Electromagnetic damper control strategies for light weight electric vehicles

Alista Fow
Department of Engineering,
University of Waikato,
Hamilton, New Zealand,
ajf8@students,waikato.ac.nz.

Mike Duke
Department of Engineering,
University of Waikato,
Hamilton, New Zealand
Dukemd@waikato.ac.nz

***Abstract*—An investigation is conducted into the performance of passive, semi-active and active electromagnetic dampers. Theoretical models are constructed of the dampers and these are included in two degree of freedom models of the suspension. The passive and semi-active electromagnetic dampers are significantly heavier than commercial hydraulic dampers. In the case of active electromagnetic damper, the reduction in passenger acceleration is 88 percent when compared to passive damper and 61 percent when compared to a semi-active damper. The power consumption is similar to a magnetorheological semi-active damper.**

Keywords—electromagnetic, active, passive, damping.

I. INTRODUCTION

Lightweight vehicles have always had a disadvantage in damping their motions and there has been on going research to improve the damper performance in lighter vehicles. In recent years there has been an increase in the use of active and semi-active suspension systems in commercial automobiles. However, the systems currently available are for heavier combustion engine vehicles. In the future, there might be a need for effective suspension systems for lightweight urban electric vehicles (EVs). A typical vehicle suspension system comprises a mechanical spring and hydraulic damper. These have been developed to a high degree of performance. As the spring has been proven to be an effective and economic element of suspension systems, it is the damper that has been the focus of much research into how to improve vibration performance. Damping for ultra-light weight electric vehicles presents several major problems that need to be overcome for commercial viability. The two most serious issues are the use of electrical power in active suspension systems and the low un-sprung mass of light weight EVs.

Any suspension system has three main measurable requirements: passenger comfort, road handling and suspension travel [1]. In the design of a suspension system these three factors are often contradictory. With conventional passive systems, improving one of these factors is usually at the expense of one or both of the remaining factors. In sports cars the road handling and suspension travel are usually optimised and passenger comfort is a secondary factor. In luxury cars the passenger comfort is optimised, usually at the expense of road handling. Even so, the passive hydraulic damper is the predominant method in modern automobiles. It has been suggested by [2], [3] and [4] and others, that a passive electromagnetic damper could be used to regenerate power for an electrical system. However, [5] demonstrated that the power generated by a lightweight vehicle suspension is not sufficient to justify the extra complexity.

In the case of passive damping systems the dynamics of the system are described by [6], [7] and [8]. In a classic single degree of freedom damper as in figure 1, the force equation is given by (1)

$$m\ddot{z} + c\,(\dot{z}_1 - \dot{z}_0) + k\,(z_1 - z_0) = 0 \qquad (1)$$

Where m is the sprung mass (body of vehicle), z'' is the acceleration of the sprung mass, c is the damping coefficient, $\dot{z}_1 - \dot{z}_0$ is the relative velocity of the sprung mass and the road surface, k is the spring constant and (z1 – z0) is the relative displacement of the sprung mass and the road surface. The damping coefficient can be described as in (2),

$$c = 2\,\zeta\,\sqrt{m\,k} \qquad (2)$$

where ζ is the damping ratio. Substituting (2) into (1) and rearranging for acceleration to get (3),

$$\ddot{z} = \frac{2\zeta\sqrt{mk}\left(\dot{z}_1 - \dot{z}_0\right) + k\,(z_1 - z_0)}{m} \qquad (3)$$

Therefore the acceleration of the sprung mass is proportional to the inverse of the sprung mass. For a given system, the lighter the sprung mass of the vehicle, then the greater the accelerations experienced by the sprung mass for a

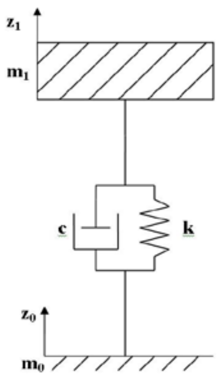


Figure 1: A Single Degree of Freedom model

given input. While reducing the spring rate will reduce the acceleration, this option is usually limited by the suspension travel of the vehicle as it is being loaded. To improve the effectiveness of damping in automobiles, alternative methods of controlling the effectiveness of the damping system have been developed. These include the use of semi-active dampers, usually magnetorheological with a Karnopp Skyhook algorithm. This paper investigates how passive, semi-active and fully active electromagnetic elements could improve the vibration of lightweight EVs.

II. SEMI-ACTIVE DAMPING

For a passive damper in a Single Degree of Freedom system, the force generated is directly proportional to the relative velocities of the unsprung mass and the sprung mass. This can be expressed as (4)

$$F = c\,(\dot{z}_1 - \dot{z}_0) \tag{4}$$

where F is the force in Newtons.

In 1974 Dean Karnopp et al. proposed a 'Skyhook' damper [9]. In this, the position of the body of the car is not measured relative to the level of the ground, rather the position of the sprung mass is fixed to an external inertial reference point.

This is usually considered as an infinitely removed point in the 'sky'. Using this external reference the equation for the force in the system at any given time is given by (5).

$$F_c = b\dot{x} + k\,(x - x_0) \tag{5}$$

where k is the spring constant, b is the damper constant for the Skyhook Damper, x is the displacement and x˙ the velocity of the sprung mass and xo is the displacement of the unsprung mass. In an ideal Skyhook system, the force generated by the damper is given by (6)

$$F_D = b\dot{x} \tag{6}$$

However to achieve this, ideal damping would require an input of energy at certain points in the cycle rather than just dissipating it. When the damper entered a regime where energy had to be supplied, then the damper is switched off so as to provide no force. This is given in (7)

$$c = \begin{cases} \dot{x}(\dot{x} - \dot{x}_o) > 0, & F = b\dot{x} \\ \dot{x}(\dot{x} - \dot{x}_o) < 0, & F = 0 \end{cases} \tag{7}$$

Crosby and Karnopp note that the performance of a semi-active damper while approaching that of an active damper, cannot exceed it. It was shown by [10] that a physical semi-active damper using a Skyhook algorithm did surpass the performance of a passive damper at all frequencies tested,.

A passive system with adjustable parameters or a semi active system in which some damping force components are generated actively or semi actively can perform, in essence, as well in filtering roadway disturbances as any then state of the art active suspensions, [11].

Due to the benefits of the semi-active damper over the traditional passive damper in providing increased ride comfort in private automobiles, the use of semi-active dampers has been investigated for other vehicles, including Sport Utility Vehicles (SUVs) [12] , trains [13] and tractors [14].

Of the three major requirements of an automotive suspension system: Passenger Comfort, Road Contact Force and Suspension Travel, the semi-active damper proposed by Karnopp in 1973 is designed to optimise passenger comfort. To optimise road forces, [15] proposed the Ground Hook Damper. Modern semi-active suspensions use magnetorheological (m.r.) dampers.

Since the 1990s that there has been a revival of interest in m.r. dampers. This includes research into both the damper design [16] and [17]; as well as the control properties and algorithms associated with these dampers, [18] and [19]. These have contributed to the significant body of work that has now been established in this field.

When switched on, the MR fluids in a modern MR damper could develop an apparent yield stress of up to approximately 100 kPa. The response time of the fluid is in the range of 10-20 ms depending upon the device and the magnetic circuit design[18]. For the production Magneride semi-active damper, the peak power is given by [20] as 20 watts per damper.

As m.r. dampers draw power from an electrical vehicles limited power supply, an investigation is conducted into using a passive linear electromagnetic damper in a semi-active mode.

III. MODELLING THE PASSIVE E .M. DAMPER

The magnet is modelled as an air cooled solenoid, the field at any point for an air cored solenoid being derived in [21]. For a single loop in a coil the flux can be approximated as a series of small concentric rings centred on the z axis, of a known thickness, dr , and area, dA = πr 2 dr and by determining the magnetic field at that point in the ring, as illustrated in figure 2. By radial symmetry the magnetic field at any point on that ring will be constant and the total magnetic field at point P is $B_P = B_r + B_z$. An integration of area for the magnetic field is performed using numerical means.

The total flux for a single loop is therefore given by (8)

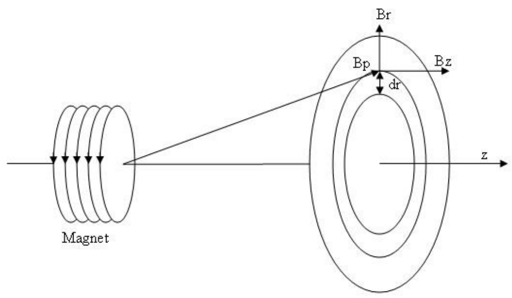


Figure 2: The flux of a given area

$$\Phi_B = \sum B_P \, dA \tag{8}$$

For a solenoid made up of more than one loop, each being a different size and/or being at a different position on the z axis, so for each loop the flux must be determined individually. The total flux of the coil being given by 9

$$\Phi_{Total} = \sum \Phi_B \tag{9}$$

As Faradays Law is dependent upon both time and the flux, movement of the coil and the magnet relative to each other must be taken into account where the E.M.F. is the change in flux caused by the relative motion divided by the time over which the motion occurred as given in 10

$$EMF = \frac{d \sum \Phi_B}{dt} \tag{10}$$

A. The model of a passive damper

The passive e.m. damper suspension model is made in three parts. There is the equations of motion of the damper which are well known used to model the undamped motion of the sprung mass. There is also the natural damping of the system which is caused by friction and hysteresis effects, among other factors. This had components of both viscous and Coulombic damping. The third factor that is added to this model is the damp force produced by the passive e.m. element.

Adding these factors to (1) yields (11)

$$k(z_1 - z_0) + c_{nat}(\dot{z}_1 - \dot{z}_0) + m\ddot{z} + F_C + F_{DAMPER} = 0 \tag{11}$$

To determine the feasibility of the passive electromagnetic damper as a component for full size automobiles a two degree of freedom model of a car suspension system is created. A model of the magnetic fields of the two larger magnets is then constructed using the techniques by [21]. This is then used to model the fluxes in the coils and to make a look up table of the force generated by the magnet and coil at various displacements, as described earlier.

For a quarter car, the masses could be modelled as a two degree of freedom as in figure 3, where m_1 is the unsprung mass/tyre, m_2 is the sprung mass/car body, z_0 is the displacement of the road surface, z_1 is the displacement of the unsprung mass, z_2 is the displacement to the sprung mass, k_1 is the tyre stiffness, k_2 is the shock absorber spring stiffness, c is the damping coefficient of the tyre and F_D is the damper force. For a two degree of freedom system the equations of the displacement of the unsprung and sprung masses were given by 12 and 13.

$$m_1\ddot{z}_1 + k_1(z_1 - z_0) - c(\dot{z}_1 - \dot{z}_0) + k_2(z_2 - z_1) + F_D = 0 \tag{12}$$

$$m_2\ddot{z}_2 + k_2(z_2 - z_1) - F_D = 0 \tag{13}$$

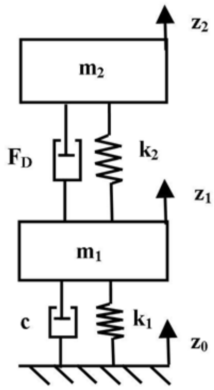


Figure 3: A two degree of freedom suspension system.

As the damping from tyres are usually considered negligible the equations simplify to 14 and 13.

$$m_1\ddot{z}_1 + k_1(z_1 - z_0) + k_2(z_2 - z_1) + F_D = 0 \tag{14}$$

$$m_2\ddot{z}_2 + k_2(z_2 - z_1) - F_D = 0$$

This is modelled in VISSIM (commercial modelling software) for a light weight electric vehicle; typical values for which are given in table 1. The natural frequency of the sprung mass is set at 1.25 Hz.

A pair of magnets were modelled for suitability for use in a passive damper. Both were cylindrical and axially magnetized. In each case the coil length is set as the same as the magnet length and each of the coils consisted of a single layer of wire. The properties of the magnets and their responding coils are given in table 2.

For the damper to be effective, a damping ratio of 0.5 should be the minimum achievable figure. For the lightweight electric vehicle, this gave a damping coefficient of 1,600 Ns/m. For a single coil and magnet the maximum damping coefficient achieved for the ND3522 magnet is 4.60 Ns/m and for the ND5550 the maximum achieved is 20.9 Ns/m. The weight of the magnet-coil system is 0.0934 kg for the ND3522 and 0.5106 kg for the ND5550. With the masses and spring stiffnesses involved a single magnet and coil of the types in table 2 achieved only a small percentage of the required damping coefficient.

The damping can be increased by increasing the number of layers of the windings. By increasing the number of layers to 5, increases the damping effect. The achieved damping for the ND3522 is increased to 23.5 Ns/m and for the larger ND5550, 1017 Ns/m and the masses are 0.171kg and 0.892 kg respectively.

Table 1: Values for a two degree of freedom quarter car system for a lightweight electric vehicle.

Properties	Value
Sprung Mass	200 kg
Unsprung mass	25 kg
Spring Stiffness	12,500 N/m
Tyre Stiffness	100,000 N/m

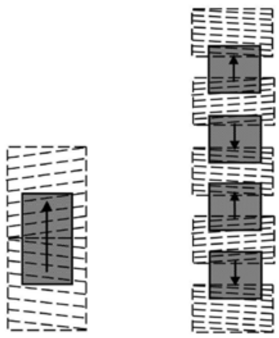
Table 2: Values for two magnets and two coils.

Property - Magnet	ND3522	ND5550
Diameter (mm)	35	55
Length (mm)	22	50
Mass (kg)	0.0740	0.4153
Field Strength at Pole (T)	0.5791	0.5783
Property - Coil		
Length (mm)	22	60
Radius (mm)	20	30
Turns	22	50
Wire diameter (mm)	1.0	1.2
Mass Coil (kg)	0.0194	0.0953
Resistance Coil (Ω)	0.0563	0.14

The damping force is also increased by using a second coil with an opposing direction as illustrated in figure 4a. This second coil is wound in the opposite direction to the first coil and has a reversed polarity. This provides approximately twice the damping of the single magnet coil system, but without the requirement of adding a second magnet. For a single magnet and two matched coils, each of five layers, the damping increased to 47.6 Ns/m and 211.8 Ns/m. Due to the non-linear nature of the magnetic fields, the increase in the damping force and damping constant are also not of a linear nature. The mass of the damping systems are 0.268kg for the ND3522 and 1.368 kg for the ND5550 magnets.

A practical damping coefficient can be obtained through the use of a stack of magnets and coils of opposing magnetic fields and polarities as illustrated in figure 4b. The magnetic/coil stack acted as a two phase linear electromagnetic generator. Modelling is done to determine combinations of magnets and coils that would produce a damping coefficient of 1,600 Ns/m. An arrangement is determined for each magnet and is given in table 3.

The damper mass given in table 3 is the mass of the magnets and the coils only. The mass does not include the connectors between the tyre and the magnet, or the magnet and the car. Nor does it include structural element so that the weight of the vehicle did not crush the coils and damper. A complete commercial hydraulic damper, without a spring, for a typical vehicle with a similar weight to the lightweight electric vehicle was weighed at 3.8kg. Both of the e.m. dampers modelled exceeded this weight before the mass of any connectors or other structure is included.



a b

Figure 4: Two dampers with multiple magnets and coils

Table 3: Passive e.m. damper properties to achieve a damping coefficient of 1,600 Ns/m.

Magnet	ND3522	ND5550
Number of magnets	17	8
Number of coils	18	9
Number of layers	10	5
Damping coefficient (Ns/m)	1,581	1,638
Mass (kg)	4.75	7.61
Length (m)	0.583	0.433

IV. ACTIVE SYSTEMS

Modern active damper systems provide a way to improve the performance of the suspension system in passenger comfort, road handling and suspension travel. By use of hydraulic or electro-magnetic systems it is possible to input energy into the suspension, thus providing force at times that passive and semi-active dampers cannot. This extra power and force of an active system allows the damping to occur over the full cycle of motion and provides more effective damping than a pure passive or a semi-active system. Due to the control algorithms and information gathering employed on a vehicle it is possible to create a coordinated strategy for all four wheels of an automobile. Thus it is possible for the suspension to adjust for dive when under braking, to resist squat when under acceleration and counter roll while the vehicle is cornering.

In practice, most fully active dampers can be divided into two types: those derived from Karnopp's Skyhook damper and those derived from Lotus' Modal Control [22].

The advantages [23] lists for the hydraulic active suspension are the very high force density of a hydraulic system, the ease of control, the ease of design, the commercial availability of parts, the reliability of modern systems and the commercial maturity. However, the disadvantages of such a system were the inefficiency due to the continuously pressurized system, the relatively high time constant caused by pressure loss and flexible hoses and environmental pollution if toxic fluids escape. In the case of electromagnetic active suspension systems there were several advantages and disadvantages noted when compared to hydraulic systems. In [23] advantages noted for an e.m. active damper are the high band width with a frequency of over 10 Hz, there is no need for continuous power to be supplied to the damper, the ease of control of an electronic system, the absence of fluids, there is improved dynamic behaviour, also stability improvement, that there is accurate force control and that the damper can be used in both directions of motion. The disadvantages noted were the increased volume of the suspension due to lower force density, the relatively high current for a 12 V system and that conventional designs need excitation to provide a continuous force. It should be noted that were such a system to be installed in an electric vehicle, then the available voltage is over 100 V and that circuitry already exists that can switch high currents.

A. Active E.M Suspensions

Another consideration is the use of a spring in the suspension system. Systems such as the Lotus Modal System and the BOSE suspension system both use a suspension system without a spring element. While this allows for direct application of the control algorithms to the system, it also requires the active element in the damper to have much higher power requirements than an active system with a spring element. In [24] it is noted that the power consumption for an active hydraulic damper, without a spring element, is 3500W r.m.s. This power consumptions make the use of a damper with a spring element a more attractive proposition than a system without a spring.

B. Basic principles

With a coil and magnet system, if a current is applied to the coil then a magnet field is generated in coil. The magnetic field of the coil then will interact with the magnetic field of the permanent magnet. This interaction produces a Lorentz Force. For a permanent magnet and a single of current carrying wire, as illustrated in figure 5, the force on the wire is given by (15)

$$F = B\,I\,L \sin\theta \tag{15}$$

where F is the force in Newtons, B is the magnetic field in Tesla, I is the current in Amperes, L is the length of the wire in meters and θ is the angle between the loop and the z axis. As the loop and magnet are both centred on the z axis and by symmetry the value of B is constant around the loop, then the length of the wire is the circumference of the loop, so $L = 2\pi r$ and the force equation simplifies to 16

$$F = 2\pi B i\ r \tag{16}$$

For a solenoid with multiple coils the magnetic field is B_z, the z component of the field of the permanent magnet. The force generated by the coil is the sum of all of the loops of the solenoid and is given by 17.

$$F = \sum_{i=1}^{i=n} 2\pi\ r\ i\ l\ B_z \tag{17}$$

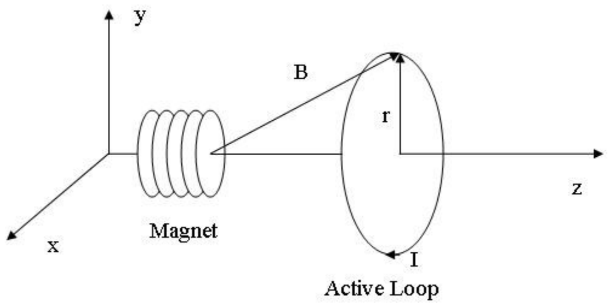


Figure 5: The force on a loop

C. Scaling up to a full size automobile

To determine the feasibility of the active e.m. damper as a component of a full size automobile, the two degree of freedom system that is developed previously, is used. The two larger magnets used above were modelled.

The car modelled is a lightweight electric vehicle and the values of the suspension system were the same as previously. These are given in table 5.5. The resonant frequency of the sprung and unsprung masses remain the same.

The magnets and coils used in the model are the same as used previously. The difference between the passive damper and the active dampers, is that the coils are powered to provide active damping.

The effectiveness of the active e.m. damper is determined by the reduction of acceleration in the sprung mass when compared to a passive damper that is suitable for use in the vehicle. As the vehicle is electrically powered, a second major consideration is the power consumption of the damper.

A random road profile is constructed using 12 different sine waves ranging from 0.5 Hz to 20 Hz. These were given a phase difference and then summed together. The profile is as illustrated in figure 6. This is then used to determine the average accelerations experienced by the passengers of the sprung mass for a passive system, for an active Skyhook damper and an active e.m. damper.

The passive damper and semi-active damper were the same as used previously. The active damper is modelled using the same N3522 and N5550 magnets that were previously. Two new dampers were modelled, one for each magnet. These dampers were constructed of several coils and one or magnets which travelled axially through the coils, as described previously. These dampers are described in table 4. The mass given is the mass of the copper wire and the magnets only. This does not include the connections between the vehicle and the magnet, nor the mounting points. Additional

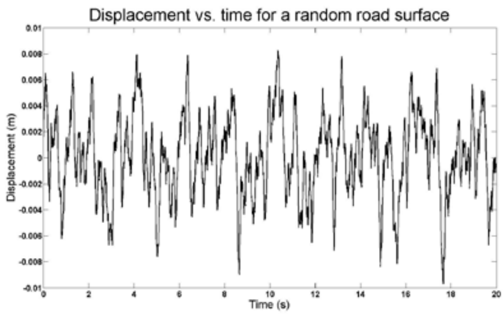


Figure 6: A modelled random road surface.

Table 4: The two active dampers modelled.

Property	ND3522	ND5550
Number of Magnets	5	1
Number of Coils	6	2
Number of Layers	5	5
Mass (kg)	0.9865	1.3683

Table 5 : Comparison between a passive, Skyhook and active e.m. damper. Sprung mass = 200kg. Unsprung mass = 25 kg.

Damper	R.m.s acceleration (m/s2)	Power consumption (W)
Passive	0.5597	0
Skyhook	0.2207	0
Active ND3522	0.0640	23
Active ND5550	0.0677	24

mass would be required for the controller and the power switching.

The determination of effectiveness is by comparing the vertical r.m.s. accelerations of the sprung mass of the dampers in the two degree of freedom system that were constructed earlier. A comparison of the dampers is given in table 5.

The two active e.m. dampers had very similar performance and power consumptions and produced a modelled reduction in r.m.s. acceleration of the sprung mass of 69 % when compared to the Skyhook damper and 88% when compared to the passive damper.

The mass of the copper and the magnets in the damper would be approximately 40 % of the mass the comparable passive damper discussed previously. This would allow extra mass for construction of the connectors and other elements required for a practical damper.

The average power consumption to achieve this damping is on the order of 24 Watts. For a vehicle with four wheels, this would produce a total continuous power production on the order of 100 W. In the case of a vehicle such as the light weight electric vehicle, this represents an increase of power consumption on the order of 1–2% under normal highway driving conditions. The final power consumption of the vehicle would be directly related to the amplitude of the displacement of the surface that is being traversed.

V. Conclusion

The use of purely passive systems on lightweight electric vehicles is problematic due to the mass of the vehicle. Semi-active damping offers a solution. While it offers some of the advantages of a fully active system. The use of an electromagnetic active suspension system potentially offers the performance of a fully active hydraulic suspension system with a great reduction in power consumption and mechanical complexity. The active e.m. damper produces better a 89% reduction in acceleration of the passenger compartment compared to a damping a passive damper. And a 71% reduction compared to a semi-active damper. For a similar power consumption.

References

[1] Sharp, R. and S.A. Hassan, The relative performance capabilities of passive, active and semi-active car suspension systems, *Proceedings Institution of Mechanical Engineers*, 200(D3), pp 219-228, (1986).

[2] Graves, K., P. Iovenitti, and D. Toncich. Electromagnetic regenerative damping in vehicle suspension systems. *International Journal of Vehicle Design*, **24(2-3)**, pp. 182–197 (2000b).

[3] Zuo, L., Scully, B., Shestani, J., & Zhou, Y. (2010). Design and characterization of an electromagnetic energy harvester for vehicle suspensions. *Smart Materials and Structures*, *19*(4), 045003.

[4] Fang, Z., Guo, X., Xu, L., & Zhang, H. (2013). Experimental study of damping and energy regeneration characteristics of a hydraulic electromagnetic shock absorber. *Advances in Mechanical Engineering*, *5*, 943528.

[5] Fow, A. J., Investigation into low power active electromagnetic damping for automotive applications. Diss. University of Waikato, 2015. suspensions. *Vehicle System Dynamics*, **15(1)**, pp. 41–54 (1986).

[6] Dixon,J., *The Shock Absorber Handbook*, SAE, Warrendale, Pa (1996)

[7] Dixon, J., *Tires, Suspension and Handling*, John Wiley and sons, Warrendale, Pa., second edition (1999)

[8] Gillespie, T.D., *Fundamentals of Vehicle Dynamics*, Society of Automotive Engineers,(1992)

[9] Karnopp, D., M. Crosby, and R. Harwood. Vibration control using semi-active force generators. *Transactions of the ASME: Journal for Industry for Industry*, **96(2)**, pp. 619–26 (1974).

[10] Krasnicki, E. J. Comparison of analytical and experimental results for a semi-active vibration isolator. *The Shock and Vibration Bulletin*, **50(9)**, pp. 69–76 (1980).

[11] Karnopp, D. Theoretical limitations in active vehicle

[12] Simon, D. E. and M. E. V. T. Ahmadian. An alternative semi-activecontrol method for sport utility vehicles. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, **216(2)**, pp. 129–39 (2002).

[13] Wang, D. H., & Liao, W. H. (2009). Semi-active suspension systems for railway vehicles using magnetorheological dampers. Part I: system integration and modelling. *Vehicle System Dynamics*, *47*(11), 1305-1325.

[14] Deprez, K., Moshou, D., Anthonis, J., De Baerdemaeker, J., & Ramon, H. (2005). Improvement of vibrational comfort on agricultural vehicles by passive and semi-active cabin suspensions. *Computers and electronics in agriculture*, *49*(3), 431-440.

[15] Valášek, M., M. Novak, Z. Šika, and O. Vaculin. Extended ground-hook-new concept of semi-active control of truck's suspension. *Vehicle system dynamics*, **27(5-6)**, pp. 289–303 (1997).

[16] Yao, G., F. F. Yap, G. Chen, W. Li, and S. Yeo. Mr damper and its application for semi-active control of vehicle suspension system. *Mechatronics*, **12**, pp. 963–73 (2002).

[17] Sapinski, B. An experimental electromagnetic induction device for a magnetorheological damper. *Journal of Theoretical and Applied Mechanics*, **46(4)**, pp. 933–47 (2008).

[18] Lee, H.-S. and S.-B. Choi. Control and response characteristics of a magnetorheological fluid damper for passenger vehicles. *Journal of Intelligent Material Systems and Structures*, **11**, pp. 80–87 (2000).

[19] Turnip, A., K. S. Hong, and S. Park. Control of a semi-active mr-damper suspension system: A new polynomial model. *Proceedings, The International Federation of Automatic Control*, pp. 4683–4688 (2008).

[20] Shutto, S, and J. R. Toscano. "MAGNETORHEOLOGICAL (MR) FLUID AND ITS APPLICATIONS." *Proceedings of the JFPS International Symposium on Fluid Power*. Vol. 2005. No. 6. The Japan Fluid Power System Society, (2005).

[21] Kuns, K. Calculation of magnetic field inside plasma chamber. *UCLA report*, **2(3)**, pp. 1–11 (2007).

[22] Williams, D. E. and W. M. Haddad. Active suspension control to improve vehicle ride and handling. *Vehicle System Dynamics*, **29**, pp. 1–24 (1997).

[23] Gysen, B., J. Janssen, J. Paulides, and E. Lomonova. Design aspects of an active electromagnetic suspension system for automotive applications. *IEEE Transactions on Industry Applications*, **25(5)**, pp. 1589–97 (2008a).

[24] Martins, I., J. Esteves, F. P. da Silva, and P. Verdelho. Electromagnetic hybrid active-passive vehicle suspension. In: *IEEE Vehicular Technology Conference*, volume 3, pp. 2273–7 (1999).