control the higher rate passive spring will reduce the power consumption.

Simulation using real world road information produces a significant improvement of body acceleration for the active system, with RMS x_2, Figure 4b, and RMS x_1, (not shown) being maintained at passive levels. The RMS body acceleration is reduced from 1.00 ms^{-2} (passive) to 0.62 ms^{-2} (active), see Figure 4c. The peak force observed was 1354N, with the RMS force being 518N. Inspection of the body acceleration suggests that body bounce is well controlled and the wheel hop less so. The results are considered satisfactory to continue with (more relevant) full vehicle modelling.

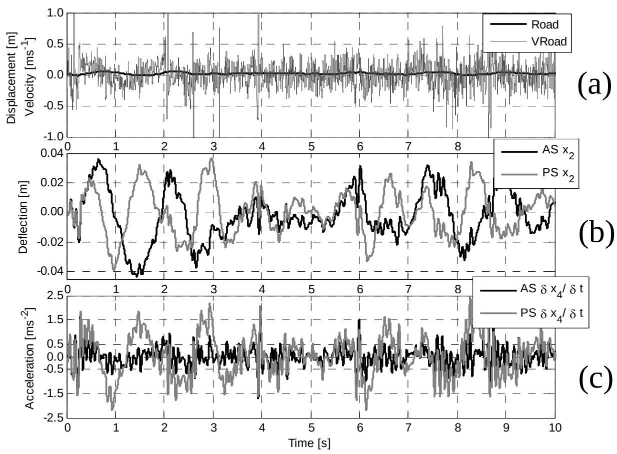

Figure 4 LQR Control on Real Road, Quarter Vehicle

5.3 Full Vehicle Model Ride Performance

For the full vehicle model with the same vertical road information, and actual hand-wheel angle and wheel speed information, the RMS force required for the hardest worked actuator increased to 658N with a peak force of 2745N observed. The increases can be accounted for by lateral load transfer and body roll. This facilitated a 32% reduction of body acceleration compared to the passive system. Figure 5 shows the proportion of time spent at specific control forces. What this clearly demonstrates is that, despite a relatively high peak force requirement, only 0.3% of time is spent above 2KN and 3% above the RMS capability of the actuator.

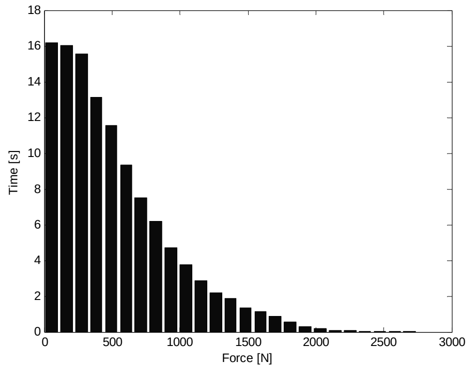

Figure 5 Time Spent at LEA Control Force

5.4 Full Vehicle Model Roll Performance

For a simulated step steer input of 2^o steer angle (at the front wheel) at 40 ms^{-1} the peak actuator force is 3080N for 0° steady state roll angle, reducing to 1688N at $20ms^{-1}$. The passive system roll angles are 4.58° and 3.03° respectively. If the roll cancellation requirement is reduced, (see Figure 6) the force requirement from the actuator significantly reduces. For example, a reduction from 100% to 50% correction reduces the maximum actuator force from 1688N to 924N.

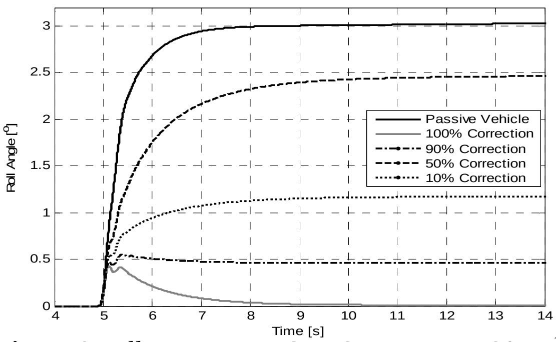

Figure 6 Roll Response to Step Steer Input at $20ms^{-1}$

When the 100% target cancellation strategy is applied to the model with test vehicle inputs the RMS force utilised is 1259N, with a peak force of 2576N. The actual reduction of body roll is 82.5%, from 1.43^o RMS to 0.25^o RMS, Figure 7.

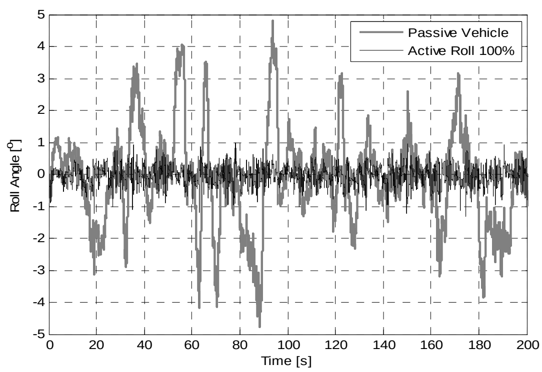

Figure 7 Full Vehicle Roll Response: Active vs Passive

6. CONCLUSION

It is clear from the simulation results that the potential peak required force from the actuator is 5825N and the RMS force is 1917N, assuming that the roughest road condition is experienced in conjunction with an extreme cornering manoeuvre. This correlates with data for a Jaguar X-type being driven on the limit at the Nürburgring track, where the peak force required would be 5KN, and the force distribution is in agreement with Figure 5. This is achievable with the actuator design, and the requirement will be reduced for a lower roll cancellation target. Consideration also needs to be given to the viability of the machine in a production situation; reducing the peak and RMS requirements will reduce the size, weight and more importantly the cost, producing a more commercially viable solution. Given the fundamental performance limitations of semi-active systems, there still remains a demand for a fully active, high bandwidth actuation solution, with high efficiency and low power consumption, as well as the potential for low cost manufacture in automotive market volumes.

REFERENCES

[1]. J. Wang and D. Howe, "A linear permanent magnet generator for a free-piston energy converter", *Proceedings of Int Electrical Machines and Drives Conference (IEMDC2005)*, San Antonio, TX, USA, pp.1521-1528.

[2]. Martins, J. Esteves, G. D. Marques, and F. P. da Silva, "Permanent magnet linear actuators applicability in automobile active suspensions", *IEEE Trans. on Vehicular Technology*, vol. 55, no. 1, pp. 86-94, 2006.

[3]. Martins, J. Esteves, and F. P. da Silva, "Energy recovery from an electrical suspension for EV", *Proc. Electric Vehicle Symposium* (EVS-15), Brussels, Belgium, 1998.

[4]. www.bose.co.uk Learning Centre; Bose ® Suspension System.

[5]. D. Howe, Z.Q. Zhu and R.E. Clark, "Status of linear drives in Europe", Invited Paper at *Linear Drives for Industry Applications* (LDIA 2001), Nagano, 2001, pp.468-473.

[6]. J. Wang and D. Howe, "Tubular modular permanent magnet machine equipped with quasi-Halbach magnetized magnets ---- Part I: Magnetic field distribution, emf and thrust force", *IEEE Trans. on Magnetics*, vol. 41, no. 9, pp. 2470-2478, 2005.

[7]. J. Wang and D. Howe, "Tubular modular permanent magnet machine equipped with quasi-Halbach magnetized magnets ---- Part II: Armature reaction and design optimisation", *IEEE Trans. on Magnetics*, vol. 41, no. 9, pp. 2479-2489,

2005.
[8]. Pacejka, H.J. and Besselink, I.J.M., "Magic Formula Tyre Model with Transient Properties", Vehicle Systems Dynamics Supplement, 1997, vol.27,
pp. 234 - 249