Feasibility Investigation of Energy Regenerative Hybrid Vehicle Suspension System

Quan Gan

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ABSTRACT

Feasibility Investigation of Energy Regenerative Hybrid Vehicle Suspension System

A conventional vehicle suspension system contains passive elements, namely a viscous damper and a spring. The damper converts vibration energy into heat energy due to the viscous friction of fluid in the device, and is finally dissipated to the external environment. Other than damper configuration, the amount of energy dissipated largely depends on the road roughness, the vehicle velocity and the vehicle mass. Under common operating conditions, this energy may be considered insignificant. On the other hand, over rough urban roads, the energy may be higher and thus presents a potential to tap the energy. For this, a hybrid linear motor/generator may be adapted to replace existing damper in the vehicle suspension system. With an appropriate controller, the device may be alternated between damping and generating device to provide damping as well as extracting energy. The energy even small may help in improving the efficiency of vehicle, especially electric and hybrid vehicle systems. With such device, it is also possible to vary the damping in a nonlinear and adaptive manner to accommodate the conflicting requirement between high speed operation and control of resonance.

This investigation examines the feasibility of a hybrid suspension damper namely, linear motor/generator in providing adequate damping for isolation of vibration while generate energy from relative motion between sprung and unsprung masses. The study utilizes a simplified quarter vehicle model with linear spring and the proposed

damping/generating device to illustrate its performance. The performances are evaluated in terms of acceleration transmissibility, rattle space, power spectral density (PSD) of acceleration response, and Dynamic Load Coefficient (DLC) to quantity the resulting pavement loads.

The regenerative damper is examined to establish its effectiveness in providing adequate damping as well as to evaluate its energy generation capability on realistic roads.

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NOMENCLATURE

Symbol	
F	Electromagnetic damping force
C_v	Electromagnetic damping coefficient
$\dot{X}$	Relative velocity
B	Magnetic flux density
V	Conducting volume
σ	Resistivity of the conducting material
V_p	Output voltage for the parallel electric circuit design
V_s	Output voltage for the series electric circuit design
N	Number of parallel conductors
R_e	External resistance
R_c	Internal resistance
E_c	Conductor voltage
M_s	Sprung mass
M_u	Unsprung mass
K_s	Spring stiffness
K_t	Tire Spring Stiffness
C_s	Nonlinear asymmetric Damping
C_t	Linear Tire Damping
C_1	Damping Coefficient at low piston velocity in compression
C_3	Damping Coefficient at low piston velocity in rebound
C_2	Damping Coefficient at high piston velocity in compression
C_4	Damping Coefficient at high piston velocity in rebound
$\dot{Z}$	The difference between velocity of sprung mass $\dot{X}_s$ and velocity of unsprung mass $\dot{X}_u$
a_1 , a_2	the damping coefficient changes from high to low values

X_s	The displacements of sprung mass
X_u	The displacements of unsprung mass
X_o	Roadway input
$\dot{X}_s$	Velocity of sprung mass
$\dot{X}_u$	Velocity of unsprung mass
$\ddot{X}_s$	Acceleration of sprung mass
$\ddot{X}_u$	Acceleration of unsprung mass
Δ_s	Static deflection due to the sprung mass.
g	Acceleration due to gravity
F_{ss}	Suspension spring force, ,
F_{sd}	Suspension damping force
F_t	Tire force
A	Wire cross-section area
R_{ave}	Mean radius of coil
L_m	Length of permanent magnet
L_c	Length of coil
T_c	Thickness of coil
Ψ	Fraction of wire areas over coil nominal cross sectional area
L	Length of the wire
ρ_c	Mass density of the wire material
i	current
E_{in}	Induced voltage generated across the coil
R_c	Resistance of the coil
R_e	External resistance connected to the coil
μ	Resistivity of the coil wire material

Ω	Constant and is defined as Equivalent Damping Coefficient, which reflects the damping characteristic of the linear motor.
M_{ext}	Externally connected mass
M_{mag}	Mass of the magnet
ω_n	Undamped system the natural frequency
ζ	damping factor, giving the ratio of system damping to that of critical damping of the system
E_b	Battery voltage
D	A full wave rectifier
V_t	Threshold of diodes
r	Battery inner charging resistance
r_b	Resistance of battery under charging state
E_d	Threshold for pair of diodes
E_{ch}	Threshold of the battery charging circuit, includes E_b and E_d
R_{total}	The sum of R_c, R_e and r_b
L_e	self-inductance of coil
T_d	Electrical delay time
P	Coil-induced flux
A_g	air gap area
L_g	air gap area length
μ_o	Permeability of free space

Chapter 1
Introduction and Literature Review

1.1 Introduction

The primary purpose of automotive suspension is to support vehicle weight and isolate vibrations transmitted to the chassis. For heavy vehicles, suspension properties also influence the dynamic wheel load and then also are considered as a measure of performance [1, 2]. The two major components of the suspension are spring and a damping device. The design of spring is relatively simple where the aim is to maintain low natural frequency of sprung mass while reasonable static deflection and rattle space are maintained [3]. The design of damping device, on the other hand, presents a challenge due to conflicting requirement over the frequency and between compression and extension strokes [1]. Such requirement has led to nonlinear asymmetric damping characteristics widely used in conventional vehicle suspension system. A typical characteristics of a conventional nonlinear damper in terms of force as a function of velocity is shown in Figure 1.1(a). The mean force-velocity characteristics of such damper as shown in Figure 1.1(b) can be easily idealized by piece-wise linear representation.

Suspension systems containing passive control elements, namely springs and nonlinear asymmetric dampers, have evolved to the point at which it seems reasonable to assume that they will not evolve much without changes in principle. In recent years, such changes in principle have become commercially viable with advances in transducers, processors and force generators. There are reports of numerous works addressing active, semi-active and adaptive suspension systems.

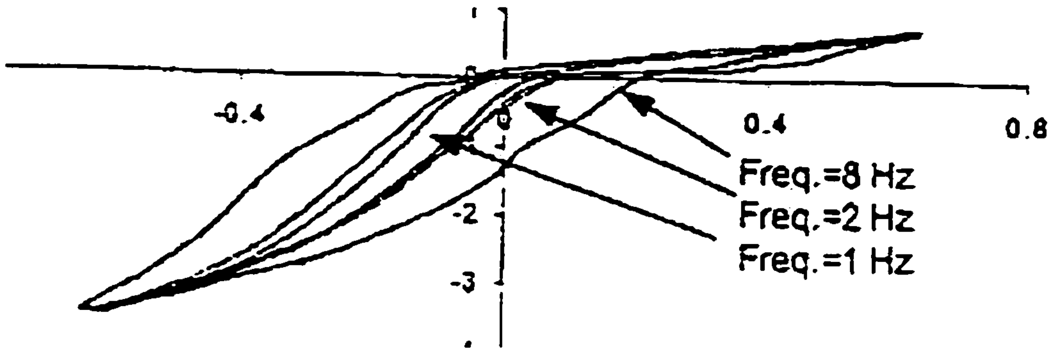

Figure 1.1 (a) Force-velocity characteristics of a suspension damper [4]

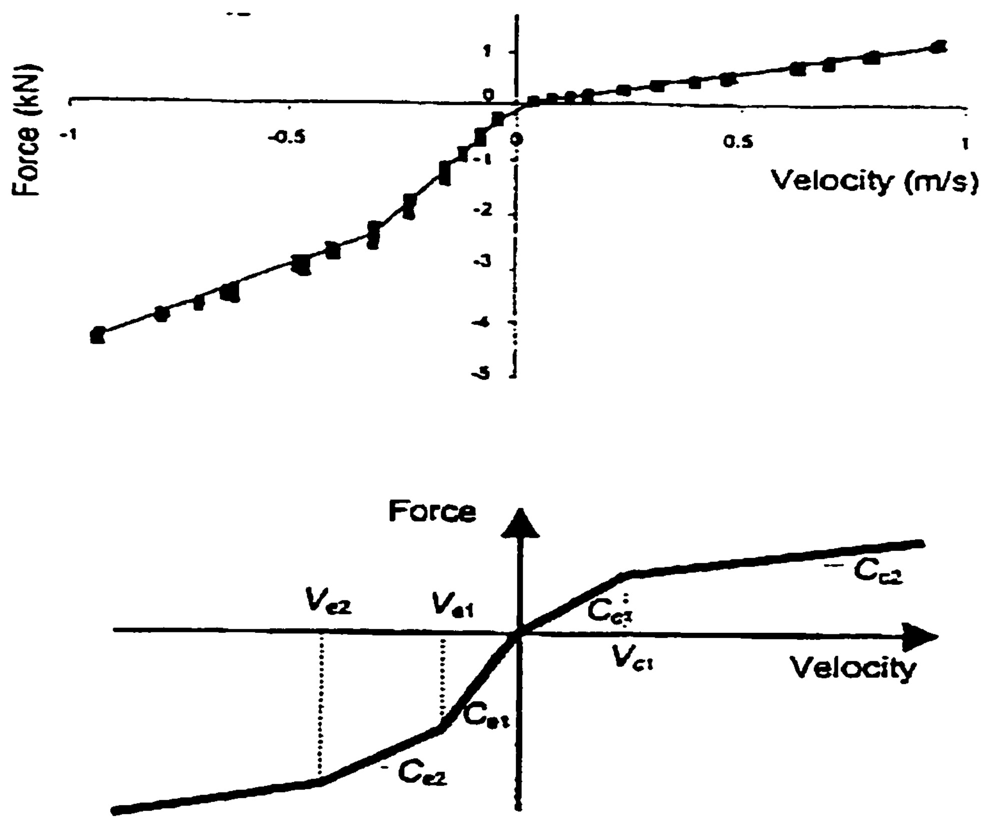

Figure 1.1 (b) Mean force-velocity characteristics and idealized representation [4]

In an active suspension system, an actuator replaces the normal spring and damper, and a control law is used for active isolation of motion experienced by the vehicle. Active vibration control has been shown to effectively suppress vibration through its high capability of reducing the vibration amplitudes and the control adaptiveness [1]. However, it has the drawbacks in producing feedback instability and requiring a large amount of energy [5, 6, 7, 8]. A semi-active system, on the other hand, uses conventional spring along with a rapidly adjustable damper. The requirement for inner loop feedback to control the damper is inherited from the fully active system. Semi-active system, requiring significantly less energy than the active system has been shown to improve the performance over the passive system in a wide frequency range and operating conditions [1, 9, 10]. Studies of these systems focus on performance improvement at the expense of sophisticated hardware and control system where additional energy must be supplied. Furthermore, these studies are not concerned with energy dissipation or the loss.

Some studies estimate on-road energy dissipation in automotive shock absorbers based on the methods of thermal measurements within the shock absorbers and simultaneous measurements of the force velocity characteristics [11, 12]. The energy dissipation rates have been found to vary from 3 to 57 watts under the effects of road roughness, vehicle velocity and vehicle mass [12].

With renewed interest on energy efficiency in recent years, engineers and researchers are examining options for improved efficiency at all levels. The best example is that of the developments of electric and hybrid vehicle systems. In these developments every watt of energy counts. Hybrid vehicles utilize the braking energy to generate electricity, which is stored [5]. Regardless of propulsion method, a need for more energy efficient vehicle motivates the studies of all energy dissipating vehicle elements as potential sources of energy recovery. The shock absorber is such an element. Many studies forecast the promising prospects of reusing the regenerated energy in form of electrical energy [5, 13, 14, 15]. Different energy regenerative devices have been introduced and discussed in these studies. Most mentioned regenerative devices, which work primarily as dampers, were electromagnetic devices, such as DC rotating motor and DC linear motor [7, 8, 16, 17]. A linear motor, when supplied with electrical energy provides force, while used as generator will provide electrical energy when subjected to linear motion. Depending on road roughness, the vehicle suspension is continuously subjected to relative motion. This motion of the mass brings a change in magnetic flux density through closed circuit, which in turn introduces the Lorentz force in a manner similar to that of a viscous damper [13, 17]. Therefore a DC motor or generator adopted as energy regenerative damper is suitable for this purpose.

This study focuses on an investigation to examine the potential of linear DC motor/generator to replace the damper in a conventional system. In the following subsections, a detailed literature review of vehicle suspension systems, regenerative devices and their modeling, as well as vehicle models for ride analysis are presented in sequence so as to develop the scope of the present investigation.

1.2 Literature Review

This section presents a review of research relevant to vehicle suspension, regenerative damping device, its design and modeling, as well as vehicle ride models.

1.2.1 Vehicle Suspension System

The primary elements that play a role in the ride quality of road vehicles include the tire properties and most importantly, the suspension elements. The common suspension elements include a parallel combination of a spring and a passive damping device. Heavy vehicle suspension using air springs may contain addition self-leveling system which is designed to compensate for variations in static load [1]. Suspension studies reveal conflicts between the ride comfort and working space, in other words, for a given road roughness and vehicle speed, comfort can be increased by reducing stiffness while the required suspension working space is increased [18, 19]. For a given stiffness, the damping which is optimal for comfort is less than that which is optimal for control of variations in wheel load [20]. Furthermore, when the spring stiffness is lowered, the difference between the damping, which is optimized for comfort, and one, which is optimized for the wheel load, increases. A study [21] examining the conflict concluded that for a conventional suspension, the damping to maximize comfort is 46 % of the damping that minimize variation in wheel load. Damper designed for comfort could result in over 20% sacrifice in the rms dynamic tire load. Most recent studies with heavy vehicles consider tire loads to be an important performance measure due to its influence on pavement damage. It is however, a concern for only heavy commercial vehicles.

Many suspension system researchers have explored the potential of advanced systems, such as active control and semi-active suspension damper. Comparative studies indicate that active system performance is considerably better than that of a standard passive system [22]. Practical active systems are likely to be electro-hydraulic [23], involving actuators and a number of precision engineering component for sensing and control. Active system has also been explored in application to seat suspensions. One such study [6] estimated the theoretical power consumption to be 1863 Watts. The experimentally measured power consumption was, however, found to be 2000 watts. This power requirement for a seat suspension clearly indicated the magnitude of the power that may be required for such a system in application to vehicle suspension. Due to power requirement, complexity and expense, such systems often remained academic in suspension applications.

As a compromise between an active and passive system, significant studies were also carried out to explore semi-active concepts for vehicle suspensions [10, 24, 25, 26]. A semi-active system focuses on the conflicting damping requirement for improved ride quality over a wide frequency range. Such system utilizes conventional viscous dampers where the damping is modified as needed or predicted by an established control law [24]. The control law based on sprung and unsprung mass motion can be designed with appropriate weighting to improve performance in terms of ride, road holding and wheel load [25]. Similar to active systems, semi-active suspensions also require sensors, control system and variable or adjustable damping device [24]. Such systems thus also demand power for its operation.

Since the focus of this investigation is to explore energy generation from the suspension element that dissipates energy, it is necessary to search for alternate devices than those used as conventional passive and semi-active systems. The following subsections thus presents review of literature on regenerative damping devices, their design and analysis.

1.2.2 Regenerative Damping Device

1.2.2.1 Introduction

Passive or semi-active vehicle suspension system contains a damper, which is a viscous element. The viscous damper converts vibrational energy into heat energy due to the flow of fluid through orifice and is generally dissipated to the external environment. Using experimental force-velocity measurements across a vehicle damper, it has been found that energy dissipation is between 3 and 57 watts per damper [12, 27], and approximately 200 watts of power is dissipated by the four shock absorbers of a passenger car traversing a poor roadway at 13.4 m/s (30mph) [12]. The amount of energy dissipated largely depends on road roughness, vehicle velocity and vehicle mass. Under common conditions, energy dissipation between 10 to 15 watts per damper was reported [16]. This energy may be considered insignificant in comparison to the overall energy consumption of a vehicle with internal combustion engine. However, the regeneration of the energy can be used as a power source for low power accessories as defrosters, windshield wipers, headlights, stereos and heaters, etc. Furthermore, the regenerative energy is perhaps more relevant to electric or hybrid vehicles, where the total energy consumption is lower, and energy efficiency is a primary concern [27].

Electric vehicles could directly benefit from an extra electrical energy supply, with the advantage of reducing battery weight and extending vehicle operating range [16].

1.2.2.2 Design and Analysis

Few research groups have investigated the options of using vehicle suspension as a device for energy regeneration. Such a device called a Variable Linear Transmission (VLT) was presented in [5]. The concept of VLT as shown in Figure 1.2 consisted of a mechanical power absorber, a movable fulcrum and a lever. Here, mechanical power absorber is essentially an accumulator that can store energy in term of pressure. Although it is not very clear, the VLT is proposed to meet the barrier potential of an energy storage device while maintaining required damping force. Although a mathematical model for the theoretical analysis of the system was set up, the experimental verification was not achieved. An ideal VLT could provide viscous damping while storing all of the energy normally dissipated by an equivalent passive viscous damper. But the results with more realistic models were not so successful because the device would consume more energy than it would absorb [28].

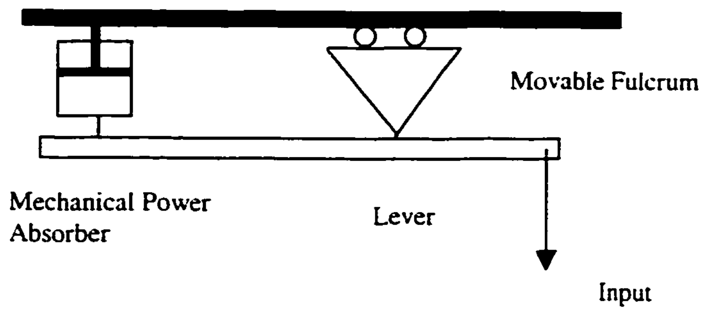

Figure1.2 Variable Linear Transmission [5]

For conversion of kinetic energy into electrical energy to be stored for long periods through use of rechargeable batteries, the most efficient energy regeneration method would require electromagnetic devices [16]. Energy conversion from kinetic energy into electrical energy is obtained under the condition that the motion brings a change in magnetic flux density through closed circuit [13]. The resulting Lorentz force affects the mass the same way as ordinary viscous damper. Thus the regenerative electromagnetic device must have a force—velocity characteristic similar to that of a conventional viscous damper. The force generated by such a device can therefore be expressed as:

$$F = C_v \dot{X} \tag{1.2.1}$$

Where $\dot{X}$ is the relative velocity across the device and C_v is a coefficient dependent on magnet and coil parameters.

An electromagnetic device placed between the sprung and unsprung masses of the vehicle will undergo linear motions. Use of rotational regenerative device will thus require conversion of linear motion to rotational motion and vice-versa. Alternatively, a linear motor replacing the damper in vehicle suspension is more practical for both energy generation and applying damping force.

Linear electromagnetic motors consisting of coils of copper wire interacting with magnetic fields produced by permanent magnets can be used to construct mechanical dampers [17]. For a "Moving Coil" electromagnetic damper, as shown in Figure 1.3, the maximum force resisting the motion on velocity can be expressed as [17]:

$$F_{max} = \frac{B^2 V}{\sigma} \dot{X} \qquad (1.2.2)$$

Where, B is the magnetic flux density, $\dot{X}$ is the relative velocity between the magnetic flux and conducting coil, V is the conducting volume, and σ is the resistivity of the conducting material. Equation (1.2.2) reveals that the electromagnetic device produces a force proportional to the relative velocity where the term $(\frac{B^2 V}{\sigma})$ can be treated as the damping constant for the damping element of the suspension.

Equation (1.2.2) further shows that for a fixed conducting volume (V) and resistivity of the material (σ), the magnetic flux density (B) can be changed to produce different damping coefficient. Furthermore, it is relatively easy to change the flux density when an electromagnet is used, and thus makes it more attractive for application in vehicle suspension.

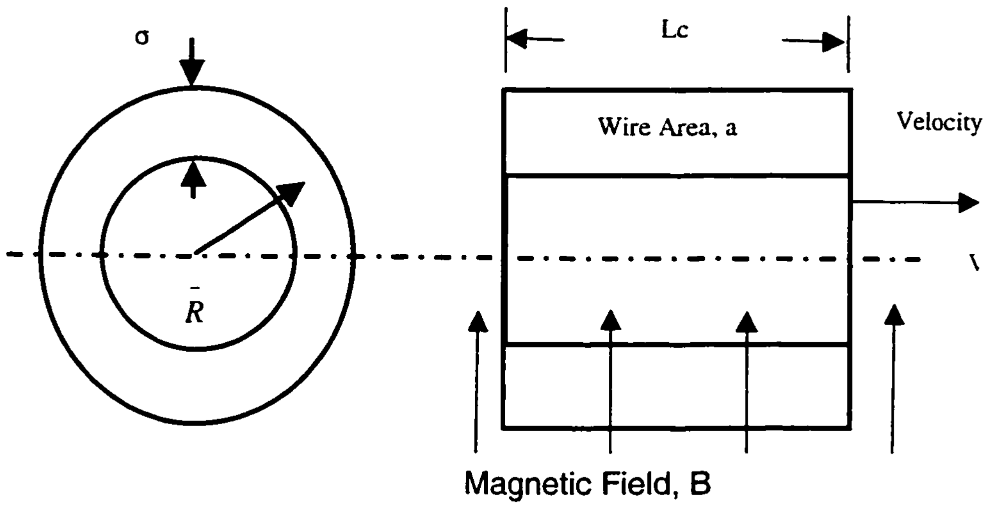

Figure 1.3 Moving Coil Design

For the electric circuit design of the regenerative device, there are two main modes: the Parallel Design Mode, such as "Radial Fin" or "Faraday Disk", and the Series Design Mode, such as "Rotating Motor", and "Moving Coil" [16]. The series design consists of one long length of conductor with the two ends providing the output terminals of the device; and the parallel design consists of more than one conducting length joined in parallel, with each element connected directly to the output terminals. The schematic circuits for the two designs are shown in Figure 1.4. The output voltage (Vp) for the parallel design may be expressed as [16]:

$$V_p = \frac{NE_c R_e}{r + NR_e} \tag{1.2.3}$$

The above equation shows that the output voltage for the parallel design is a function of the number of parallel conductors, N, conductor voltage, E_c and the internal and external resistances denoted by r and R_e respectively. E_c is proportional to the product of the magnetic field strength B, the length of the wire L and the velocity of the coil through the magnetic field $\dot{Z}$. As shown as [17], the conductor voltage E_c can be expressed as:

$$E_c = BL\dot{Z} \tag{1.2.4}$$

On the other hand, for series design, the output voltage will be much larger than the parallel design due to the increased conductor length L and is expressed as [16]:

$$Vs = \frac{E_c R_e}{R_e + R_c} \tag{1.2.5}$$

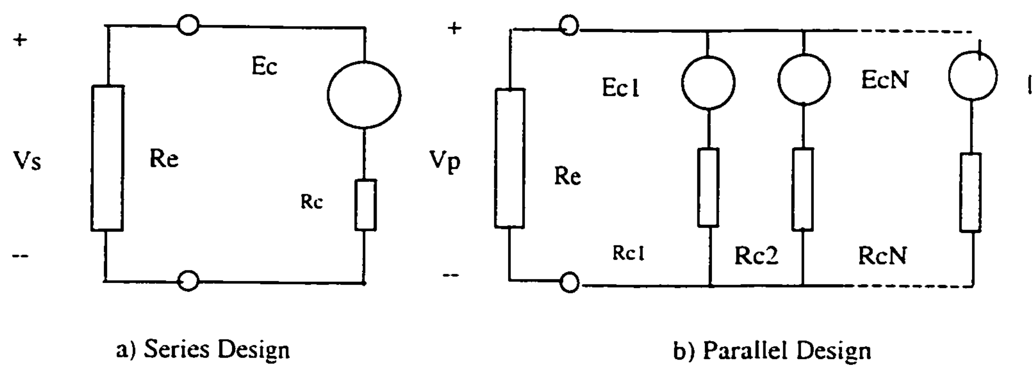

Figure 1.4 Electromagnetic Circuits

From the analysis and simple examples presented in [16], it can be concluded that although the force generated by both series and parallel configurations is similar, the voltage generated by them is significantly different. It seems that the output voltage of the series design regenerative damper can continually meet the potential of the vehicle battery storage device, usually 12 volts. However, the methodology in order to achieve such a performance is not being discussed or reported.

For a moving coil in a uniform B-field stretching over the entire range of travel of the coils is the most effective scheme for generating damping forces. The moving magnet system with high-energy product magnets can be almost equally effective unless very long strokes are required [17].

In a study [13] of energy regenerative damper, a DC motor was modeled and tested to determine the damping coefficients and efficiency. The performance of energy regenerative damper was formulated with the parameters of DC motor. The characteristics of energy regenerative damper were verified by experiments. Experimental results showed good agreement with theoretical predictions. The experimental results showed that the energy regenerative damper had an adequate damping effect. The calculated damping coefficient was 110 (Ns/m), which correlated well with the theoretical damping coefficient of 99.44 (Ns/m) [13]. The efficiency of the regenerative damper was also found to be encouraging. A set of results giving efficiency versus input frequency revealed that the efficiency was close to the theoretical maximum efficiency (25 % in this example) for low frequencies [13].

A rotational DC motor although not ideal for direct application in suspension with linear motions, it does present the potential of amplification of the motion through gear ratio. One such configuration studied using a single degree of freedom system is shown in Figure 1.5(a) [13]. The study showed significant influence of the rotating inertia on the dynamic force of the damper. In order to improve the performance, a further study included additional spring (Kr) and damper (Cr) as shown in Figure 1.5(b) [28].

The results of analysis reveal that by using the hybrid suspension system, the dynamic response of the modified rotating system is almost identical to the linear system. Therefore it is possible to have a mechanically amplified, rotating, regenerative damper in a suspension system without the negative effect of rotating system dynamics. But the added spring—damper system will also prevent the input disturbance being transmitted to the regenerative damper and dissipate the vibration energy in the added damper [28]. Furthermore, it is difficult to make constructive conclusion from this study as it does not consider the coupled motions between the sprung and unsprung masses.

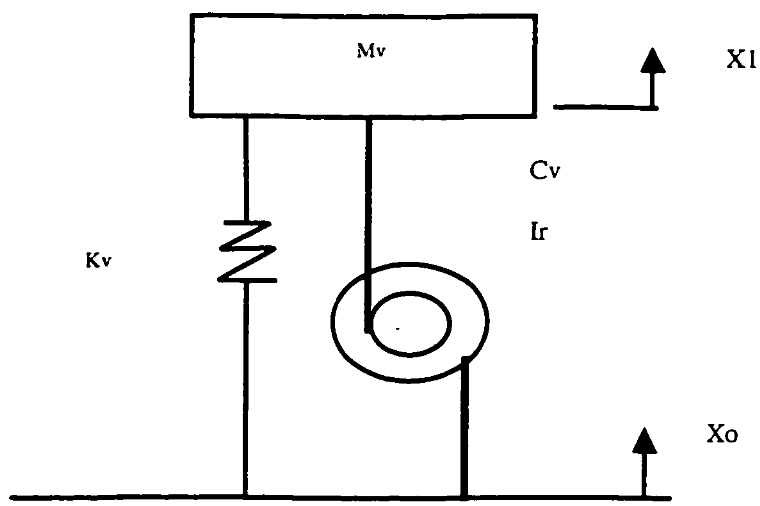

Figure 1.5 (a) One DOF Rotating damper [13]

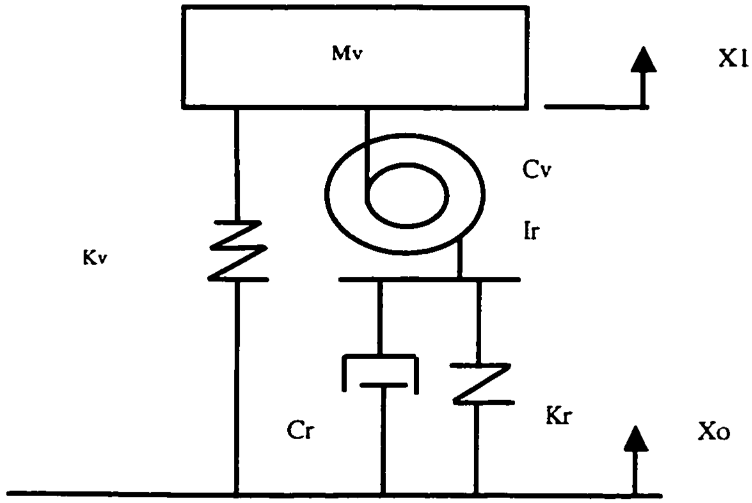

Figure 1.5 (b) Rotating damper with added damper and spring [28]

The regenerative damping device in the form of linear motor will avoid some of the complications encountered in [28]. The study must however consider a more comprehensive vehicle model that can reproduce the relative motion under realistic road excitations.

1.2.3 Vehicle Models

The studies of ride quality and pavement-vehicle interaction involve the development of a representative dynamic model that can closely describe the vehicle behavior. In vehicle research, many models were adopted from two degrees of freedom (DOF) linear quarter vehicle model to the complex three-dimensional models with up to 19 DOF. Since the simple one-dimensional vehicle model can not be used to predict the complex dynamics associated with real vehicles, a number of comprehensive two and three dimensional vehicle models have been developed to study the tire force and ride quality of the vehicles. Vehicle models with limited number of DOF, which are realistic enough to provide reasonable estimate of the tire force characteristics and ride quality, are desirable for design and optimization studies [11, 29]. Many research studies have demonstrated that a four DOF in plane model of a single unit can be effectively used to determine vehicle behavior pertaining to dynamic tire loads and ride quality [30, 31]. Assuming vehicle structure as rigid body for practical range of low ride frequencies, a 7 DOF model is considered comprehensive. This model including bounce, pitch and roll of sprung mass and bounce of each unsprung mass is well equipped to predict the ride behavior of ground vehicles. For study of suspension system, however, the model can be greatly simplified if pitch and roll influence of the vehicle mass is neglected. Such model commonly referred to as a 2 DOF quarter-vehicle model consisting sprung and unsprung masses, suspension and tire properties are widely need for suspension studies. [1, 3, 7, 14]. Such models with realistic suspension elements and ties properties are often considered adequate for suspension studies. Quarter vehicle model can predict qualitative ride behavior as well as suspension relative motions and tire loads. [32]. The simplicity of this model presents an opportunity to introduce complexity in the suspension parameters while using efficient analysis tools.

Other than suspension, tire properties play an important role in ride studies. Review of literature indicates that the tire models can be grouped under four categories based upon the characterization of the tire-road contact. These models are referred to as: point contact; rigid tread band; fixed foot print; and adaptive foot print model. For a typical car tire, contact length of 0.25m, will provide a significant filtering of the road profile input for wavelengths shorter than 0.75m [2]. When input frequencies are greater than 80 Hz, tire carcass resonance can occur. [1, 33] Within the speed and frequency ranges of study, it is adequate to consider the tire making contact with the ground at a point. Use of point contact tire models are thus often considered appropriate for analysis of ride and dynamic tire forces. Linear or nonlinear tire models based upon the point contact are utilized depending on the consideration of wheel hop conditions.

The vertical stiffness of a tire is quite closely related to its load and the range of ratios for tire stiffness to weight supported is typically 40 to 50 m^{-1} [1]. Typical sprung mass to unsprung mass ratios for cars lie in the range of 5 to 8, for smaller and larger cars, respectively.

1.3 Scope of the thesis

As discussed in previous sections, the focus of the thesis research is to explore the feasibility of energy regenerative device as suspension damping element. The most ideal such device would be a linear motor utilized in parallel with the suspension spring. A two DOF quarter-vehicle model will be employed with point contact tire model and linear suspension spring. A detailed model developed for the regenerative device incorporated within the simplified vehicle ride model will be investigated.

The overall objective of this dissertation research is to analysis the ride performance and road friendliness of the vehicle with energy regenerative damper and to evaluate the energy regenerating efficiency of the regenerative damper system. As discussed in the literature review, the damping in vehicle suspension has conflicting requirements and a nonlinear characteristic similar to one as shown in Figure 1.1 is commonly used. As discussed in section 1.2.2.2, a regenerative damper can be easily controlled to create desired characteristics for the damper. An analogy between the regenerative damper system and the conventional nonlinear viscous damper system is established in order to illustrate the adaptability and the potential benefit of the regenerative damper system in the fields of both shock absorbing, ride performance and energy reuse. Here the proposed linear motor as damper is used to provide damping as well as extract energy, and hence referred to as a hybrid regenerative suspension element.

The feasibility study is carried out in sequence as presented in the following. The objectives can be outlined as:

1. To formulate an electromagnetic damper model and to analyze its force velocity characteristics, which leads to the study on the feasibility of the electromagnetic damper in a vehicle model.

2. To study and establish electromagnetic parameters for their influence on the fluctuation of the electromagnetic damping coefficient and regenerative characteristics.

3. To incorporate the developed hybrid damping model within a simplified 2DOF quarter car model.

4. To establish the comparison in ride performance and dynamic road load between the regenerative dampers and a conventional nonlinear damper, and to study the feasibility of the regenerative damper in a vehicle system.

5. To design the basic electrical circuit whose primary purpose is to yield controllable electromagnetic induced force as damping force based on the feedback of velocity. The circuit should also provide means to capture the energy generated by the hybrid device as the mechanical motion is converted to electrical energy.

6. To carry out a detailed parametric study and to evaluate the energy regenerative efficiency of the electromagnetic suspension system with respect to different road profiles and to identify the main factors affecting the regenerative efficiency of the device.

7. To conclude on the application potential of such device in vehicle suspension application and its limitation.

1.4 Layout of the Thesis

In Chapter 2, a quarter-vehicle model with a regenerative damping system replacing the conventional nonlinear viscous damping system is formulated with the purpose of assessing its ride performance. A multi-stage nonlinear and asymmetric force-velocity characteristic for the damper is identified, and electromagnetic parameters are established to reproduce similar damping characteristics.

In Chapter 3, the ride quality and dynamic tire load of the vehicle with regenerative damping system are analytically investigated. A comparative study is carried out between the ride characteristics of regenerative damping system and those of a conventional nonlinear viscous damping under various road roughness.

Chapter 4 presents a discussion on the energy storage elements. Then power distribution of the electrical elements in the circuit is analyzed to establish power dissipation in resistance and rms of power generated for the purpose of storage. Study on the energy regenerative efficiency of the regenerative damping system under different road roughness profile is finally evaluated and analyzed to examine the feasibility of such concept as energy regenerative device in vehicle suspension.

The conclusions and recommendations for future work are outlined in Chapter 5.

Chapter 2
Formulation of Vehicle Model

2.1 Introduction

The objective of the investigation is to explore the feasibility of a hybrid regenerative damping device in vehicle suspension application. It is therefore necessary to formulate a credible vehicle ride model capable of exhibiting ride performance in term of sprung and unsprung mass motions. As it was suggested in the previous chapter, a two DOF quarter-vehicle model should be adequate to explore the feasibility. The performance of the proposed configuration is to be evaluated in terms of ride vibration under road excitations, resulting suspension motion and road loads and most importantly, the ability of the hybrid regenerative damping device to produce adequate damping. Here regenerative damping device is used to generate energy from the suspension motion, and hence the model must include provision to determine and establish the energy level and the efficiency of the energy regeneration.

The mathematical model formulated in this investigation includes a two DOF vehicle model with damper replaced by a linear motor. A detail model of the linear motor is developed, formulated and incorporated within the vehicle model. A model is also formulated with conventional nonlinear damper for comparison of results with those of the proposed system.

Energy regeneration of the regenerative damping system is associated with the parameters of vehicle model and strongly related to the roughness of the road profile and operating speed. Three road profiles, namely sinusoidal and random profiles characterized as city and off road profiles, to be used in the present investigation are presented and discussed.

Finally for a feedback control system, such as regenerative damper, there may be delay in system response. The aspect of time delay is thus presented and discussed.

2.2 Characteristics of Vehicle Model

A two DOF vehicle model with conventional suspension system is first presented here. The model can readily be validated for reliability and used for comparison with the results of proposed system. This model will be next extended to replace the conventional damping device with the proposed regenerative damping device.

The two DOF ride model as shown in Figure 2.1 consists of sprung mass M_s representing the mass of the vehicle supported by the suspension springs. Unsprung mass M_u represents the combined effective mass of the wheels, tires, brakes etc. The suspension stiffness, tire stiffness and tire damping are considered to be linear while the suspension damping is nonlinear with multiphase damping characteristics. The masses may represent the complete vehicle or quarter of the vehicle. In the latter case, the unsprung mass and suspension element represents one wheel assembly.

Numerous studies [24, 34, 35] have utilized such a model for comparison of various suspension control configurations, and to predict passenger isolation, suspension stroke, and tire contact force. This model with linearized elements is frequently used for evaluation of frequency response. Frequency response is a widely used performance measure adopted for comparison of different suspension concepts. The values that have been suggested in [36] for the quarter car model masses, stiffness, and suspension damping representing a typical passenger vehicle are utilized in this investigation. These values are also adopted to provide direct comparison of regenerative damped suspension performance with published performance characteristics for conventional passive suspension. The baseline parameters [36] for the quarter car model used in this investigation are presented in Table 2.1, except for the suspension damping. As discussed in the literature review in Chapter 1, conventional suspension dampers are nonlinear with asymmetric characteristics in compression and extension. A representative characteristic for the damper is shown in Figure 2.2. Such characteristics in a passive damper are determined through design of bleed control and blowoff valves [36]. This allows a reduction in damping at higher frequencies for improved ride quality while asymmetricity ensures improved road holding. The transition from high to low damping in compression and extension occurs at certain preset velocities.

As discussed in Chapter 1 and shown in Figure 2.2, the asymmetric nonlinear damper characteristics can be represented by four stage piecewise linear segments. The coefficient for each segment can be established from the force-velocity characteristics. A set of parameters obtained from [36] and presented in Table 2.2 are utilized in this investigation for simulation of conventional damper performance; and to establish the characteristics for the regenerative damping device.

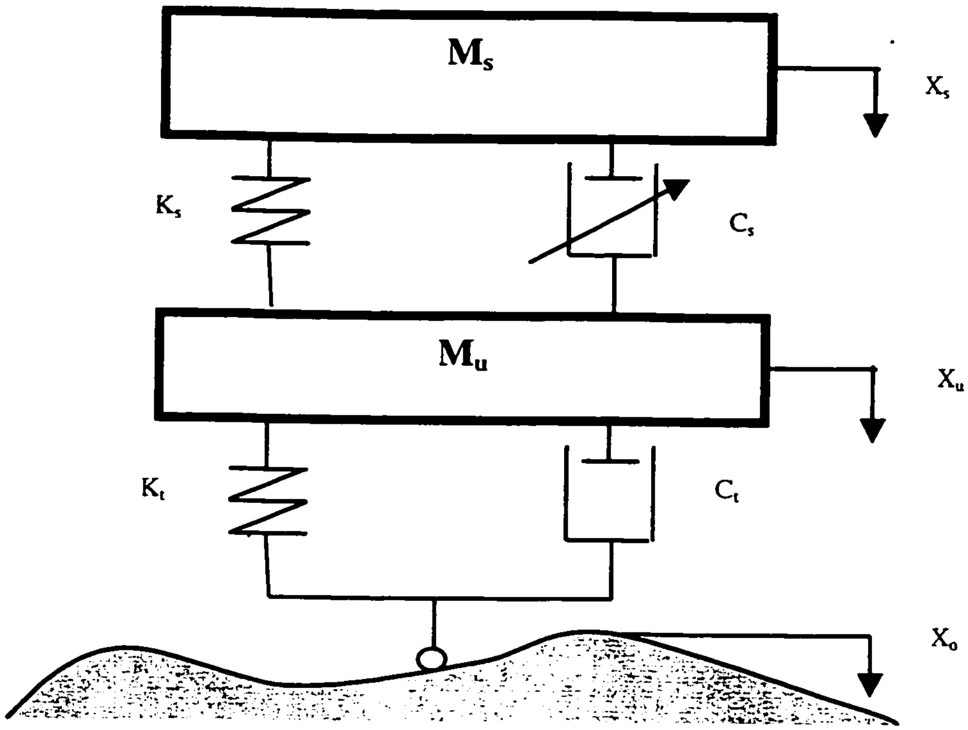

Figure 2.1 Two DOF Quarter Vehicle Model

Table 2.1 Vehicle Parameters for a quarter car model

Vehicle Parameters		
Sprung Mass	M_s	240 Kg
Unsprung Mass	M_u	40 Kg
Spring Stiffness	K_s	16 KN/m
Tire Spring Stiffness	K_t	120 KN/m
Nonlinear asymmetric Damping	C_s	Shown in Figure 2.2 & Table 2.2
Linear Tire Damping	C_t	200 Ns/m

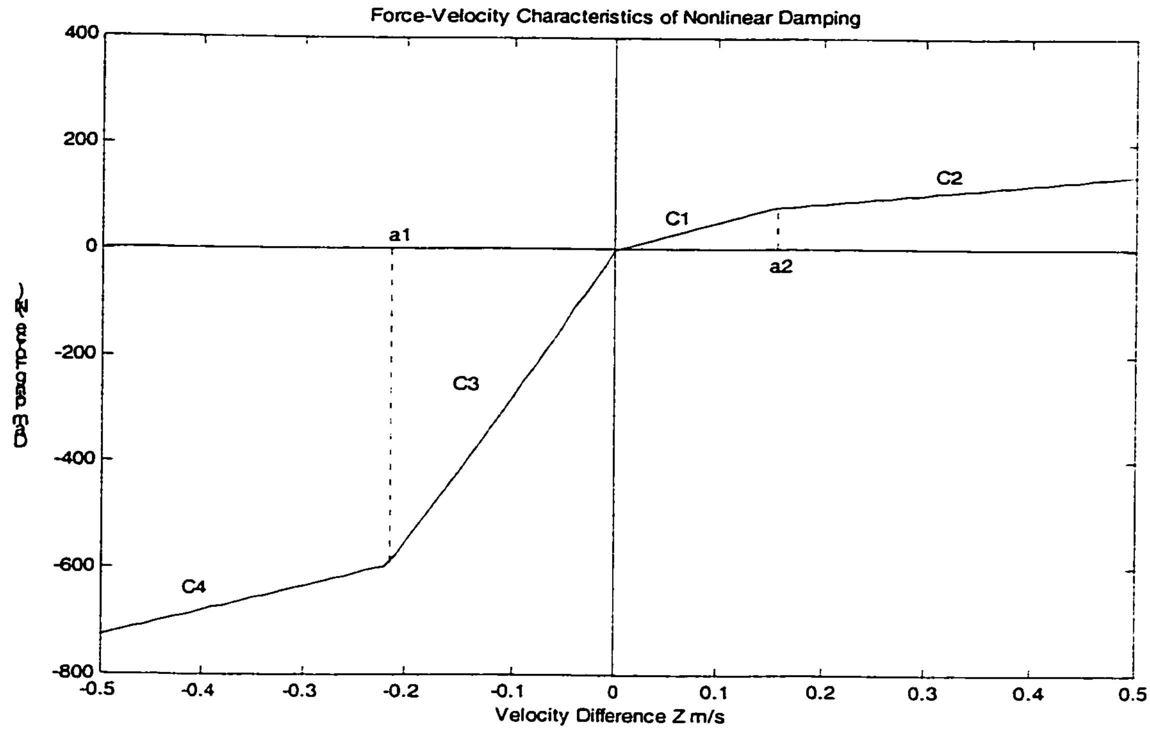

Figure 2. 2 Force-Velocity Characteristics of nonlinear damper

Table 2.2 Asymmetric Nonlinear Damping Parameters

Asymmetric Nonlinear Damper parameters	
Describe of Parameter	Values
Damping Coefficient at low piston velocity in compression C_1	514.5Ns/m a_1 =0.1524 m/s
Damping Coefficient at low piston velocity in rebound C_3	2747.5Ns/m a_2 =-0.2163 m/s
Damping Coefficient at high piston velocity in compression C_2	176.75Ns/m
Damping Coefficient at high piston velocity in rebound C_4	462Ns/m

For the four linear segments, the damping force of the suspension system can be expressed as:

$$F_{sd} = \begin{cases} C_1 \dot{Z} & 0 < \dot{Z} < a_1 \\ C_3 \dot{Z} & a_2 < \dot{Z} < 0 \\ C_1 a_1 + C_2(\dot{Z} - a_1) & \dot{Z} > a_1 \\ C_3 a_2 + C_4[\dot{Z} - a_2 sign(\dot{Z})] & \dot{Z} < a_2 \end{cases} \qquad (2.2.1)$$

Where $\dot{Z}$ is the difference between velocity of sprung mass $\dot{X}_s$ and velocity of unsprung mass $\dot{X}_u$. C_1, C_2, C_3, and C_4 are the damping coefficients. At certain preset velocities, a_1 and a_2, the damping coefficient changes from high to low values. The sign function assumes a value of either 1 or −1 for positive and negative values of velocity to ensure the phase relationship between the damping force and the velocity.

The linear DC motor proposed in this investigation to replace the damper in conventional suspension can be configured to provide damping force velocity characteristics similar to those shown in Figure 2.2 and expressed by Equation (2.2.1). The working principle of the regenerative damping element is different from the principle of hydraulic shock absorbers, where the damping characteristic is realized by changing the controllable electrical components of the regenerative damping device. The most commonly used controllable electrical components are variable resistance. In the following subsection, a general model of quarter vehicle is formulated. It is then followed by sequential development, formulation and analysis of the regenerative damping device.

2.3 Analytical Vehicle Model

A quarter vehicle model contains the most basic features of the vehicle ride behavior for suspension system study. In this thesis, assuming negligible contributions of roll and pitch dynamics of the vehicle, a quarter vehicle model is formulated to study the characteristics of the regenerative damping in vehicle ride performance, and its potential of energy regeneration. A generic representation of the model is presented in Figure 2.3, where the suspension damping element Cs may be a linear or nonlinear conventional damper, or regenerative damping device. The vehicle body is characterized by a rigid sprung mass M_s. The wheel and axle assembly is represented by another rigid mass, referred as unsprung mass M_u. The sprung mass is coupled to the unsprung mass through the suspension components, modeled as parallel combination of energy restoring spring, and energy dissipative element, or the regenerative damper with energy regenerative capability. The tire is modeled as damped elastic element, assuming a point contact with the road.

The equations of motion for the two-DOF quarter vehicle model shown in Figure 2.3, can be easily obtained by applying Newton's second law. The two equations describing motions X_s and X_u are:

$$M_s \ddot{X}_s + F_{sd}(\dot{X}_s, \dot{X}_u) + F_{ss}(X_s, X_u) - M_s g = 0 \quad (2.3.1)$$

$$M_u \ddot{X}_u - F_{sd}(\dot{X}_s, \dot{X}_u) - F_{ss}(X_s, X_u) + F_t(\dot{X}_u, \dot{X}_o, X_u, X_o) - M_u g = 0 \quad (2.3.2)$$

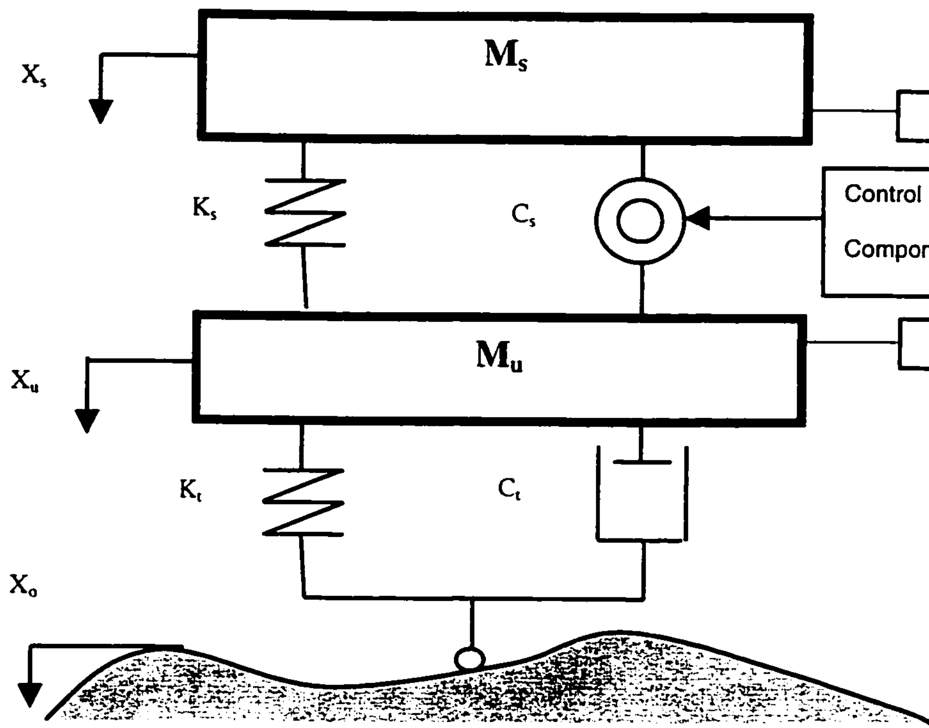

Figure 2.3 Two DOF Quarter Vehicle Model with Regenerative Damper

Where M_s and M_u are the masses due to sprung weight of the vehicle and the wheel assembly respectively. F_{ss}, F_{sd} and F_t represent the suspension spring force, suspension damping force, and tire force respectively. For linear suspension spring, the force is:

$$F_{ss}(X_s, X_u) = K_s(X_s - X_u + \Delta_s) \quad (2.3.3)$$

Where K_s is the linear spring rate of the suspension, X_s and X_u are the displacements of sprung and unsprung masses respectively. Here Δ_s is the static deflection due to the sprung mass.

$$\Delta_s = {-m_s g}/{K_s} \tag{2.3.4}$$

The damping force F_{sd} developed by linear suspension damping can be expressed as:

$$F_{sd}(\dot{X}_s, \dot{X}_u) = Cs(\dot{X}_s - \dot{X}_u) = C_s(\dot{Z}) \tag{2.3.5}$$

Where $\dot{Z}$ is the relative velocity across the damper and C_s is the equivalent damping coefficient. For a nonlinear asymmetric damper, the damping force expression is to be replaced by the piece-wise linear expressions presented in equation 2.2.1. In the case of an regenerative damper, the expression for C_s will be established in the following subsection.

The tire is modeled as point contact with the road and as a parallel combination of a linear spring and a linear damper. The force F_t developed by the tire is a function of relative motion between the unsprung mass (X_u) and road profile (X_o).

Although the tire elements are considered linear, in reality, the tire can not generate any force when wheel lifts of the road as there is a loss of contact. This behavior can be easily incorporated in the model by describing the tire force with two conditions as follows:

$$F_t(\dot{X}_u, \dot{X}_o, X_u, X_o) = K_t(X_u - X_o + \Delta_t) + C_t(\dot{X}_u - \dot{X}_o)\, when\ (X_u - X_o + \Delta_t) \le 0 \quad (2.3.6)$$

$$F_t(\dot{X}_u, \dot{X}_o, X_u, X_o) = 0 \qquad when(X_u - X_o + \Delta_t) > 0 \quad (2.3.7)$$

Where K_t is the stiffness of linear spring of the tire, X_o is the displacement excitation encountered at tire road interface. Δ_t is static deflection of the tire, which can be expressed as:

$$\Delta_t = \frac{-(M_s + M_u)g}{K_t} \qquad (2.3.8)$$

Equations (2.2.1) to (2.3.8) describe the vertical dynamics of the quarter vehicle. Important responses from such model include: spring mass acceleration ($\ddot{X}_s$) which is a direct measure of ride quality; relative motion Z= (X_s – X_u) describing motion of the suspension system; and dynamic tire load (F_t) indicating the dynamic tire load developed by the vehicle suspension system.

2.4 Regenerative Damping Device

A regenerative damping device proposed in this investigation to replace the viscous damper of the vehicle suspension is a linear DC motor/generator. Suitable linear motors are not readily available and should be designed for this purpose [37]. Linear electrodynamics motors consist of coils of copper wire interacting with magnetic fields produced by permanent magnets or electromagnets. In order to meet the required multi-stage damping characteristics, the damping coefficient of these linear motors working as mechanical damping should be rapidly varied. This can be achieved by changing the external resistance connected to the coil of the motors as suggested in [17]. When an electrodynamics motor coil is short circuited or connected to an external resistor, it becomes a linear mechanical damper. In the close circuit state, the damping coefficient will vary according to the change in external resistance. When the electric circuit is on open state, the damping coefficient vanishes, but when the coil is short circuited, the damping coefficient reaches the maximum value. Since the external resistance can be rapidly varied electronically, the electrical motor can work as a semi-active damper in vehicle suspension system [15, 38, 39]. The damper here is referred to as semi active since the damping coefficient will be varied based on system response.

Finally, the suspension model with regenerative damper should predict the power regenerated by the regenerative damper. It calls for an efficient energy charging circuit and storing device to evaluate the energy regenerative characteristics. Automobile 12-volt battery is chosen as the regenerated energy-storing device and the detail device layout and charging circuit are discussed in following sections.

2.4.1 Regenerative Damper Layout Design

A layout of "moving coil" introduced in [16, 17] can be adapted for the design of proposed damper. This concept is most often used in loudspeakers, in which a coil of conducting material moves linearly within a radial magnetic field. A coil moving in a uniform magnetic field stretching over the entire range of travel of the coil is the most effective scheme for generating a damping force [17]. But the moving coil design has the disadvantage that flexible electrical leads or brushes must be attached to the coil if it is to be connected to a variable resistance. This can be overcome by introducing a device mode called “moving magnet’ as suggested in [16, 17]. The moving magnet system with high-energy magnets can be almost equally effective as the moving coil. The fixed coil system has several advantages, namely:

(a) Flexible wires or sliding brushes are not required;

(b) Heat energy dissipated in the coil can be easily conducted to the core and the case of the device.

For moving magnet system, the coil is fixed to a soft iron core and a ring shape magnet is the moving element. Figure 2.4 shows the section view for the configuration, where the various dimensions are:

A:	wire cross-section area;	R_{ave}:	mean radius of coil
L_m:	length of permanent magnet;	L_c:	length of coil
T_c:	thickness of coil		

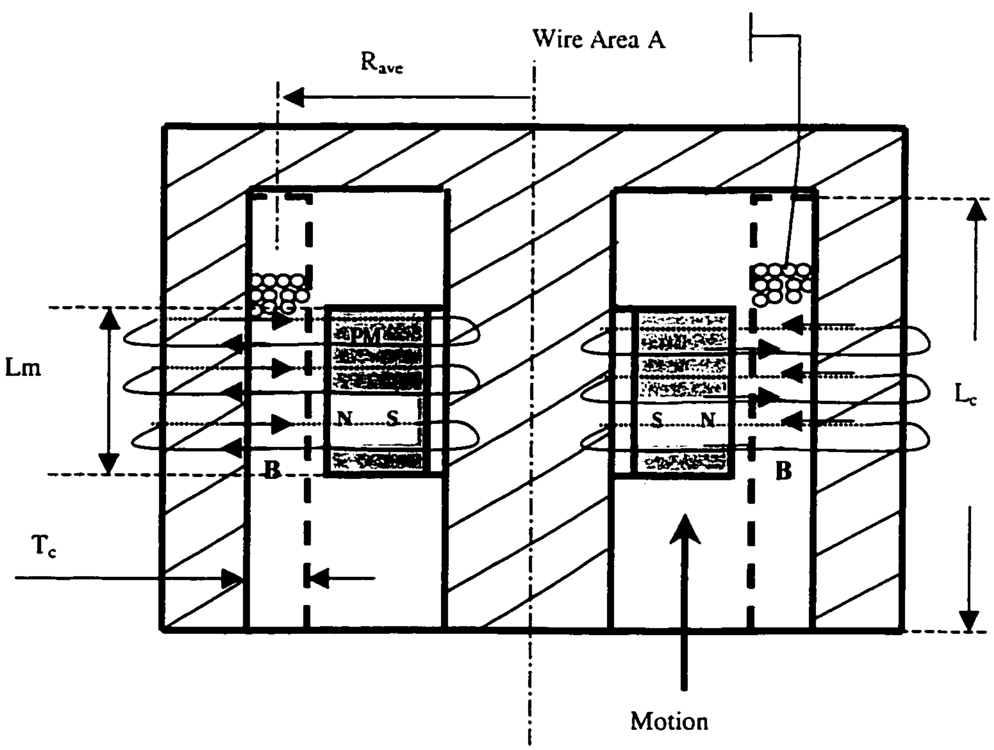

Figure 2.4 Section of moving magnet system

The force analysis of the regenerative device presented in this section is adapted from references [6].

2.4.2 Electromagnetic Force Analysis

The fixed coil in Figure 2.4 is obtained by winding wire of cross-section (A) for N number of turns. The number of turns (N) can be established from the relationship [17].

$$N = \frac{L_c T_c \Psi}{A} \tag{2.4.1}$$

Where, L_c and T_c are defined in Figure 2.4, and Ψ is the fraction of wire area over coil nominal cross sectional area. The total length of the wire required for the coil can then be expressed as:

$$L = 2\pi R_{ave} N \tag{2.4.2}$$

Therefore the conductor volume and mass of coil can be calculated from:

$$V = LA \tag{2.4.3}$$

$$M_{coil} = V\rho_c \tag{2.4.4}$$

Where ρ_c is the mass density of the wire material.

When a current, i , passes through the coil, the motion of the magnet is resisted due to the magnetic field B. The resistive force can be related to the current (i) using the expression [17]:

$$F_{ed} = BL\frac{L_m}{L_c}i \tag{2.4.5}$$

Alternatively, if the magnet is given a linear motion, an induced voltage is generated across the coil, and is a function of the velocity of the magnet motion:

$$E_{in} = BL\frac{L_m}{L_c}\dot{Z} \qquad (2.4.6)$$

The induced voltage can further be expressed as:

$$E_{in} = (R_c + R_e)i \qquad (2.4.7)$$

Where R_c is the resistance of the coil and R_e is the external resistance connected to the coil. The value for the internal resistance of the coil can be established from:

$$R_c = \frac{L\mu}{A} = \frac{V\mu}{A^2} \qquad (2.4.8)$$

Where μ is the resistivity of the coil wire material.

Substitution of (2.4.6) and (2.4.7) into Equation (2.4.5) leads to an expression for force as a function of velocity and is:

$$F_{ed} = \frac{BV}{A}\frac{L_m}{L_c}(\frac{BV}{A}\frac{L_m}{L_c}\dot{Z})/(R_c + R_e) \qquad (2.4.9)$$

When the coil is short circuited, the external resistance in the above equation is zero. The regenerative force thus reaches the peak value. The maximum force can therefore be expressed by equating $R_e = 0$ in (2.4.9) and substituting for R_c from (2.4.3), (2.4.4) and (2.4.8) to yield:

$$F_{ed} = B^2(\frac{L_m}{L_c})^2\frac{M_{coil}}{\rho * \mu}\dot{Z} = \Omega\dot{Z} \qquad (2.4.10)$$

$$\Omega = B^2(\frac{L_m}{L_c})^2\frac{M_{coil}}{\rho * \mu} \qquad (2.4.11)$$

Where Ω is a constant and can be defined as Equivalent Damping Coefficient, which reflects the damping characteristic of the linear motor.

The motion of a linear motor when applied to an oscillation can be represented as a single degree of freedom system shown in Figure 2.5. The equation of motion for the external mass with the magnet of the linear motor is:

$$(M_{mag} + M_{ext})\ddot{X} + \Omega\dot{X} + KX = 0 \qquad (2.4.12)$$

Where M_{ext} is the externally connected mass, the mass of the magnet is designated by M_{mag}, and K is the stiffness of a spring. For the undamped system the natural frequency ω_n is the function of mass and stiffness only:

$$\omega_n^2 = \frac{K}{M_{mag} + M_{ext}} \qquad (2.4.13)$$

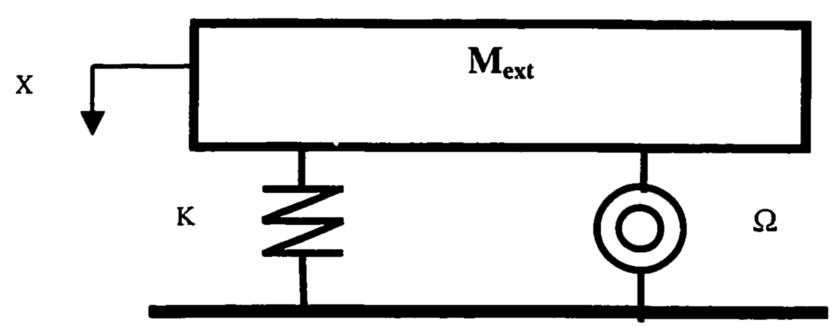

Figure 2.5 One DOF System with a linear motor

And the damping factor ζ giving the ratio of system damping to that of critical damping of the system is:

$$2\zeta\omega_n = \frac{\Omega}{M_{mag} + M_{ext}} \tag{2.4.14}$$

Substituting for Ω from equation (2.4.11) representing short-circuited maximum damping, the damping ratio is:

$$\zeta = \frac{\omega_n}{2}\frac{B^2 M_{coil}}{K\rho\mu}\left(\frac{L_m}{L_c}\right)^2 \tag{2.4.15}$$

If an external resistance is used, the damping coefficient will be reduced by a factor $\frac{1}{1+R_e/R_c}$ as follows:

$$\zeta = \frac{\omega_n}{2}\frac{B^2 M_{coil}}{K\rho\mu}\left(\frac{L_m}{L_c}\right)^2\frac{1}{(1+R_e/R_c)} \tag{2.4.16}$$

2.4.3 Electric circuit Design

In the previous section, the regenerative damper design was described and analyzed for its force generating capabilities. Expressions for damping force obtained indicate force characteristics as a function of velocity and that they behave like a linear damper with fixed coefficient. In order to obtain the characteristics of the regenerative device as a nonlinear damper and an energy regenerative damping device, a related electric circuit must be introduced. Figure 2.6 shows a series electric circuit for regenerative damper. The circuit consists of a linear DC generator (its induced voltage

E_{in} and inner resistance R_c), an externally adjustable resistance R_e, switch (A, B, C), a full wave rectifier D, and a battery E_b and its inner charging resistance r.

Equation (2.4.9) reveals that the equivalent regenerative damper force is determined by the structural characteristics of the linear generator, or determined by A (wire cross sectional area), L_m (length of permanent magnet), L_c (length of coil), μ (resistivity of copper), ρ (density of material), B (flux density of magnetic field), as well as internal and external resistance's. In a design of a regenerative damper, all the parameters will be fixed except the external resistance. To realize the nonlinear damping, identified in previous section (Table2.2, Figure2.2), an external adjustable resistance R_e is then the most convenient means to achieve the required characteristics.

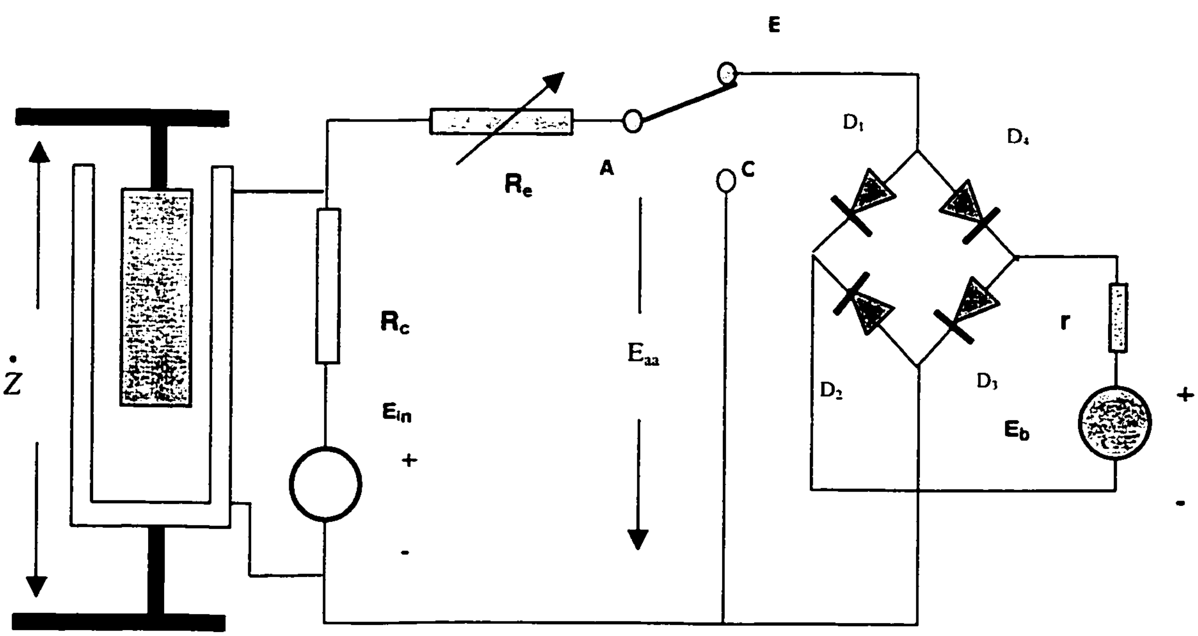

Figure 2.6 Electric Circuit

In order to adapt the regenerative damping device as regenerator of energy, it is necessary to focus on its charging capability and means of storage. The induced voltage from oscillating motion of the damper magnet will lead to an oscillating wave form for the voltage. Such wave form under sinusoidal motion is shown in Figure 2.7 (a). As part of battery charger, the full-wave rectifier is therefore necessary, where the pulsating current supplied by the DC generator will be modified prior to charging the battery. The bridge full-wave rectifier shown in Figure 2.6 requires four diodes, where D_1 and D_3 conduct during one-half cycle while D_2 and D_4 conduct during the other half cycle. The bridge-circuit inverts the negative voltage across its two diodes which lead to an identical output voltage waveform of the bridge circuit as shown in Figure 2.7(b).

Considering the situation when the vehicle moves at low speed and on smooth road surface, the induced voltage of the DC generator is expected to be relatively low. As shown in Figure 2.6, when the voltage E_{aa} is less than the threshold of battery charging circuit, which consists of the thresholds of diodes and the voltage drop of battery, an inverse current will occur in the closed circuit. And the inverse current will lead to an undesired regenerative damper force generated in the regenerative damper. In order to eliminate such a situation, and to supply a continuous, desired damping force to the suspension system, the switch ABC is designed and added to the circuit. When E_{aa} is greater than the threshold of the battery charging circuit, switch A-B is connected; otherwise switch A-C is connected, that leads to a short circuit. The electrical system in an automobile is a 12-volt system, with a 12-volt lead acid battery. The charging system in most cars will generally produce a voltage between 13.5 and 18 volts while the engine is running. It has to generate voltage higher than the battery's rated voltage to overcome the internal resistance of the battery. Also, taking account of the threshold of diodes, which is around 0.7 volts each, the total threshold of the battery circuit ranges from 14.9 voltage to 19.4 volts. In this study, the battery charging voltage is chosen as 18 volts.

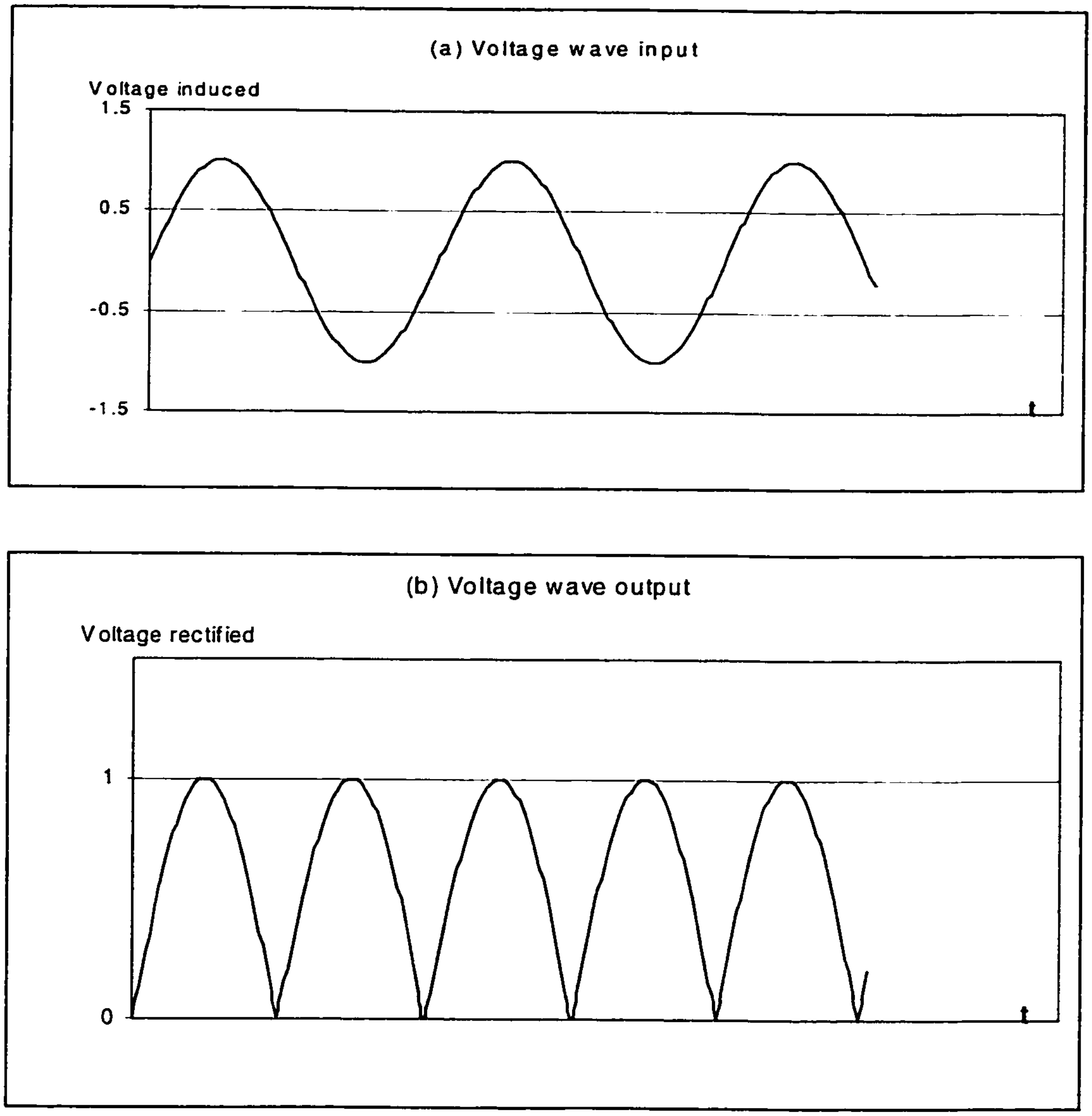

Figure 2.7 Induced Voltage waveform under Oscillating motion and Rectified waveform

2.4.4 Nonlinear Regenerative damper force

As discussed previously and presented in Figure 2.2, vehicle suspension dampers are designed to produce nonlinear asymmetric characteristics. The objective of this phase of the study is to realize such characteristics for the proposed regenerative damping device. It was pointed out under the formulation of regenerative damper, that external resistance R_e is the most convenient and practical variable for such modification. It should be noted however, that as shown in Figure 2.6, R_e also has direct influence on the charging system of the energy regeneration.

As was derived in equation (2.4.5), the regenerative damper force for equal length of magnet and coil is:

$$F_d = BLi \tag{2.4.17}$$

Substituting 2.4.7 into equation 2.4.17, the regenerative damper force can be expressed in terms of induced voltage to yield:

$$F_d = BL\frac{E_{in}}{R_c + R_e} \tag{2.4.18}$$

Then taking the threshold of battery charging circuit into account, the regenerative damper force can be rewritten as:

$$\begin{cases} F_d = BL\dfrac{(E_{in} - E_{ch})}{R_{total}} & when\ E_{in} > E_{ch} \\ F_d = BL\dfrac{E_{in}}{R_c + R_e} & when\ E_{in} \le E_{ch} \end{cases} \tag{2.4.19}$$

Where induced voltage E_{in} is a function of $\dot{Z}$, and E_{ch} is the threshold of the battery charging circuit, which includes the charging voltage of the battery E_b and the threshold for pair of diodes (E_d). The total resistance in the circuit, R_{total}, is equal to the sum of inner resistance of the linear generator R_c, the variable external resistance R_e and the resistance r_b of battery under charging state. It is estimated that reasonable value for the parameters, are $E_b = 18$ Volts, $E_d = 1.4$ Volts and $r_b = 2$ Ohms.

In order to achieve the nonlinear control of the regenerative damper force according to the performance characteristics of a conventional damper shown in Figure 2.2, the control of variable resistance R_e should be determined. Furthermore R_e selected to achieve a damping force must be determined based on the criteria whether $E_{in} > E_{ch}$ or $E_{in} \le E_{ch}$. In order to express damping force of the regenerative damper as a function of velocity, Equation (2.4.6) and (2.4.19) can be combined in the following manner:

For non-generating case $E_{in} \le E_{ch}$:

$$F_d = \frac{(BL)^2 \left({L_m}/{L_c} \right)^2}{R_c + R_e} \dot{Z} \tag{2.4.20}$$

In the above equation all regenerative damper parameters are fixed except R_e, which is to be varied to obtain the variable damping. An expression for R_e to achieve the characteristics shown in Figure (2.2) can be easily obtained by equating equation (2.4.20) to damping force expressions of conventional asymmetric damper presented in equation (2.2.1) to yield:

$$
\begin{cases}
R_e = \dfrac{(BL)^2 \left({L_m}/{L_c} \right)^2}{C_1} - R_c & 0 < \dot{Z} < a_1 \\
R_e = \dfrac{(BL)^2 \left({L_m}/{L_c} \right)^2}{C_3} - R_c & a_2 < \dot{Z} < 0 \\
R_e = \dfrac{(BL)^2 \left({L_m}/{L_c} \right)^2}{C_2 + (C_1 - C_2)\, {a_1}/{\dot{Z}}} - R_c & \dot{Z} > a_1 \\
R_e = \dfrac{(BL)^2 \left({L_m}/{L_c} \right)^2}{C_4 + [C_3 - C_4 sign(\dot{Z})]\, {a_2}/{\dot{Z}}} - R_c & \dot{Z} < a_2
\end{cases}
\tag{2.4.21}
$$

For generating case $E_{in} > E_{ch}$:

Similar to above, for generating case, equations (2.4.19) and (2.4.6) can be combined to yield:

$$
F_d = \frac{(BL\, {L_m}/{L_c})[(BL\, {L_m}/{L_c}) - E_{ch}]}{R_c + R_e + r_b} \dot{Z} \tag{2.4.22}
$$

Equating the above to the damper force of the conventional asymmetric characteristics [Equation (2.2.1)], the variable parameter R_e can be established as:

$$
\begin{cases}
R_e = \dfrac{(BLL_m / L_c)[(BLL_m / L_c) - E_{ch}]}{C_1} - (R_c + r_b) & 0 \le \dot{Z} \le a_1 \\
R_e = \dfrac{(BLL_m / L_c)[(BLL_m / L_c) - E_{ch}]}{C_3} - (R_c + r_b) & a_2 \le \dot{Z} < 0 \\
R_e = \dfrac{(BLL_m / L_c)[(BLL_m / L_c) - E_{ch}]}{C_2 + (C_1 - C_2) a_1 / \dot{Z}} - (R_c + r_b) & \dot{Z} > a_1 \\
R_e = \dfrac{(BLL_m / L_c)[(BLL_m / L_c) - E_{ch}]}{C_4 + [C_3 - C_4 sign(\dot{Z})] a_2 / \dot{Z}} - (R_c + r_b) & \dot{Z} < a_2
\end{cases}
\tag{2.4.23}
$$

From the above equation (2.4.21) and (2.4.23), it is evident that the regenerative damper parameter is also a function of R_e. Comparison of equations (2.4.21) and (2.4.23) further shows that the R_e required during non-regenerating phase will be significantly larger than the regenerating phase and needs to be modulated based on $\dot{Z}$ as well as E_{in}.

Assuming that an ideal controller can modulate the variable resistance based on E_{in} and $\dot{Z}$, a continuously varying force for the damper can be established as a function of $\dot{Z}$ only. For a set of fixed parameters representing the regenerative dampers, the asymmetric characteristics can be represented by:

$$\begin{cases} F_d = 78.41\dfrac{1-e^{-12\dot{Z}}}{1+e^{-12\dot{Z}}}(1+2.15\dot{Z}) & \dot{Z} \geq 0 \\ F_d = 594.3\dfrac{1-e^{-11\dot{Z}}}{1+e^{-11\dot{Z}}}(1+0.7\left|\dot{Z}\right|) & \dot{Z} < 0 \end{cases} \qquad (2.4.24)$$

Similarly a symmetric damper characteristic can be established as:

$$F_d = \frac{(78.41+594.2)}{2}\frac{1-e^{-10\dot{Z}}}{1+e^{-10\dot{Z}}}(1+0.34\left|\dot{Z}\right|) \qquad (2.4.25)$$

The above equations are valid regardless of E_{in} since R_e is appropriately modulated using (2.4.21) or (2.4.23) to provide the required damping.

The force-velocity characteristics of the proposed damper expressed by equation (2.4.24) are presented in Figure 2.8. As discussed in Chapter 1, such characteristics more closely represent a realistic asymmetric damper than those presented idealized representation using piece-wise linear relationships. The symmetric characteristic based on the average of asymmetric system presented in equation (2.4.25) is plotted in Figure 2.9.

These two damper characteristics and representations are used to evaluate the performance of regenerative damper along with its properties in term of R_e required and E_{in} produced under different conditions.

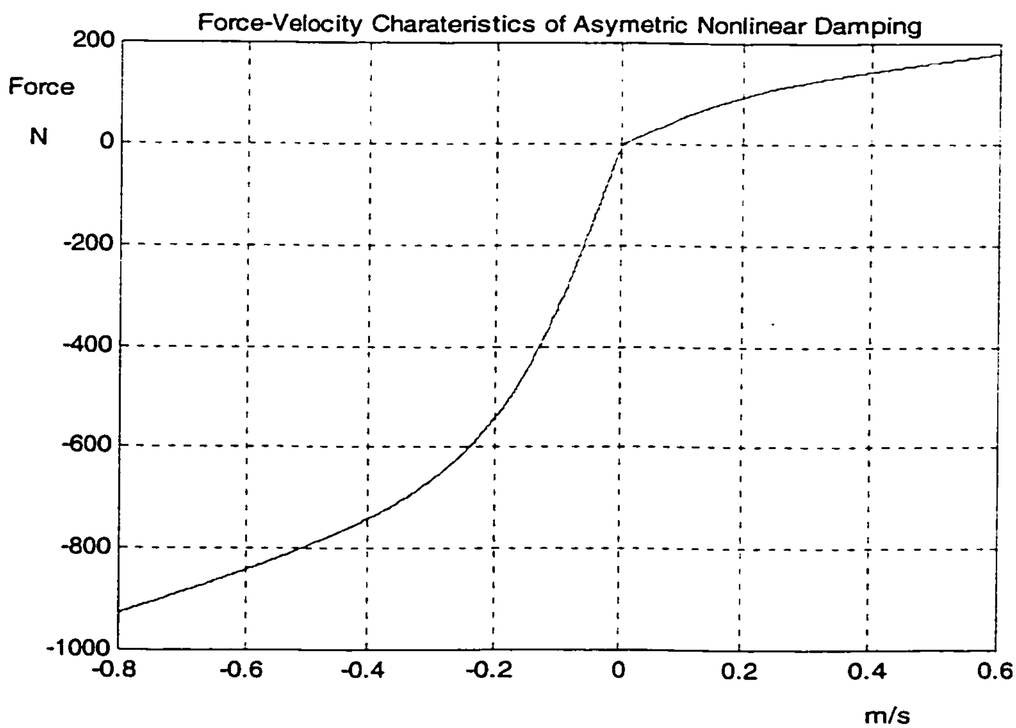

Figure 2.8 Asymmetric Nonlinear Damper Characteristics produced by a regenerative damper

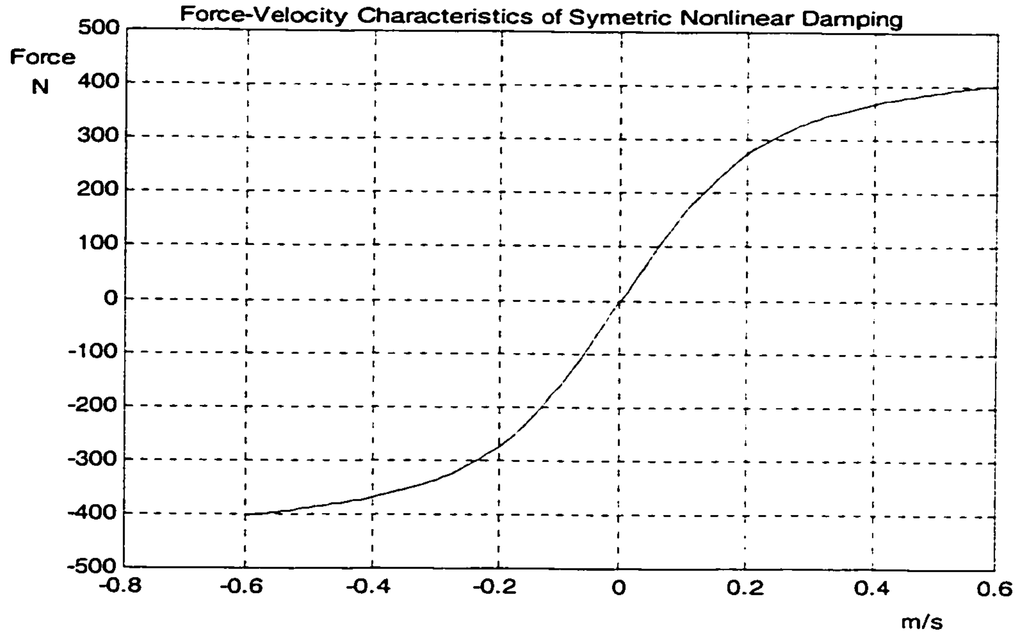

Figure 2.9 Symmetric nonlinear Damper Characteristics produced by a regenerative damper

2.4.5 Electrical Delay Time

For the proposed design presented in Figure 2.4, a current in the coil will tend to induce a magnetic field. Generally, the flux lines will tend to encircle the coil, lying mainly in the central core, the external shell and at the two ends. The result is that the coil will possess self-inductance L_e as well as resistance R. An electrical time constant, delay time, can thus be defined [17].

$$T_d = \frac{L_e}{R} \tag{2.4.26}$$

If only the resistance of coil is taken into account, it can be rewritten as:

$$T_d = \frac{L_e}{R_c} \tag{2.4.27}$$

In order to make the regenerative damping device design suitable for the vehicle suspension system, the electrical delay time should be shorter compared to the period of oscillations to be damped so that the current and damping force are essentially proportional to the relative velocity.

The electrical delay time should therefore be expressed in terms of the coil parameters, as the inductance L_e can be expressed in terms of the number of turns, N and the permeance of the path of the coil-induced flux, P [17]:

$$L_e = N^2 P \qquad (2.4.28)$$

Considering the state of short circuit, or $R = R_c$, using equation (2.4.1), (2.4.2), (2.4.3), (2.4.8) to substitute in (2.4.27):

$$T_d = \frac{N^2 P}{L\mu / A} = \frac{N^2 P}{2\pi R_{ave} N \mu / A} = \frac{L_c T_c \Psi P}{R_{ave} 2\pi\mu} \qquad (2.4.29)$$

The permeance is hard to estimate, but for the order of magnitude calculations, it can be defined in terms of air gap area A_g, length L_g, and the permeability of free space μ_o. Then the permeance can be written as:

$$P = \frac{A_g \mu_o}{L_g} \qquad (2.4.30)$$

From Figure 2.4, A_g is assumed to be $A_g \cong \pi R_{ave}^2$. Substituting value from equation (2.4.30) into equation (2.4.29), the electrical delay time can thus be rewritten as:

$$T_d = \frac{L_c T_c \Psi R_{ave} \mu_o}{2\mu L_g} \qquad (2.4.31)$$

Assuming $L_c=L_g$, $\Psi=0.7$, $T_c=0.02$ m, $R_{ave}=0.05$ m, $\mu_o=4\pi*10^{-7}$H/m, and $\mu=1.72*10^{-8}$ Ωm, the value can be found to be: T_d=25ms. Therefore, by probability, response delay due to the self-inductance effect will not be a limitation at the typical vibration frequencies encountered in vehicle suspensions. Of course, if the external resistance is used in addition to the coil internal resistance, the time constant is even smaller as may be seen from equations (2.4.26) and (2.4.27).

2.5 Road Profile

A vehicle ride performance analysis through a computer simulations or analytical models necessitates accurate characterization of model parameters and excitation. The dynamic characteristics of the vehicle, the dynamic wheel loads and the ride quality are strongly related to the road profile. The roads are well known to exhibit randomly distributed roughness. In the studies of roads, the measured road profiles are invariably expressed in terms of spatial power spectral density (PSD) of the roughness, with the assumption that random roughness characteristics follow the stationary Gaussian distribution [1, 40]. In this dissertation research, all preliminary results are obtained for sinusoidal input of different magnitudes. Responses to such input are easily interpreted in relation to the input. The feasibility of the proposal regenerative damping concept is however also examined under realistic random road excitation. A road excitation representing city road presented in Figure 2.10, as PSD is considered in this investigation. The off road spectra are estimated by multiplying the urban road data by a factor of 1.5.

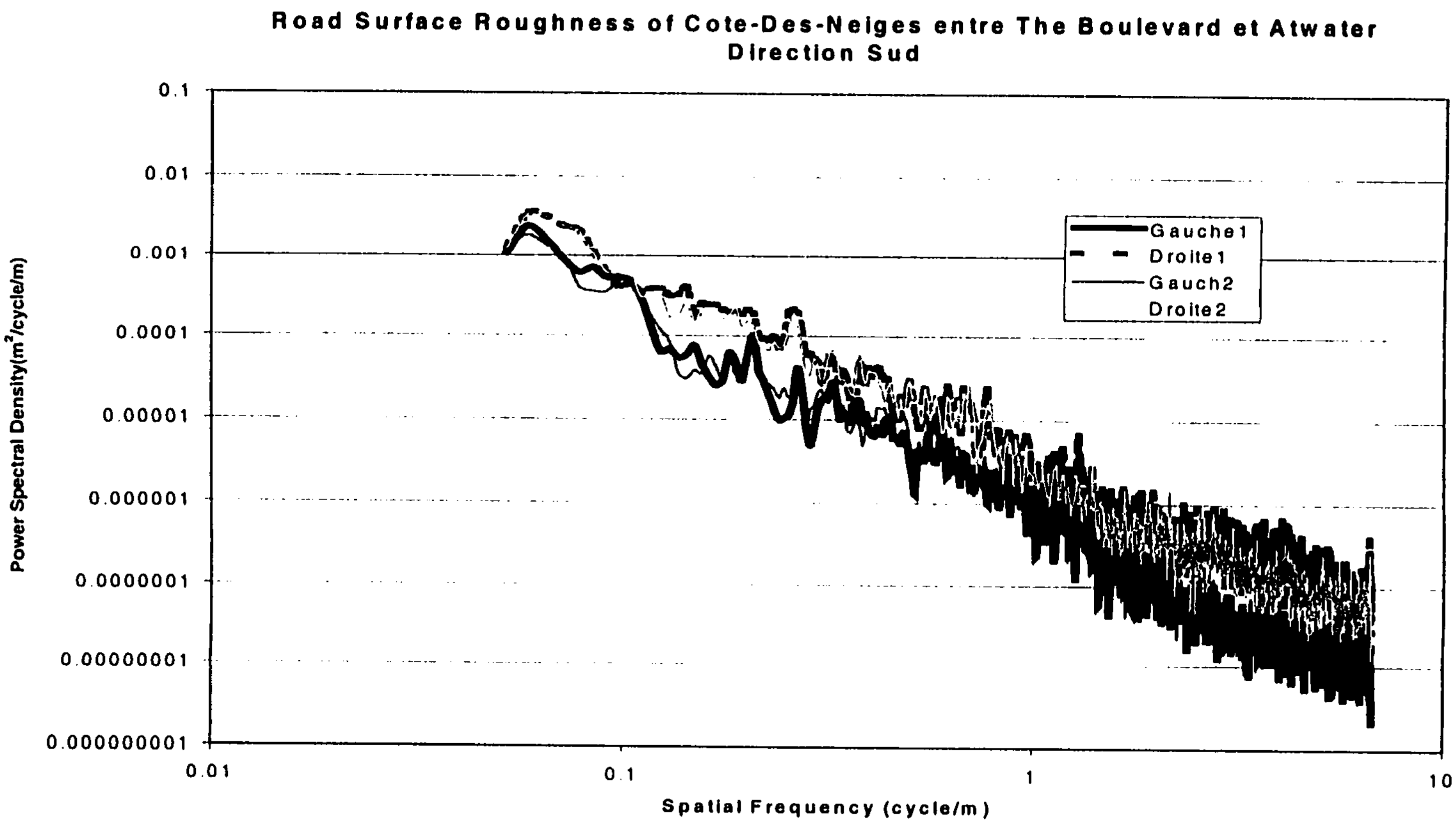

Figure 2.10 Road Roughness Spectra for an Urban Road [41]

2.6 Vehicle Parameters

The weight and dimension parameters of the quarter vehicle model are identified from the data adopted by [36]. The spring stiffness, nonlinear damping coefficient and tire model are also identified from [36]. The simulation parameters for the vehicle model are thus complied and summarized in table 2.3.

Table 2.3 Vehicle Parameters

Description of the parameter	Parameter values
Sprung mass (M_s) including coil & frame mass 10Kgs	240 Kg
Unsprung mass (M_u) including magnet mass 5Kgs	40 Kg
Axle suspension stiffness (K_s)	16 KN/m
Axle suspension damping coefficient / Regenerative damping coefficient (C_s) (see Figure 2.2)	C_1=514.5 Ns/m C_2=176.75 Ns/m C_3=2747.5 Ns/m C_4=462.0 Ns/m A_1=0.1524 m/s A_2=-0.2163 m/s
Tire suspension stiffness (K_t)	120 KN/m
Tire suspension damping coefficient (C_t)	200 Ns/m

The simulation parameters for the regenerative damping system, including DC linear generator; energy recharging circuit; and energy storing device, are summarized in table 2.4.

Table 2.4 Parameters of Regenerative Damping System	
DC Linear Generator	
Wire cross area: A	1 square mm
Length of wire: L	561.8m
Length of permanent magnet: L_m	50 mm
Coil length: L_c	100 mm
Thickness of wire wound on the coil: T_c	20 mm
Mean radius of coil: R_{ave}	50 mm
Magnetic field flux density: B	1.37 T
Mass of copper wire: M_{coil}	5 Kgs
Resistivity of copper: μ	1.72×10^{-8} Ω/m
inner resistance of generator: R_c	11.24 Ω
Charging & Storing system	
Energy storing device: automobile battery	12 voltage
Battery charging voltage: E_b	18 voltage
Battery charging resistance: r_b	2 Ω
Threshold of diodes: V_t	0.7 voltage

2.7 Summary

A simplified vehicle model incorporating a detailed suspension system is developed for study of ride performance. A regenerative damping device is proposed and modeled to develop its characteristics through equations and to express force and regenerative energy in terms of system parameters. A circuit is proposed to provide hybrid applications in providing damping and generating electric energy. A set of parameters is identified for the proposed system to mimic the characteristics of a conventional nonlinear asymmetric damper. Some random road inputs are presented to characterize city roads and off roads. The equations of motion and component forces can readily be used to carryout simulations for evaluation of ride performance. The models developed here are utilized in the next chapter for detailed analysis of performance in term of ride and dynamic tire load under sinusoidal and random road excitations. Analysis of energy regeneration is presented in Chapter 4.

Chapter 3

Simulation of Ride Performance and Dynamic Wheel Loads

3.1 Introduction

Vehicle suspension system performances are primarily judged on compared through their vibration alteration capacity over a frequency range. Since typical sprung mass and unsprung mass natural frequencies are around 1 and 10 Hz, it is considered adequate to examine the performance in the range of 0 to 15 Hz. Vehicle suspension performances are thus typically evaluated in terms of sprung and unsprung mass acceleration transmissibility over the frequency range. Other important performance measure may include suspension relative motion transmissibility and dynamic tire forces.

Variations in the dynamic tire forces of commercial vehicles, known to accelerate the road damage, are strongly related to the sprung and unsprung masses and suspension properties. The high levels of tire induced road damage caused by vehicles has prompted a growing demand for the design of road-friendly vehicles. Since the vehicle ride quality and dynamic wheel loads are both related to vehicle vibration modes, the suspension damping will influence both ride quality and road friendliness of the vehicle. Assuming negligible contributions due to pitch and roll dynamics, the quarter vehicles with the conventional, asymmetric and symmetric damping systems installed are studied in the chapter. Comparisons of ride quality of each damping system, and the dynamic load characteristics of the vehicle models with the regenerative damping systems installed are discussed. Responses are obtained for both sinusoidal road excitations and random excitations of different magnitudes. All simulations are carried out in time domain using MATLAB “Simulink”. Response to random excitations are established in terms of Power Spectral Density (PSD) and rms values of sprung mass acceleration and tire forces.

3.2 Ride Performance Characteristics

The regenerative damping system proposed and modulated in Chapter 2 is utilized to study the ride performance of quarter vehicle model. Various damper characteristics considered include the conventional asymmetric based on damping coefficients; regenerative asymmetric, continuous asymmetric and continuous symmetric damper representations based on electromagnetic properties. Adopting harmonic excitations, the simulation results for conventional and regenerative asymmetric damper parameters are first compared for verification of the developed models. The performances with various regenerative systems are then investigated under deterministic and stochastic road excitations. A performance criterion comprising acceleration transmissibility, rms acceleration response of sprung mass is obtained to study the ride quality of the regenerative damping systems.

3.2.1 Model Verification

In order to verify the vehicle model with a regenerative damping system installed, the regenerative damper model based on electromagnetic parameters expressed as equation (2.4.20 – 2.4.23) is compared with that of conventional damper. Although regenerative damper is based on different working principles, its parameters were selected to give the same force-velocity properties as that of a conventional damper shown in Figure 2.2. The vehicle models installed with these two damping systems are examined under harmonic excitation. The results are analyzed to verify the regenerative damping system can generate the same damping force and have the same influence on vehicle performance as the conventional one over a wide frequency range. Vehicle models are subjected to the identical harmonic excitations with displacement amplitude of 0.01m for frequency in the range of 0 to 15 Hz. The responses are established from steady state time history response at each selected frequency.

Figures 3.1 and 3.2 illustrate the acceleration transmissibility and the displacement transmissibility of sprung mass in frequency domain respectively, which are defined as the ratio of the peak response of steady state to the peak value of the sinusoidal wave input. As the results show, the regenerative damper can be designed to accurately replicate the characteristics of a nonlinear damper with asymmetric properties over the frequency range. These results further show that the sprung and unsprung mass natural frequencies are around 1.25 and 9 Hz respectively. Further results obtained in terms of the peak damping force and relative motion referred to as rattle space are presented in Figures 3.3 and 3.4, respectively. These results further confirm the similarity of the proposed system to that of systems with conventional dampers. As these results indicate, the damping force is maximum at wheel hop natural frequency where the relative motion across the suspension is maximum. A detailed analysis of the results, presented in Figures 3.1 to 3.4, show very small diffidence between the responses of two systems except around the wheel hop natural frequency, where a difference of 1 to 2% is observed. It should be noted that around wheel hop natural frequency, the relative motion and velocity are large causing transition between the damping stages in every cycle. Furthermore, the criteria for change in damping over a cycle as presented in equation (2.2.1) for conventional damper is different from electromagnetic system presented in equation (2.4.20 – 2.4.23).

Overall, these results indicate that the electromagnetic damper can produce the desirable damping characteristics, and that the proposed model can reliably simulate the ride response of a road vehicle with accuracy.

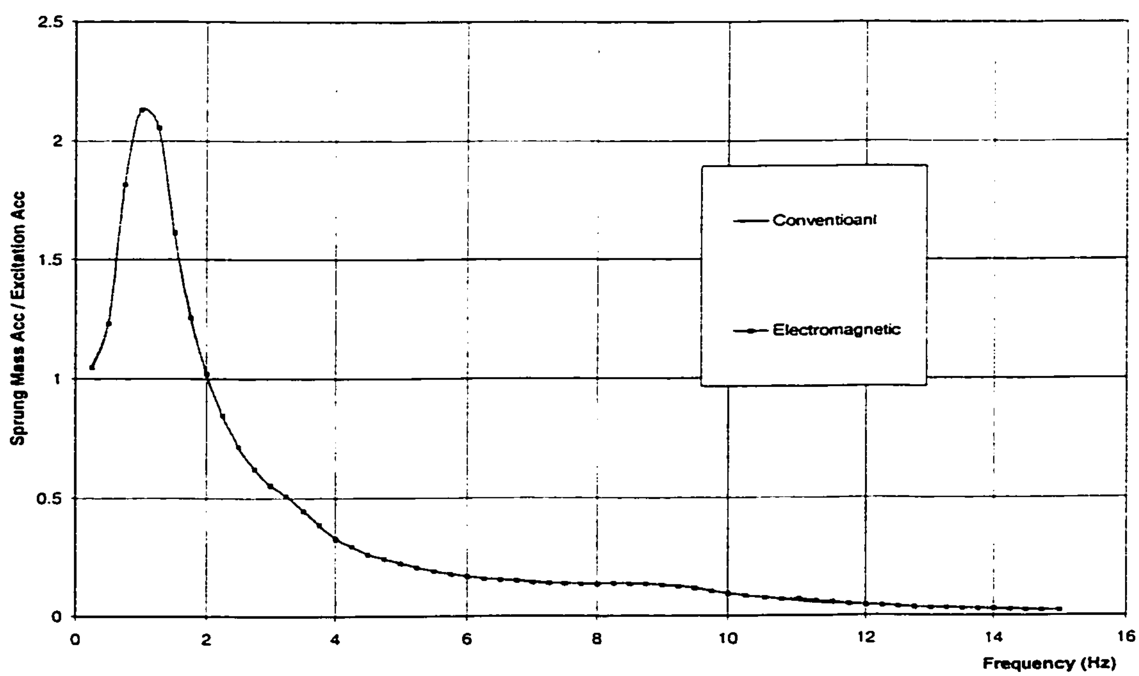

Figure 3.1 Acceleration Transmissibility Comparing Electromagnetic with Conventional Damper

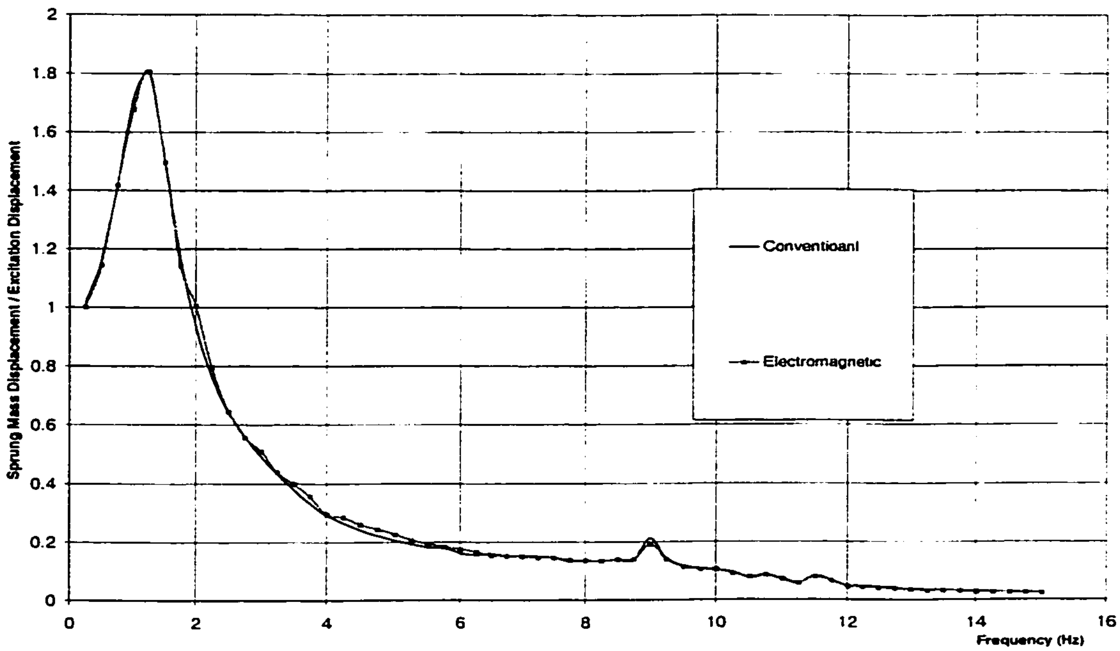

Figure 3.2 Displacement Transmissibility Comparing Electromagnetic with Conventional Damper

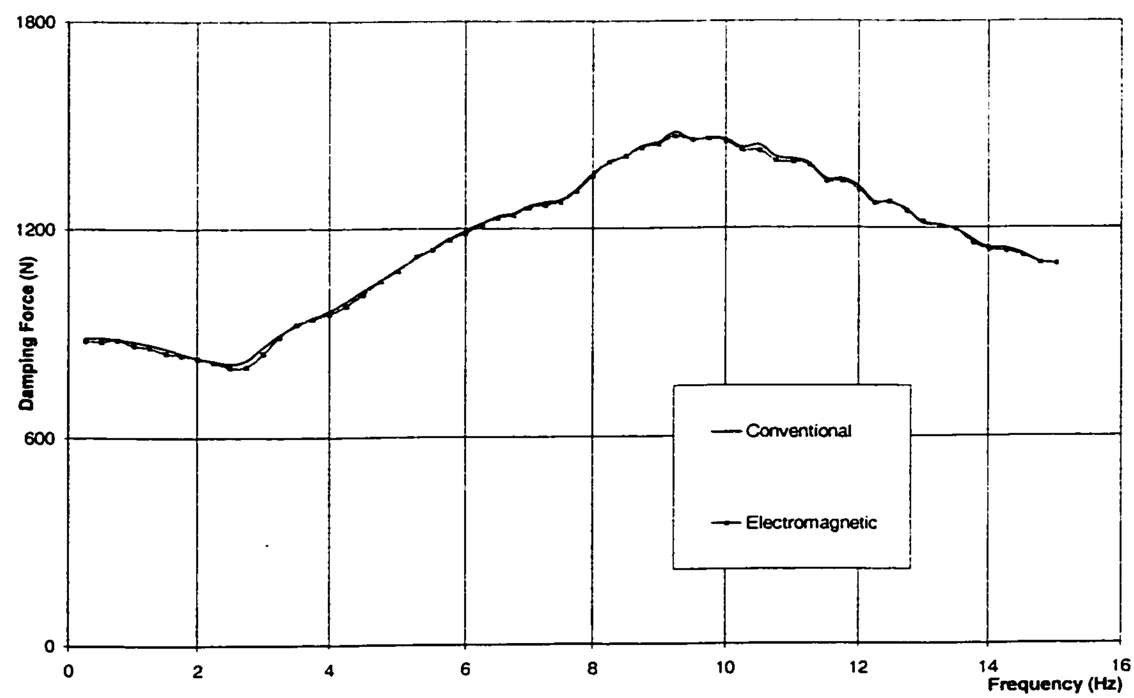

Figure 3.3 Peak Damping Force produced by Conventional and Electromagnetic Dampers

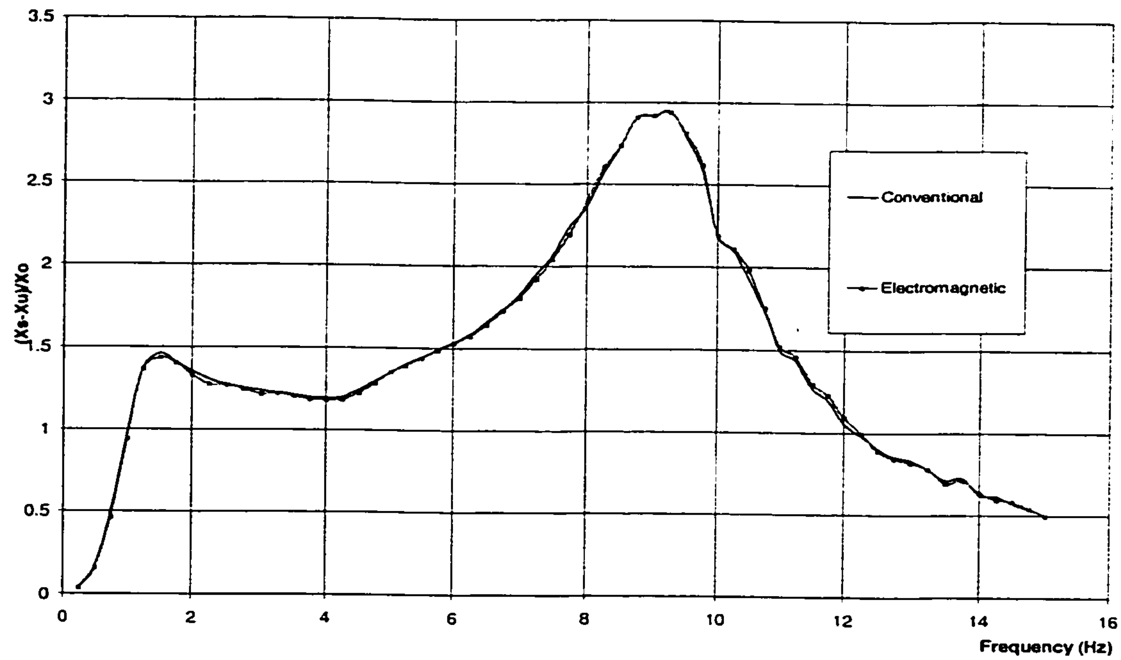

Figure 3.4 Rattle Space Response of Conventional and Electromagnetic Dampers

3.2.2 Ride Response to Harmonic Excitations

As discussed in the previous chapter, the regenerative damper property is varied by modulation of the variable resistance R_e. Furthermore, the property can be varied in discontinuous manner as that of an idealized conventional damper or in a continuous manner as illustrated by equation (2..4.22 and 2.4.23) in Chapter 2. In this section, the ride response of the vehicle is examined for different regenerative dampers. For these three configurations are selected, namely: discontinuous asymmetric; continuous asymmetric; and continuous symmetric. Based on model verification presented in section 3.2.1, it can be taken that the response due to discontinuous asymmetric dampers is the same as that produced by a conventional asymmetric damper configuration.

The vehicle model equipped with these different dampers is studied under harmonic excitations in time domain. The frequency response characteristics are expressed in terms of acceleration transmissibility, damping force and rattle space. Frequency responses are obtained for identical harmonic excitations with the displacement amplitude of 0.01m.

Figure 3.5 illustrates the acceleration transmissibility of sprung mass in the frequency range of 0 to 15 Hz for the three electromagnetic damping cases. These results reveal that discontinuous asymmetric and continuous asymmetric representation yield very similar results, where the continuous representation produces slightly lower peak transmissibility at sprung mass resonance. This will indicate that the effective damping for idealized representation is slightly lower. More interesting result is obtained for symmetric case, as shown in Figure 3.5. A symmetric damper, which is established from average values of an asymmetric case as shown in Figure 2.9, leads to noticeable improvement of acceleration transmissibility at sprung mass resonance as well as at low frequencies. The improvement of asymmetricity becomes evident when the relative motion transmissibility across the suspension is evaluated as shown in Figure 3.6. These results clearly reveal significantly large relative motion transmissibility corresponding to wheel hop natural frequency, when symmetric characteristics is used. In practice asymmetric characteristics is need to control wheel hop and to ensure adequate tire-road contact for handling. As these results indicate, this, however, come at the expense of slight deterioration of sprung mass resonance performance. The results presented in this section for harmonic excitations show the capability of the regenerative damper in producing the damping force for adequate ride qualities. The responses to harmonic excitation further present an opportunity to examine the properties within the regenerative damping device, which is presented under the following subheading.

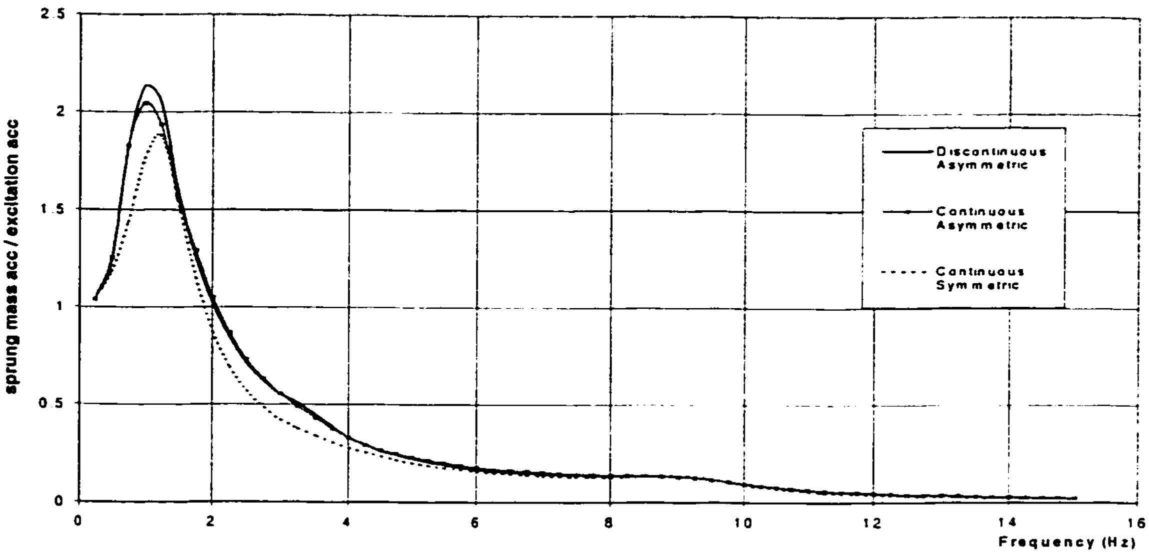

Figure 3.5 Sprung Mass Acceleration Transmissibility for various electromagnetic dampers

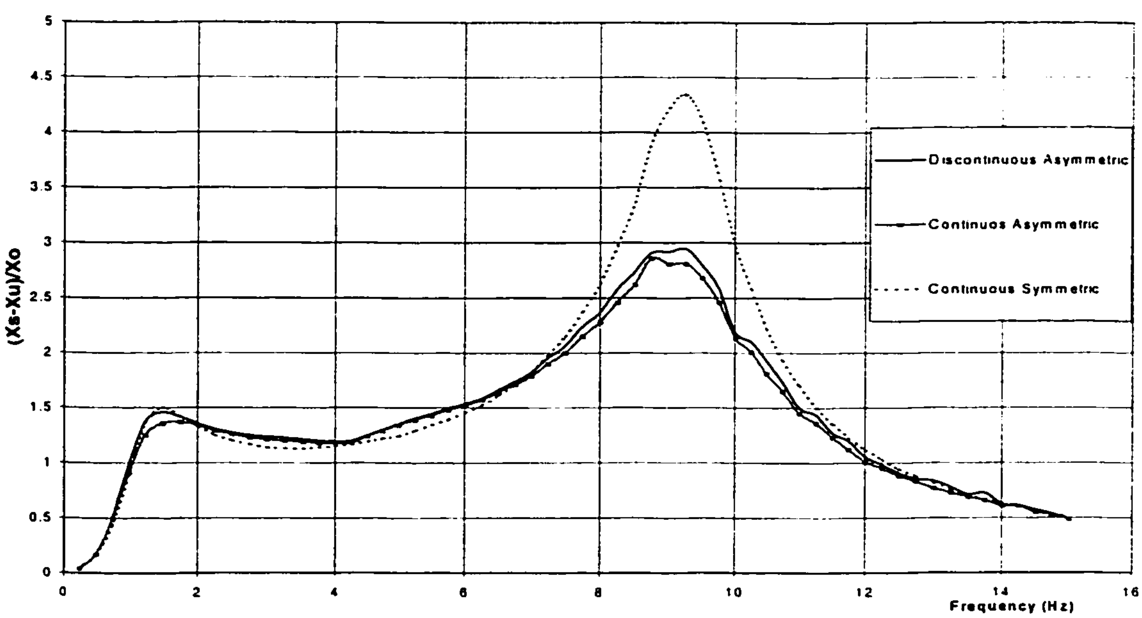

Figure 3.6 Relative motion transmissibility for various electromagnetic dampers

3.2.2.1 Analysis of Regenerative Damper Properties under Harmonic Excitation

As discussed in Chapter 2, the damping force for the regenerative damping device is produced by electromagnetic force. The damping force in the proposed design is modulated through variation in external resistance R_e. Furthermore, as it is shown in Section 2.4.4, the selection of R_e also depends on whether the system is in charging mode or not. This section studies the time history response of the system as well as regenerative damper parameters at selected amplitude of excitations and frequency.

Figures 3.7, 3.8, 3.9 and 3.10 show steady state time history for an excitation of 0.01 m at 1 and 9 Hz, respectively. Each figure presents the responses of the system in terms of relative velocity across the damper, the output of the damper voltage, and the current as well as the value of the variable resistance R_e. The results show that the inducted voltage in generators change with the relative velocities between sprung and unsprung mass, while the variable external resistance changes with inducted voltage. Coinciding with the prediction of equation 2.4.21 and 2.4.23, external resistance keeps constant in the state of non-charging and varies nonlinearly in the state of charging to determine the proper current in circuit, which finally determines the damping force. Figures 3.11 and 3.12 illustrate the peak values of external resistance and the peak values of inducted voltage in generator within the considered frequency range from 0 to 15 Hz. The results show that both external resistance and inducted voltage reach the maximum around the resonance frequency of unsprung mass.

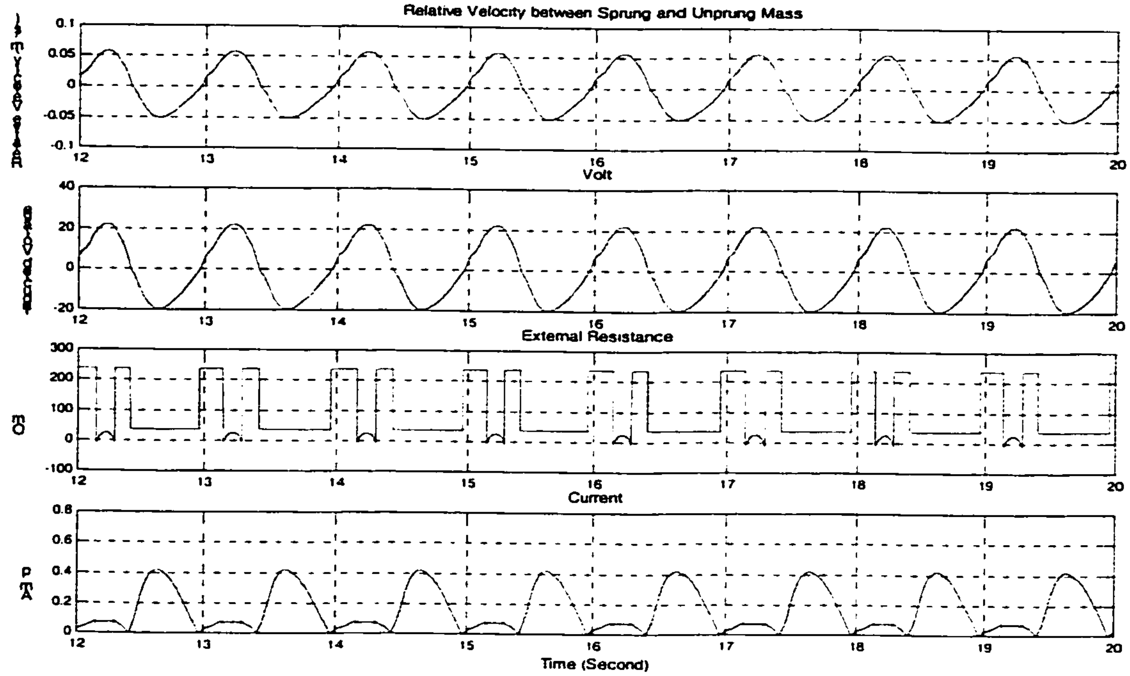

Figure 3.7 Steady State Time History for Excitation of 0.01 m at 1 Hz

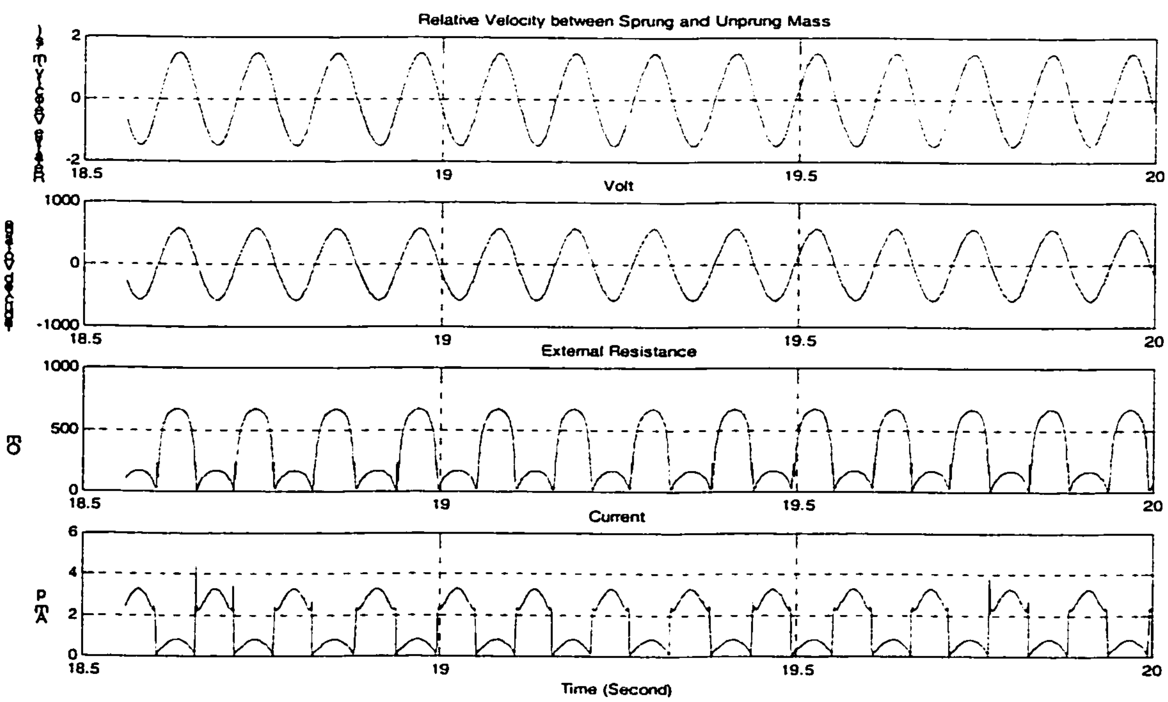

Figure 3.8 Steady State Time History for Excitation of 0.01 m at 9 Hz

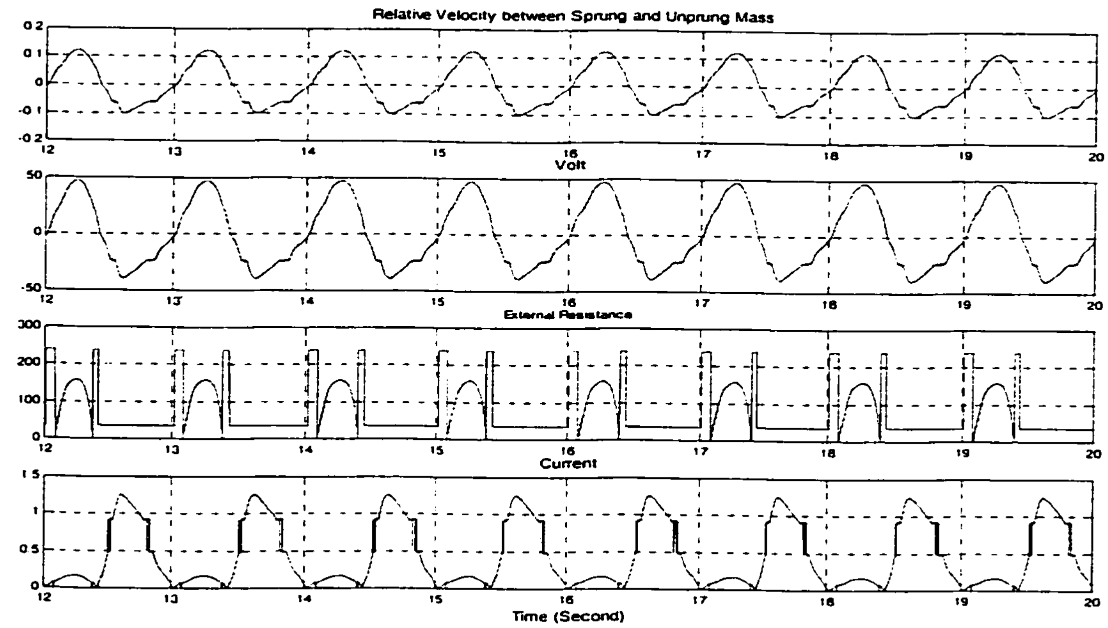

Figure 3.9 Steady State Time History for Excitation of 0.025 m at 1Hz

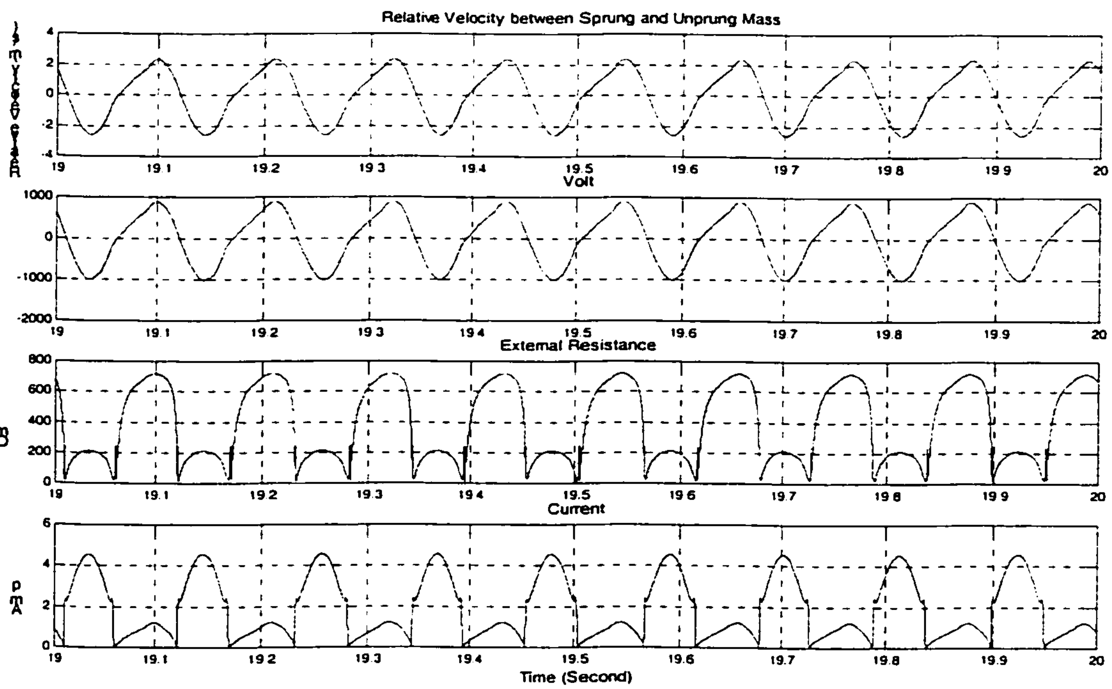

Figure 3.10 Steady State Time History for Excitation of 0.025 m at 9Hz

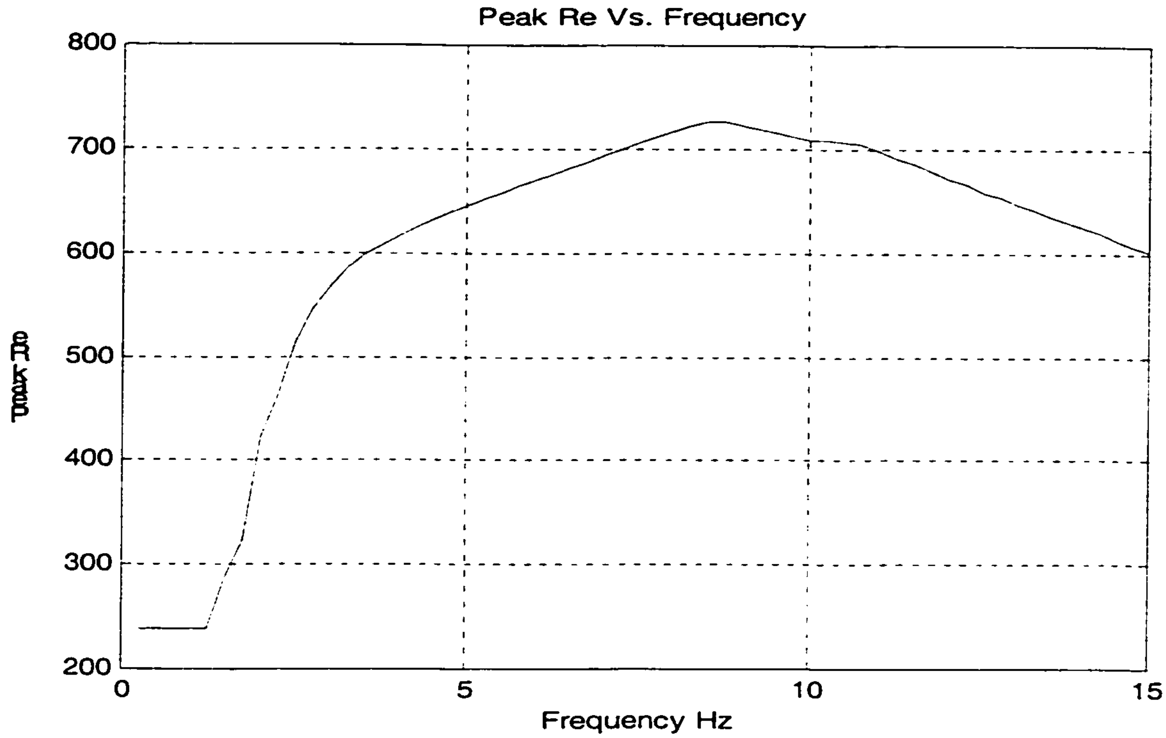

Figure 3.11 Peak value of external resistance Vs. Frequency

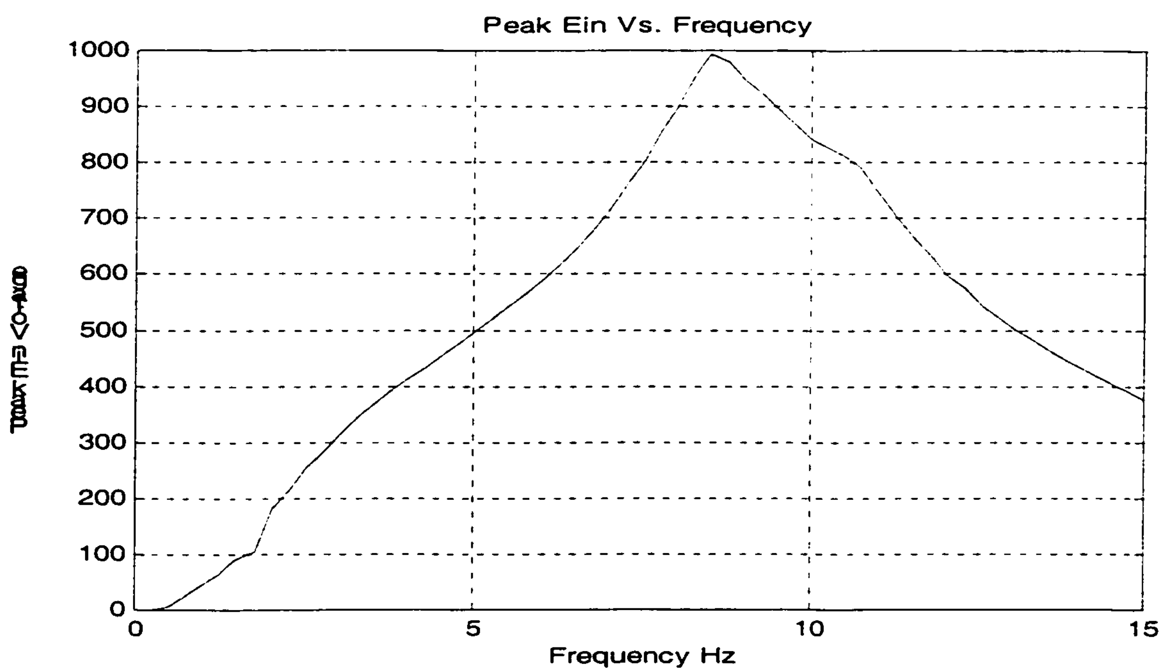

Figure 3.12 Peak value of inducted voltage in generator Vs Frequency

The results presented so far are for sinusoidal road excitation, where responses could be examined in depth in relation to the deterministic inputs: in reality, road vehicles are exposed to random excitations with varying magnitude based on roughness of the road. The following section of this chapter examines the ride performance of the vehicle with regenerative damper under random road excitation. The following chapter is devoted to the analysis of power output and efficiency of the device for realistic road excitations.

3.2.3 Ride Response to Stochastic Road Excitations

The vehicle models incorporating the electromagnetic nonlinear damping systems are analyzed under the excitations arising from three different roads, which are city road (Road Cote Des Neiges, Montreal), and off road. The roughness characteristics of the selected roads in term of PSD have been described in Section 2.5. The road profile is also available in time domain and as a function of distance. The distance data can be used to simulate road profile at selected speeds. The vehicle speed is varied from 10 to 80 km/h on both city and off road. The vehicle response characteristics are analyzed and expressed in terms of Root Mean Square (RMS) and Power Spectral Densities (PSD). The response characteristics of the vehicle model equipped with regenerative damping systems are also compared with those of the conventional vehicle to demonstrate the effectiveness of the regenerative damping devices.

City Road (Cote Des Neiges)

Figure 3.13 presents the RMS of sprung mass acceleration response as a function of vehicle moving speed, while the vehicle speed varied from 10 to 80km/h with an interval of 10 km/h. These results show comparable performance for all three regenerative damper characteristics, where the symmetric system provide slightly lower rms acceleration levels throughout the speed range considered. On a typical city road, the rms acceleration increases with speed from 0.35 m/s^2 at 10 km/h to 1.45 m/s^2 at 80 km/h. the results computed as PSD at 50 km/h is presented in Figure 3.14. Once again comparable response is produced by all configurations, where improved performance for symmetric damper is noticeable at higher frequency.

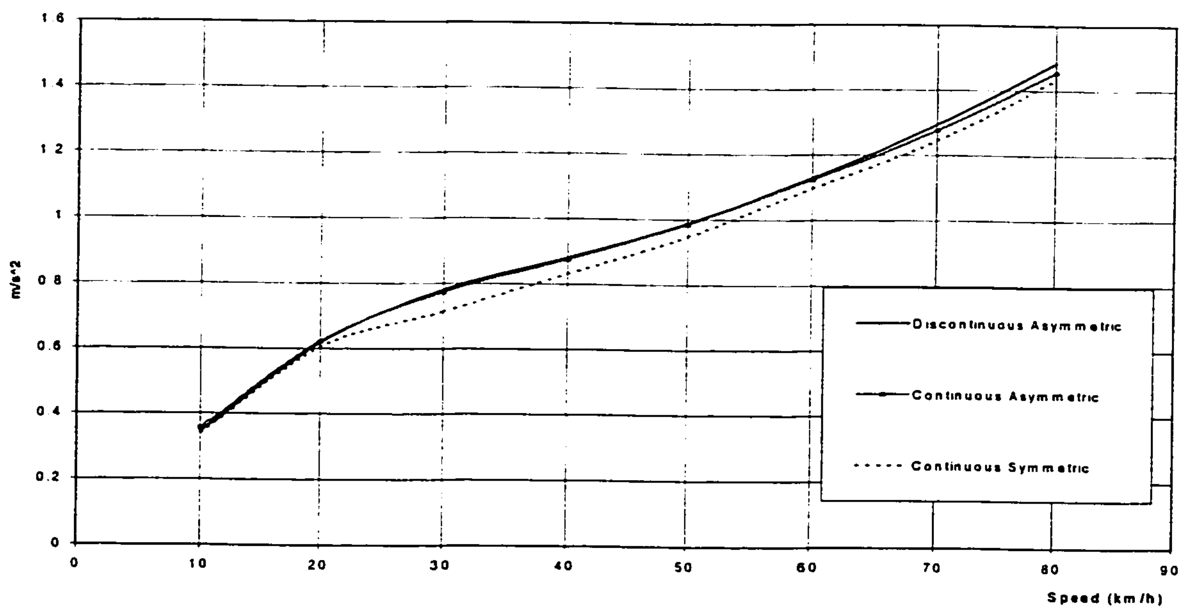

Figure 3.13 RMS of Sprung Mass Acceleration

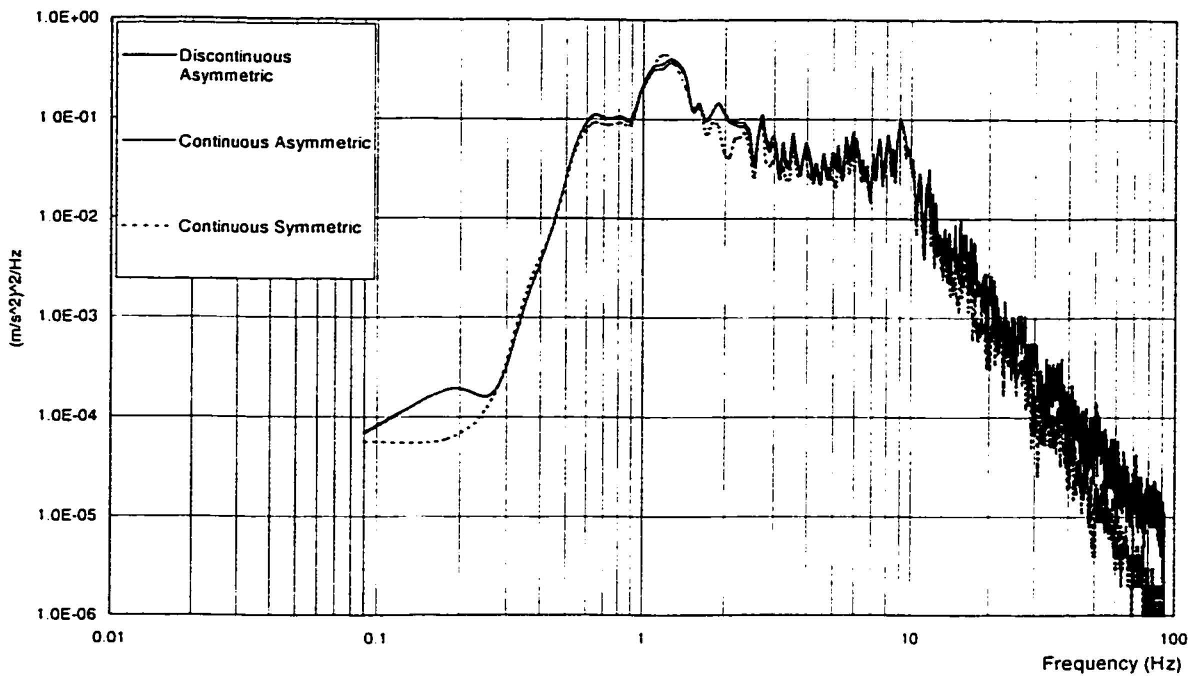

Figure 3.14 PSD of Sprung Mass Acceleration

Off Road

Figure 3.15 presents the RMS of sprung mass acceleration response as a function of vehicle moving speed, which is varied from 10 to 80km/h with an interval of 10 km/h. The performances of discontinuous asymmetric and continuous asymmetric are approximate throughout the speed range, while symmetric system provides slightly low level of rms acceleration in the speed range from 10 to 55 km/h and relatively high level at higher speeds. PSD of sprung mass acceleration at 30 km/h, which is considered as a typical off road vehicle moving speed, is computed and illustrated in Figure 3.16. The comparable responses are produced by all three configurations. The results indicate that the performances of three regenerative damping systems are similar in low frequency range below 1 Hz and the symmetric system slightly improves performance in the frequency range above 1 Hz.

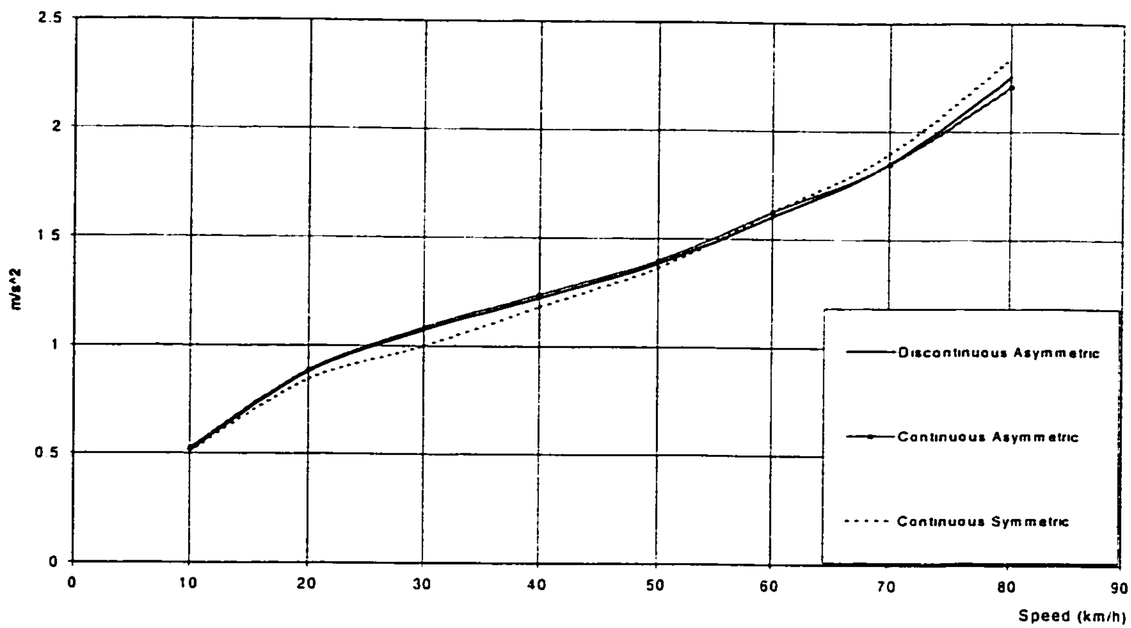

Figure 3.15 RMS of Sprung Mass Acceleration

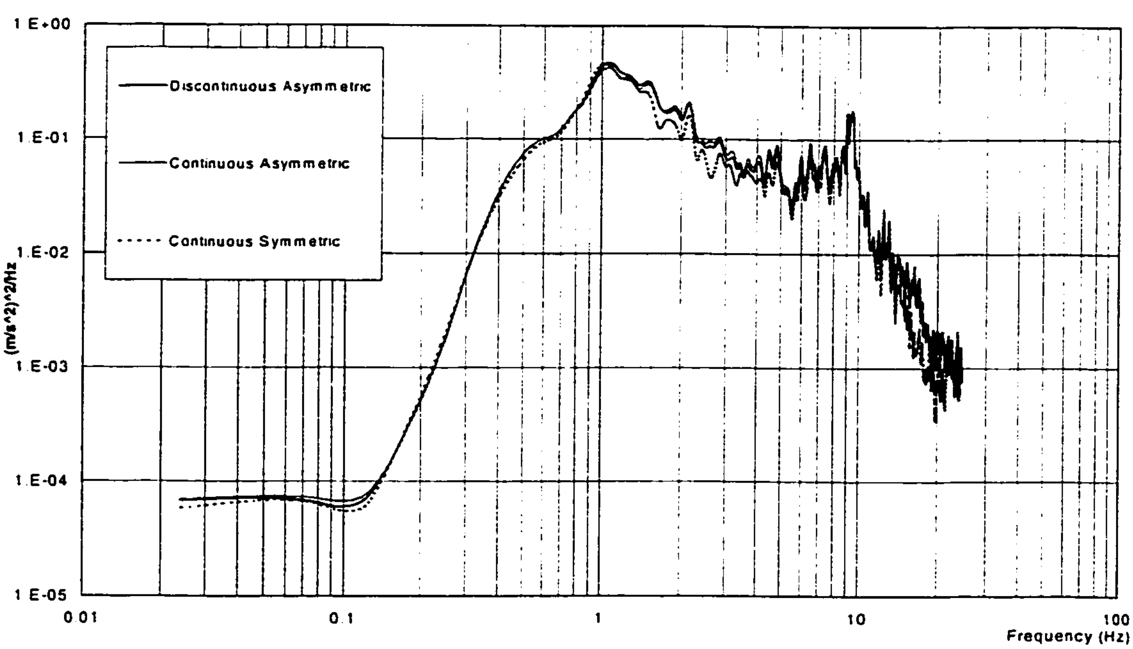

Figure 3.16 PSD of Sprung Mass Acceleration

3.3 Dynamic Tire Loads Analysis

Vehicles transmit high magnitudes of dynamic tire loads to the pavement and bridges leading to their fatigue and premature failure. The dynamic wheel loads, arising from the dynamics of the vehicle and the tire-road interactions are believed to be an important cause of the premature pavement damage. In this thesis, the tire loads are analyzed in terms of Dynamics Tire Force and Dynamic Load Coefficient (DLC). Dynamic interactions between the tire and road surface cause considerable fluctuations in the tire loads. Such fluctuations about the static load are refereed to as the dynamic tire forces. DLC is a convenient measure of variation in the tire force over a period of time. It is a statistical measure reflecting tire force deviation from a mean tire force. The DLC is defined as: $DLC = RMS\ dynamic\ tire\ force / Mean\ tire\ force$ [42, 43, 44]. Both dynamic wheel loads and ride quality are directly related to the suspension mode associated with vertical motion of the vehicle. Therefore the influence of the different damping systems on the DLC is thus further investigated as a function of the vehicle speed and the road roughness.

City Road (Cote Des Neiges)

The dynamic tire load characteristic is firstly investigated with the input of city road (Cote Des Neiges). Figure 3.17presents PSD of dynamic tire force at 50 km/h produced by all three regenerative damping systems. The results show that the discontinuous asymmetric and continuous asymmetric system produce similar responses throughout the frequency range from 0.1 Hz to 90 Hz. The symmetric slightly improves the performance in frequency range between sprung and unsprung mass natural frequencies, while its peak value excesses that of discontinuous asymmetric by 12.6% around the natural frequency of sprung mass. The Comparable responses of Dynamic Load Coefficient (DLC) produced by the three regenerative damping systems are presented in Figure 3.18 while the vehicle moving speed is varied from 10 to 80 km/h. The results show that the symmetric system slightly reduces the DLC in the speed range from 10 to 55 km/h, while producing relatively high level of DLC in the speed range above 55 km/h.

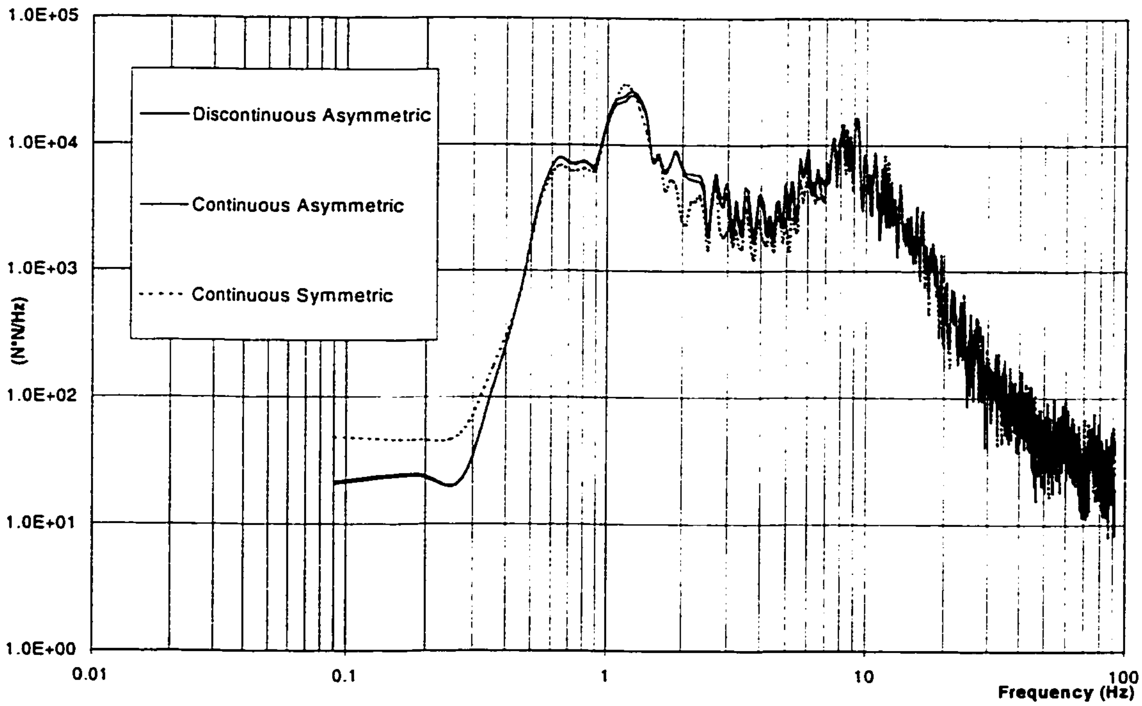

Figure 3.17PSD of Tire Dynamic Load (CDN)

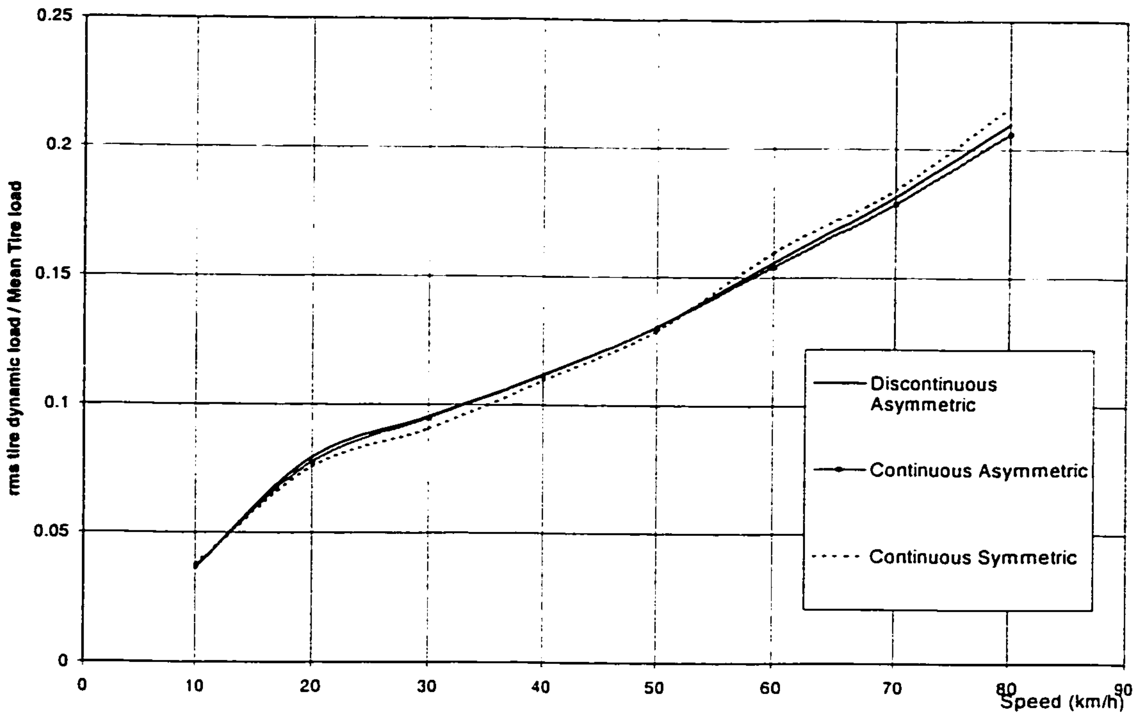

Figure 3.18 Dynamic Load Coefficient (CDN)

Off Road

The PSD of tire dynamic load produced by the three regenerative damping systems on off road at speed of 30 km/h is presented in Figure 3.19. The results show that the symmetric damping system yields significant performance improvement in the frequency range between 1 Hz to 6 Hz, while all three regenerative damping systems provide similar performance at other frequencies. Figure 3.20 illustrates the Dynamic Load Coefficient of three damping configurations. In the speed range from 10 to 60 km/h discontinuous and continuous asymmetric systems provide approximate performance, while symmetric system slightly improve the performance in speed range from 10 to 40 km/h.

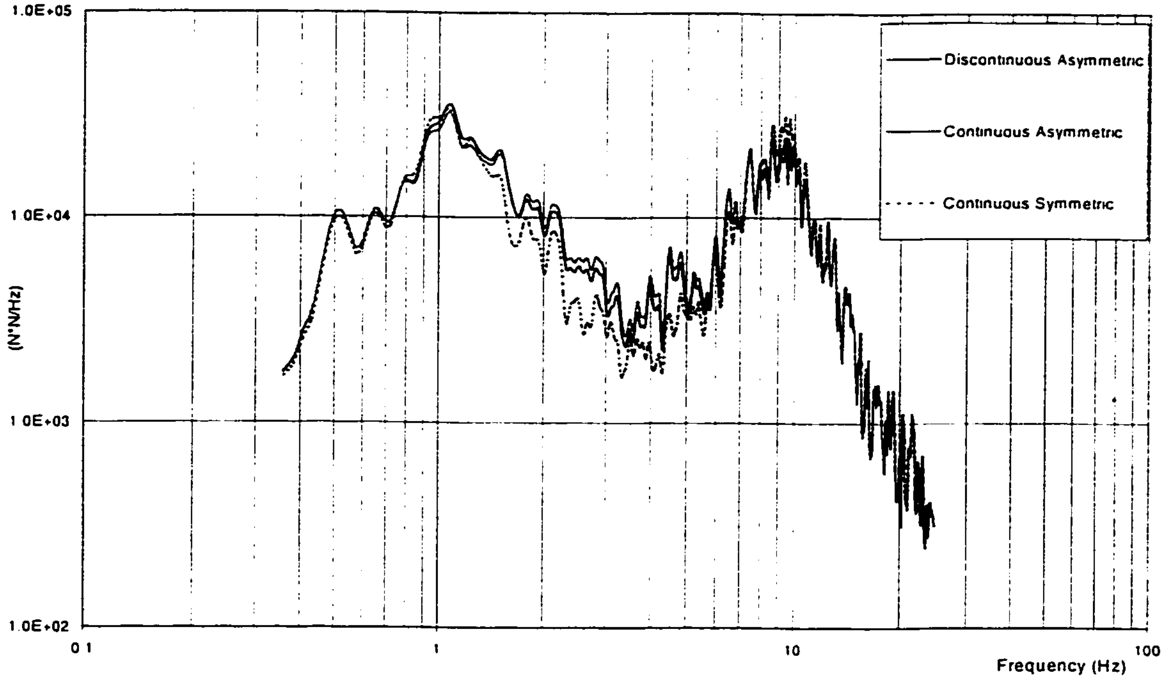

Figure 3.19 PSD of Tire Dynamic Load (Off Road)

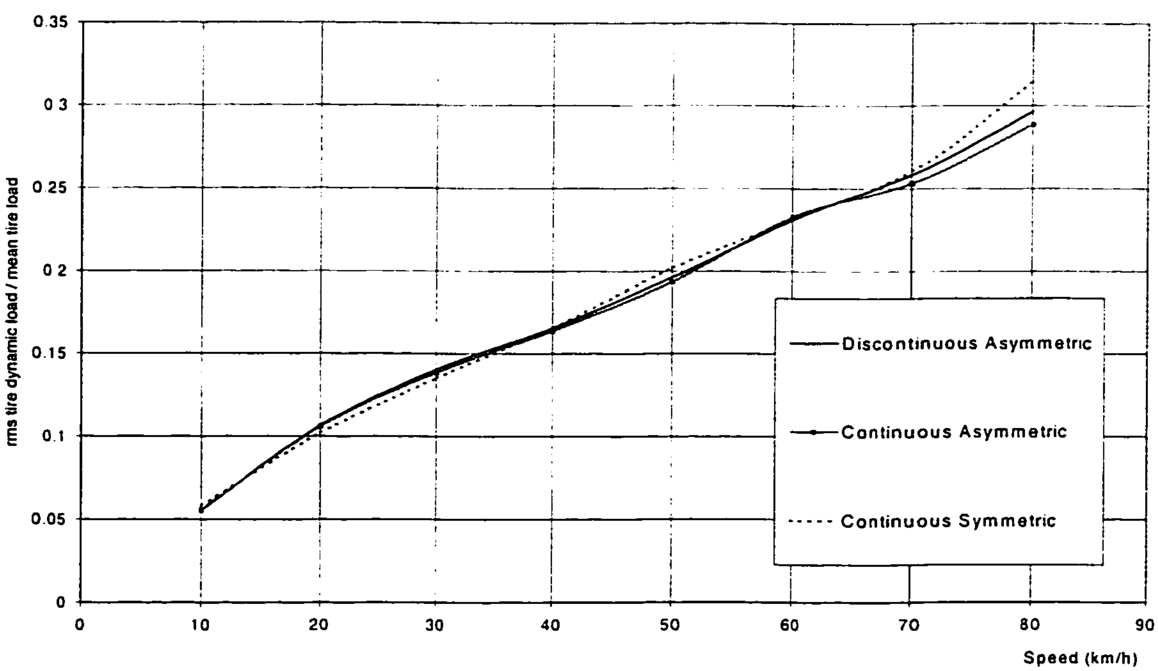

Figure 3.20 Dynamic Load Coefficient (Off Road)

3.4 Summary

The ride quality and dynamic wheel load characteristics of the vehicles equipped with regenerative damping systems are analytically investigated with deterministic and stochastic road excitations. The ride quality and dynamic load characteristics of the regenerative damping systems are discussed on different road excitations with different speeds. The results suggest that a regenerative damper can provide the characteristics of a conventional damper and replace it in vehicle suspension systems. The further evaluation of energy regenerative efficiency of regenerative damping systems is conducted in Chapter 5.

Chapter 4

Analysis of Energy Regeneration on Selected Roads

4.1 Introduction

The energy regenerative type of damper and suspension system, that can convert the vibration energy into electric energy, is analyzed. In previous chapters, the ride performance of these hybrid suspension systems have been studied to demonstrate that a regenerative damper can produce the characteristics of a conventional damper and can thus be used to replace the suspension damping device. In this chapter the energy regeneration characteristics of hybrid suspension system is studied as the function of vehicle speed and road roughness. Two linear DC generators, namely asymmetric and symmetric, with different damping properties, are applied and studied as hybrid dampers. A normal voltage charging circuit connected to an automobile battery is used as the energy regenerative circuit. The regenerative dampers can regenerate vibration energy converting it into electric energy when the induced voltage is higher than the voltage of the charging circuit. The regenerative dampers have a dead zone for low speed motion between the sprung and unsprung masses, in other words, when the induced voltage is under the sum of the diode voltage in the circuit and the battery voltage. To evaluate the energy regenerative characteristics of proposed regenerative dampers, the vehicle models with the different hybrid dampers installed are simulated on computer under two type road inputs. Then the energy regenerated by the regenerative damping systems is studied and compared.

4.2 Speed influence on Energy Regeneration

Properties of the energy regeneration are studied with two road inputs, city road input, and off road input. The total power in the regenerative damping systems (POD) is calculated by multiplying the damping force and relative velocity between sprung and unsprung mass. The energy regenerative power of the generator (POG) is equal to inducted voltage in generator multiplying current in circuit, where POG should be constantly equal to POD due to the assumption of 100% efficiency for generator to convert mechanical energy into electric energy. The energy dissipated in the adjustable resistance and inner resistance of the generator (POR) is computed by total value of external and inner resistance multiplying square value of current. And the battery charging power (POB) is calculated by charging voltage multiplying charging current. All of these indicators are analyzed in terms of RMS. Comparison of energy regeneration properties of the regenerative dampers is then made.

City Road Input (Cote Des Neiges)

Figure 4.1 illustrates the RMS of the power of asymmetric system with the road input of city road of CDN. Assuming that the vibration energy is completely converted into electric energy by the regenerative damping system, POD and POG are exactly identical over the whole speed range as shown in Figure 4.1. The regenerative power of the generator POG has relatively sharp increasing rates at the speed below 20 km/h and in the high speed range above 40 km/h, while the POR shows the same trend as the POG does. The difference between the POG and POR, defined as POB, increases at a constant rate with speed. The power RMS of the symmetric system is shown in Figure

4.2. The general variation trend of POG, POD, POR and POB is similar to those of the asymmetric system.

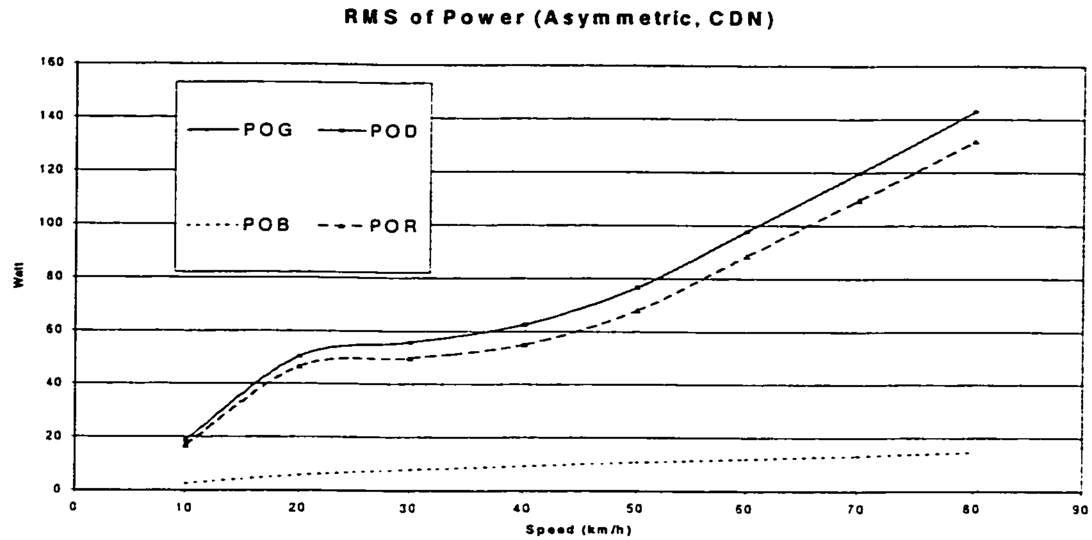

Figure 4.1 RMS of Power (Asymmetric, CDN)

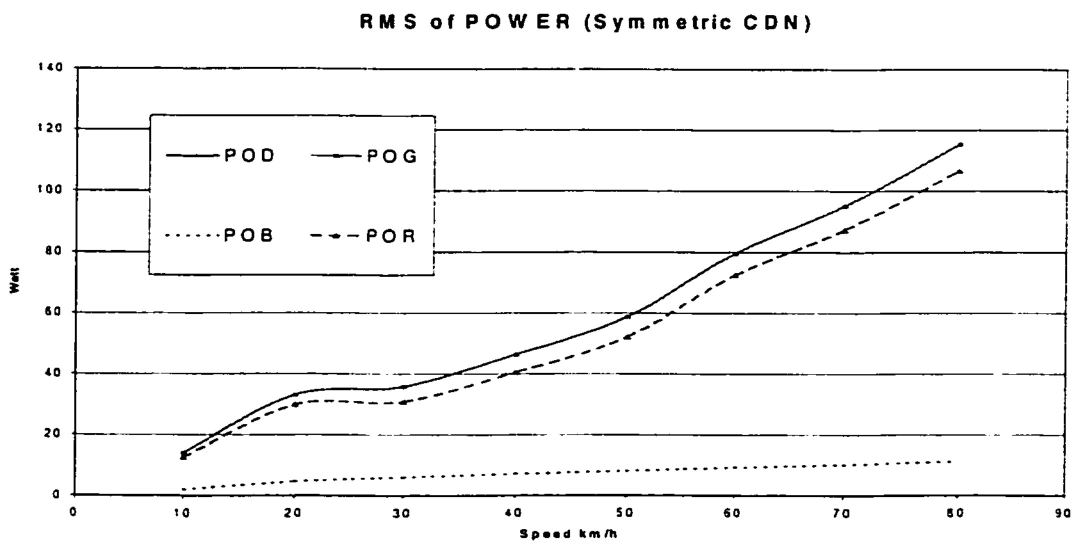

Figure 4.2 RMS of Power (Symmetric, CDN)

The comparison of the asymmetric system and the symmetric system in POG, POB and energy regenerative efficiency, defined as POB/POG, is shown in Figures 4.3, 4.4 and 4.5. Compared with the symmetric system, the asymmetric system steadily shows higher POD and POB over the whole speed range from 10 to 80 km/h with the city road input. The energy regenerative efficiency of the regenerative damping systems varies diversely over the speed range, as shown in Figure 4.5. At the low speed range from 10 to 40 km/h the symmetric damping system reveals relatively higher regenerative efficiency than the asymmetric damping system. The peak of regenerative efficiency of symmetric damping system appears at about 30 km/h, reaching 16.8%, while the asymmetric reaches its regenerative efficiency peak value of 14.7% at about 40 km/h. Since POG increases at a relative higher increasing rate than POB does in the speed range above 40 km/h, the energy regenerative efficiency of both asymmetric and symmetric damping systems thus decreases with the increasing of speed in this speed range.

The results indicate that both asymmetric and symmetric damping system show an increase energy regenerative power with increase of speed. Even though the POG and POB of the asymmetric system are steadily higher than those of the symmetric system, the symmetric damping system can convert vibration energy into electric energy more efficiently at relatively lower speed.

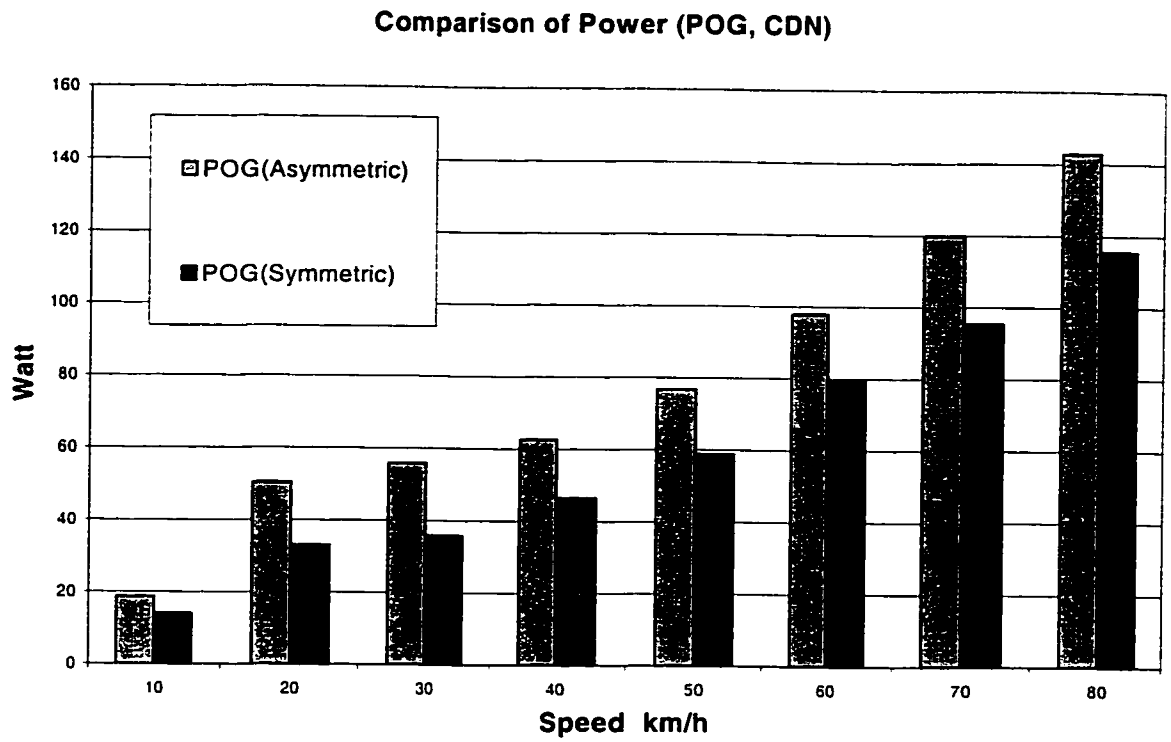

Figure 4.3 Comparison of Power (POG, CDN)

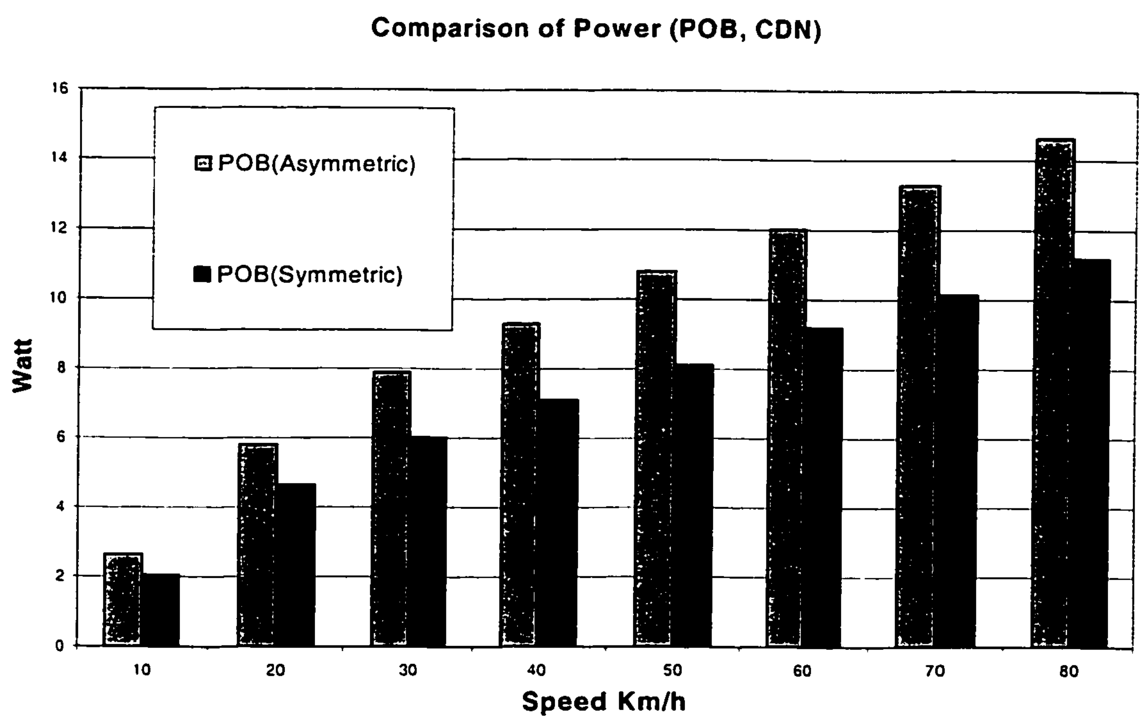

Figure 4.4 Comparison of Power (POB, CDN)

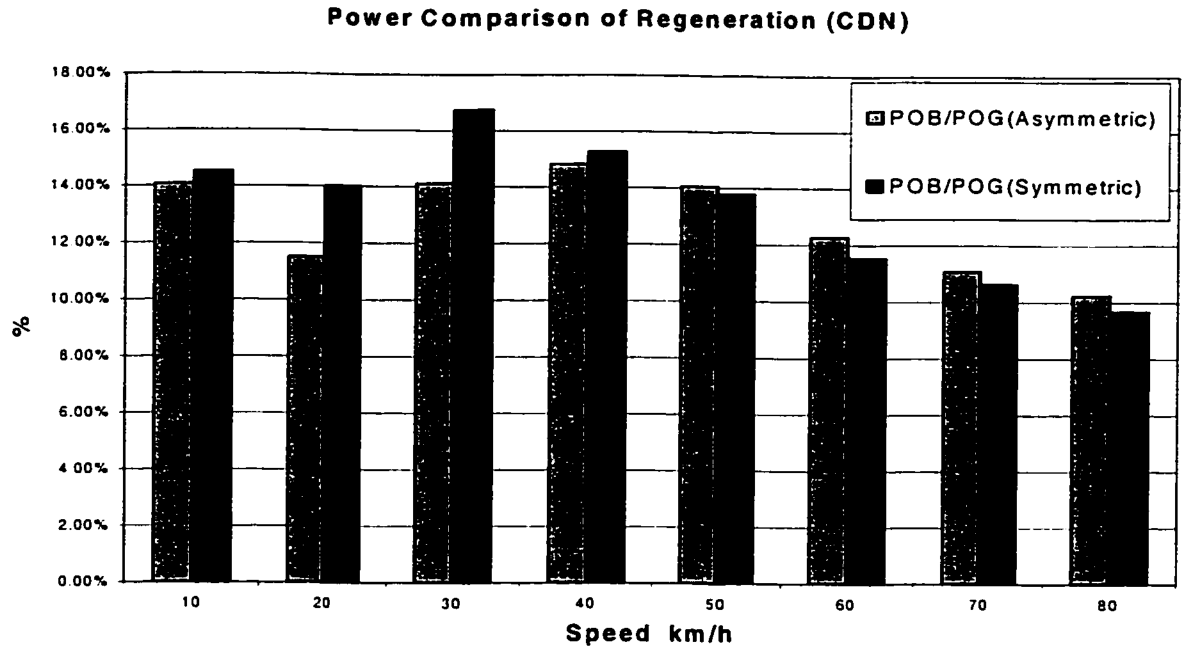

Figure 4.5 Comparison of power Regeneration (CDN)

Off Road Input

Figure 4.6 illustrates the RMS of regenerative power of asymmetric system with the off road excitations. The regenerative power of the generator POG more steadily increase with the off road input throughout the whole speed range than that with the city road input. It also shows that POG, POD, POB and POR almost increase linearly with speed in the speed range from 10 to 50 km/h. When the speed is over 50 km/h, the increase rates of POG and POR are much higher. The power RMS of the symmetric system is illustrated in Figure 4.7. It reveals that the general variation trend of POG, POD, POR and POB is very similar to that of the asymmetric system. For symmetric damping system, POG and POR vary almost linearly from 10 to 40 km/h. And above 40 km/h, the increase rate becomes higher. Compared with the result of the previous case, POG with city road excitation increases more rapidly throughout the whole speed range than it does with the off road input. The result suggests that speed yield a more significant effect in energy regeneration with city road excitation than it does with an off road input.

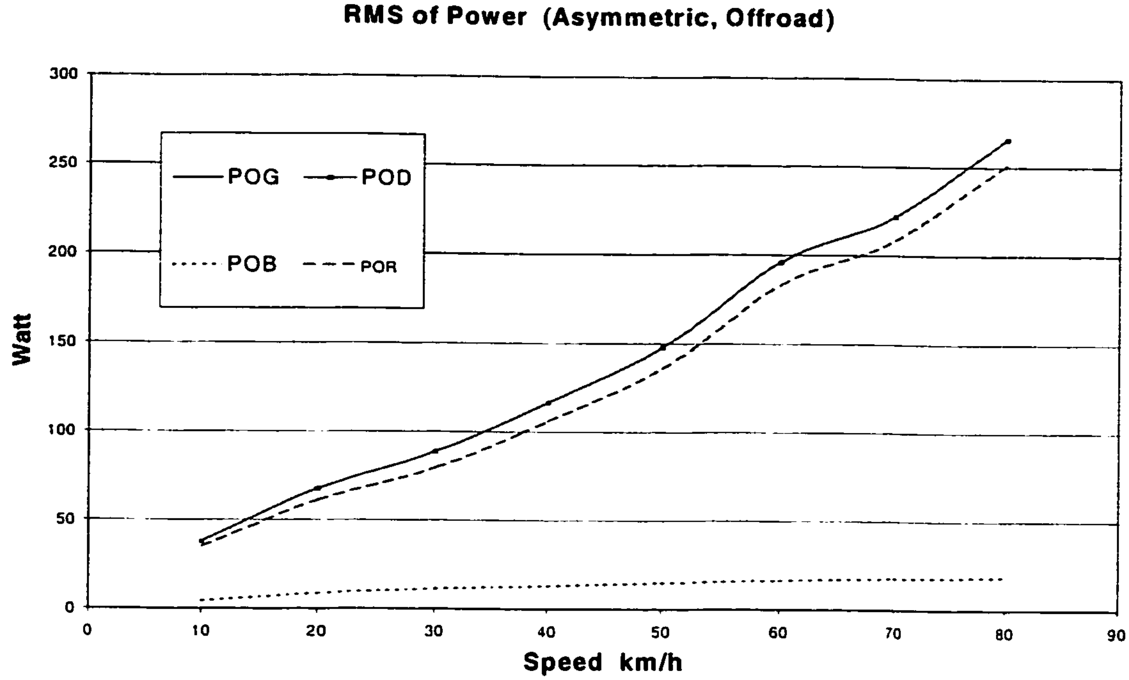

Figure 4.6 RMS of Power (Asymmetric, Off road Input)

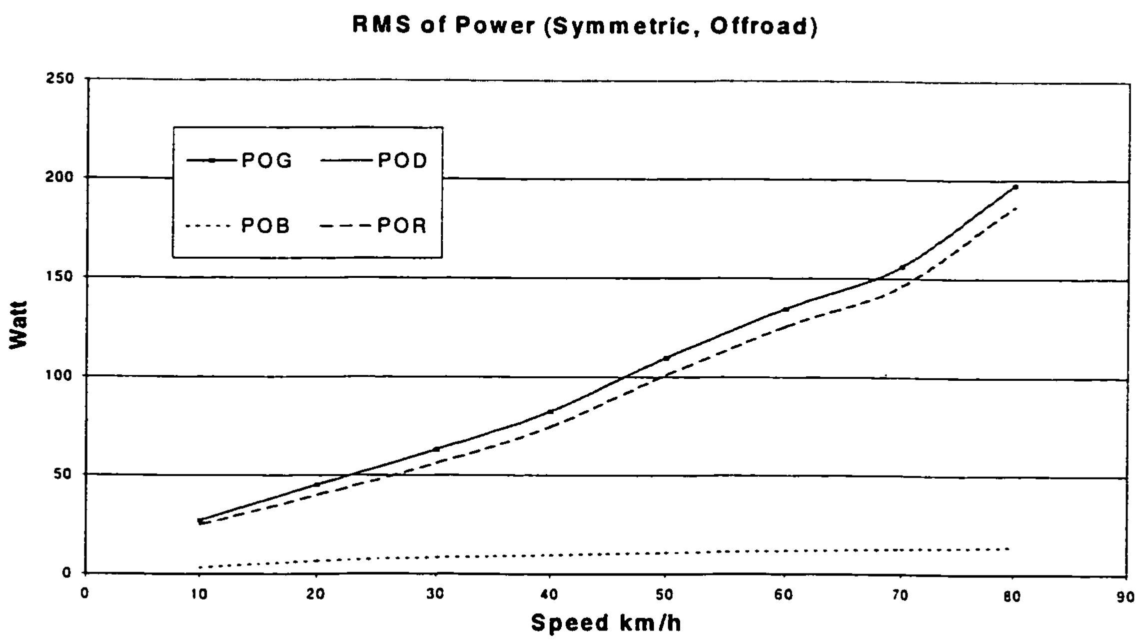

Figure 4.7 RMS of Power (Symmetric, Off Road Input)

Figures 4.8 and 4.9 provide the comparison between the asymmetric system and the symmetric system in terms of the POG, and POB respectively. Compared with the symmetric system, the asymmetric system steadily shows larger POD and POB in the whole speed range from 10 to 80 km/h with the off road input. Figure 4.8 shows that the rates of the POG sharply increase with speed for both asymmetric and symmetric damping systems, while Figure 4.9 indicates that the increase rates of the POB decrease with speed for the both damping systems. The regenerative efficiency of the regenerative damping systems is illustrated in Figure 4.10. As anticipated, both the asymmetric and the symmetric damping systems reach their highest regenerative energy efficiency at low speed and decrease with speed. The result also shows that the regenerative efficiency of the symmetric is higher than that of the asymmetric system at equal speed, except at 50 km/h. The peaks of the two systems appear at about 20 km/h, which is 12.8% and 14.6% for asymmetric and symmetric respectively.

The results with the off road input indicate that energy regenerative power of both asymmetric and symmetric steadily increase with speed, while the peaks of regenerative efficiencies of the two systems occur at low speed. They suggest that off-road-vehicle with a regenerative damping system installed has an economical speed range from 10 to 40 km/h

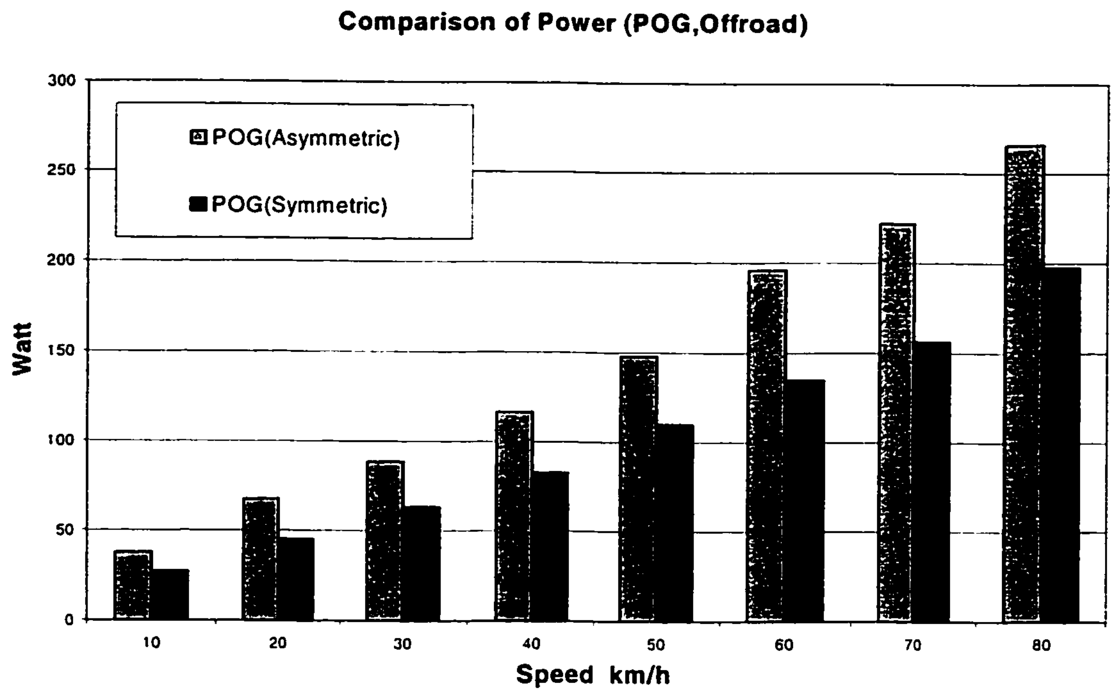

Figure 4.8 Comparison of Power (POG, Off Road)

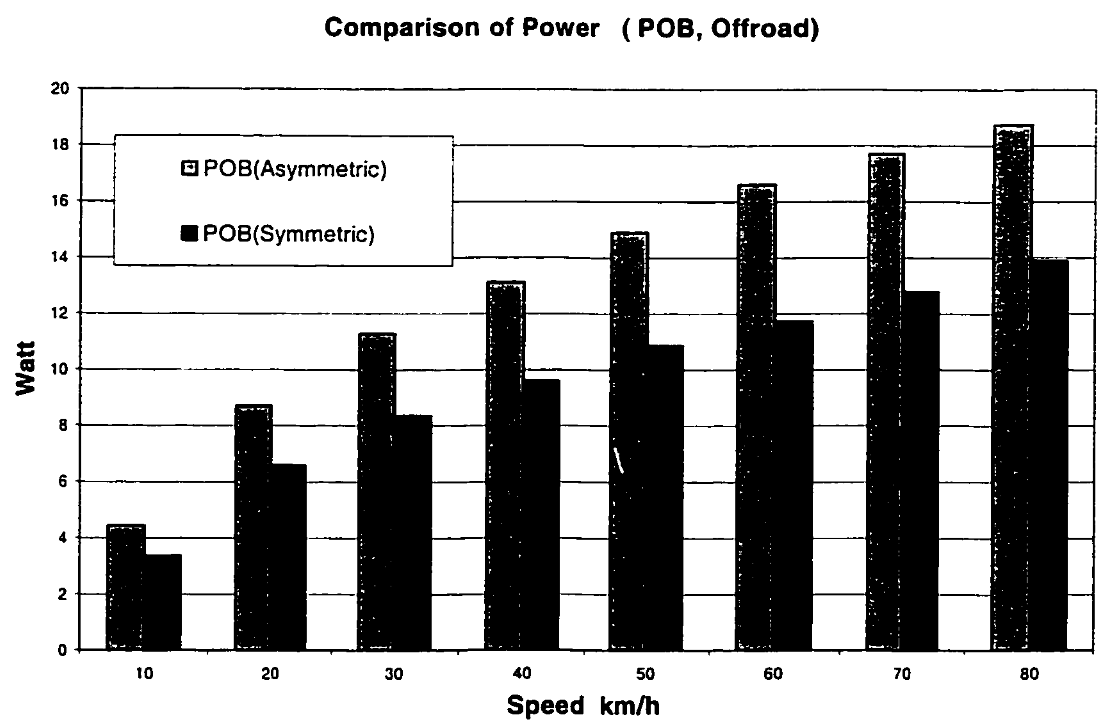

Figure 4.9 Comparison of Power (POB, Off Road)

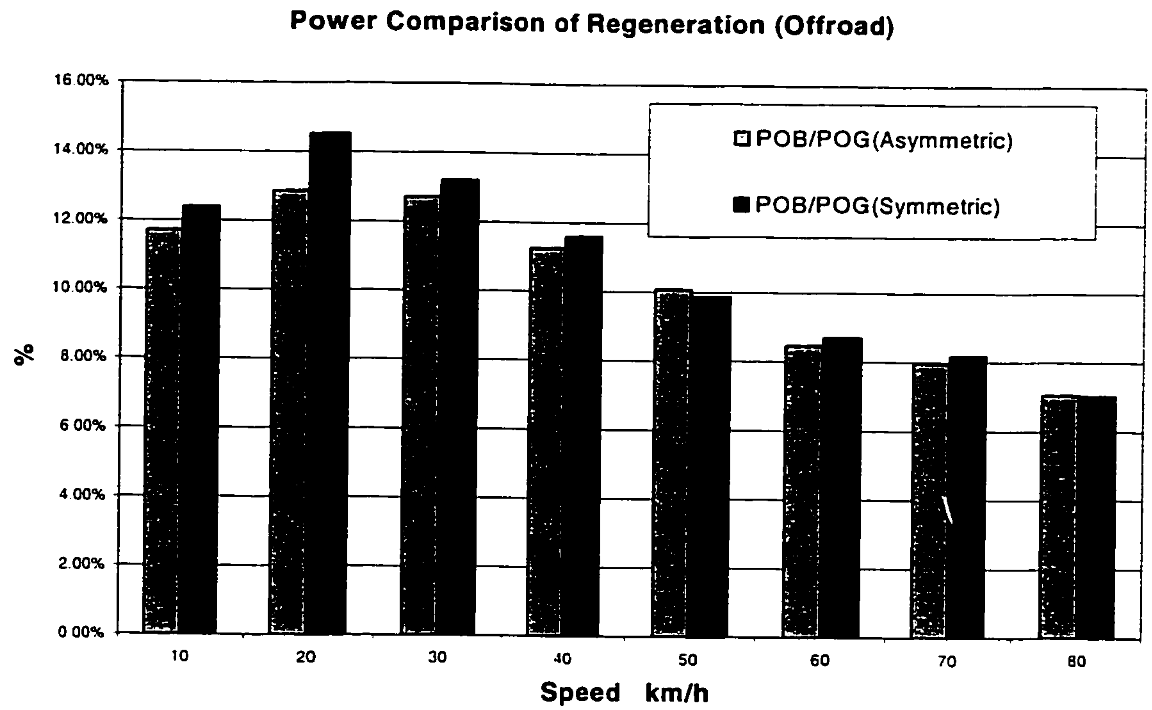

Figure 4.10 Power Comparison of Regeneration (Off Road)

4.3 Road Roughness Effect

Figures 4.11 and 4.12 present the road roughness effect on energy regeneration of the asymmetric system in terms of POG, POB and regenerative efficiency. Figure 4.11 shows that the POG of the asymmetric system with the off road input is the highest of the two inputs. The POG with the off road input at the speed of 30 km/h is about 1.6 times higher than that with the city road input at the same speed. Compared numerically the POG of the off road input at 40 km/h is, around 117 watts, which is very close to the POG of the city road input at 70 km/h, a value of 120 watts. It reveals that the regenerative power in the generator with off road input is greatly affected by the road roughness, and roughness yields more significant effects in energy regeneration than speed. Similar situation is also presented in Figure 4.12, which reveals the POB of asymmetric system with the city road input and off road input. In other words, the regenerative damping system can convert more vibration energy into electric energy and save it to battery, when the vehicle moves on the off road.

Figure 4.13 and 4.14 illustrate energy regeneration of the symmetric system. It also reveals that the road roughness has significant effects on the energy regeneration of the regenerative damping system, where it can convert much more vibration energy into electric energy with the off road input than it does with city road input.

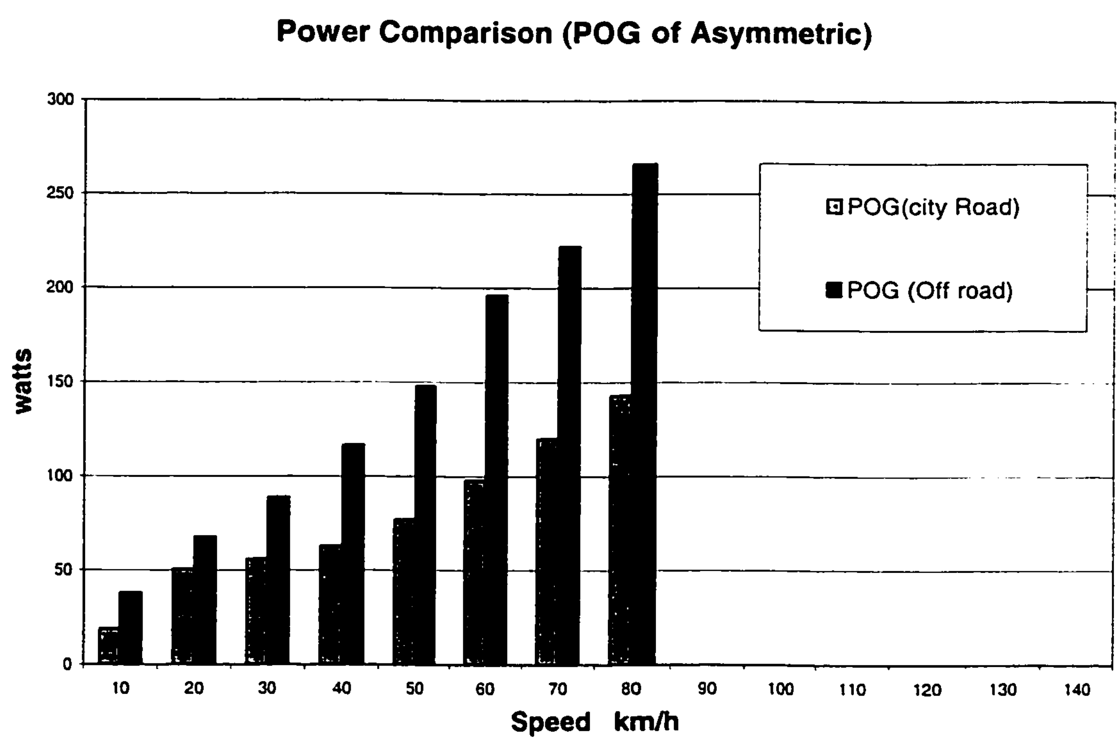

Figure 4.11 Power Comparison (POG of Asymmetric)

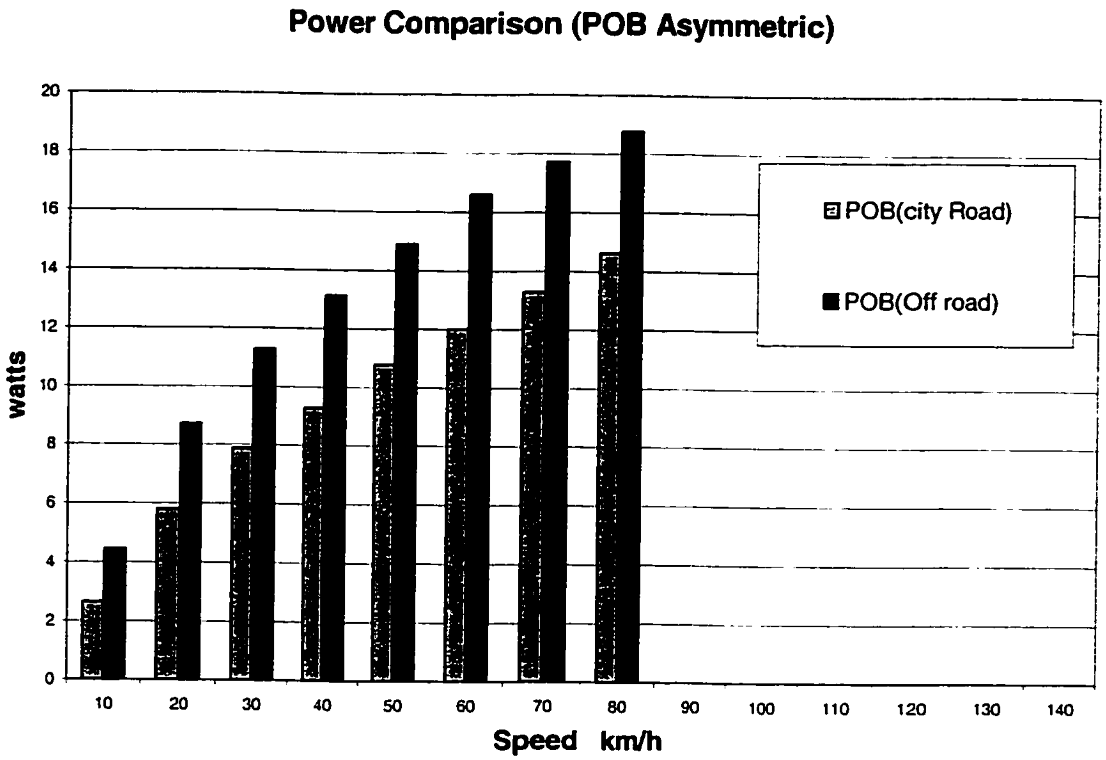

Figure 4.12 Power Comparison (POB Asymmetric)

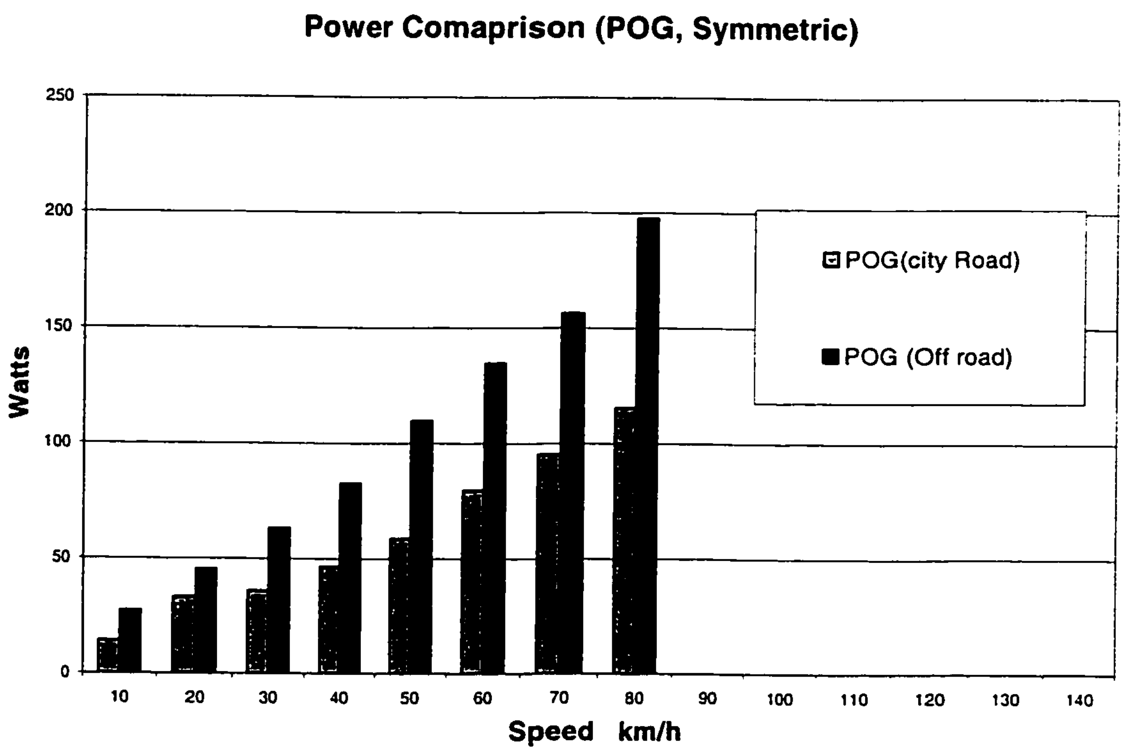

Figure 4.13 Power Comparison (POG Symmetric)

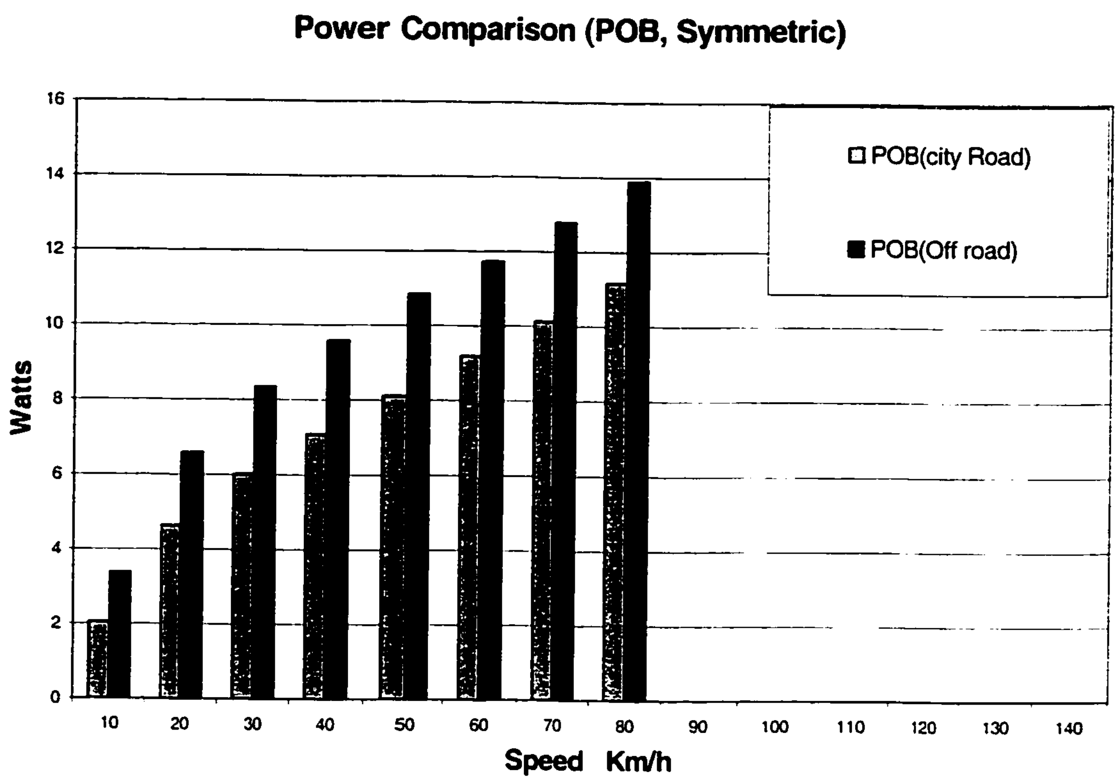

Figure 4.14 Power Comparison (POB, Symmetric)

4.4 Summary

In this chapter, the effects on the energy regeneration of the regenerative damping systems, which comes from the vehicle speed and the road roughness, are studied. Analysis reveals that regeneration powers always steadily increase with speed, in other words the higher vehicle speed the higher regenerative power. The study also indicates that road roughness yields a significant effect on the energy regeneration either, and the influence of road roughness is more significant than that of vehicle moving speed.

Chapter 5

Conclusions and Recommendations for Future Work

5.1 Highlight of the Study

Although most studies on the regenerative suspension systems investigated the design and energy regeneration of regenerative dampers, only few studies investigated the effect of regenerative damping systems on ride performance and dynamic wheel loads, which results in the reduction of tire induced road damage. The reported studies, however, are mostly limited to linear damping. In this thesis, regenerative dampers with nonlinear properties are formulated, and the performance potentials and energy regeneration of different regenerative dampers are investigated. In a view of the improvement of ride quality, dynamic wheel loads and energy regeneration efficiency, it has been proven that employing the proper regenerative dampers can improve the performances of the vehicles on different roads, while regenerating the electric energy from mechanical vibration motion. The highlights of findings are described below:

- A quarter vehicle model of a compact car is formulated for purpose of studying its performance and energy regeneration capability with different types of dampers. Neglecting contributions of the roll and pitch dynamics of the vehicle, vertical acceleration and vertical tire loads exerted on the road are analyzed. The vehicle body is represented by a rigid sprung mass with a single vertical degree of freedom. The wheel and axle assembly is represented by a rigid mass also with a single vertical degree of freedom. The tire road interactions are represented by a linear force-deflection relationship of the tire assuming point contact with a rigid road. The vehicle models employing different regenerative damping systems are analyzed with deterministic and stochastic inputs arising from sinusoidal, city and off roads.

- The regenerative dampers, employed in the vehicle suspension model, should exhibit variable damping characteristics associated with the relative velocity of the sprung and unsprung mass. A linear DC generator and a circuit connected to the generator are used. In order to fulfill the function of a vibration damping, the damping coefficient of the linear generator used as a mechanical damper should rapidly vary with the relative velocity of sprung and unsprung mass. This is achieved by changing the external resistance connected to the coil of the generator.

- A performance criterion, comprising acceleration transmissibility, RMS and PSD acceleration response at sprung mass has been formulated in order to study the ride quality of the damping systems. The ride quality characteristics of the vehicle with conventional and regenerative damping systems installed are investigated with the deterministic and stochastic road excitations arising from different roads and different speeds.
- In this study, the tire load is analyzed in terms of Dynamics Tire Force and Dynamics Load Coefficient (DLC). The effect of different damping systems on the DLC and dynamics tire force is formulated as a function of the vehicle speed and the road roughness.

- The energy regeneration characteristics of the regenerative damping systems is studied in function of the vehicle speed and road roughness. The effect of the speed and road roughness on the energy regeneration is studied and energy regeneration is compared for different regenerative damping systems at different road and speeds.

5.2 Conclusions of the Investigation

The conclusions drawn from the investigation are summarized in the following sections.

- A quarter vehicle model at a vertical mode of vibration can yield significant information on the ride and dynamic tire load performances characteristics achieved by the regenerative suspension systems employed in the vehicle model.

- Since the ride quality and the dynamic wheel loads are related to vehicle vibration modes, a vehicle model with the regenerative damping system formulated to study the ride performance characteristics is suitable for evaluation of its tire-road interactions.

DC linear generators offer several potential advantages for the generation of forces or motions as compared with hydraulic and other mechanical devices. The linear generators can be arranged to have very low static friction and control of current through power electronics, they offer rapid and reliable control of force level. The effectiveness of the regenerative damping systems, however, is dependent upon the tuning of the value of the external resistance. The model verification and design optimization is performed to derive parameters of an effective regenerative damping system. From the results, the following conclusions are drawn:

- When a linear generator coil is short-circuited or connected to an external resistance, the device becomes a linear mechanical damper. The damping coefficient varies with the variation of external resistance. As the external resistance can be rapidly varied electronically, a linear generator therefore can function as a semi-active damper in a vehicle suspension system.
- Roughly calculated the generator coil self-inductance time constant indicates that for reasonable sized coils and for the vibrational periods encountered in vehicle suspensions, the electrical time constant produces only small delays in the damper response.

Ride quality performance and dynamic wheel loads characteristics of the vehicle with regenerative damping systems installed are investigated in order to illustrate the capability of regenerative damper in providing the characteristics of a conventional damper and the potential benefits in improving the performance. The results of the study reveal the following:

- With harmonic excitation, the asymmetric regenerative damping system provides very similar ride performance to that of the conventional damping system, while the symmetric damping system significant improves the ride performance at low frequencies. The asymmetric system reduces the rattle space in the whole frequency range, while the symmetric system amplifies it around the resonant frequency of unsprung mass.

- With the city road input, the symmetric system provides considerable reduction in overall RMS acceleration and PSD of acceleration in high frequency range, while the asymmetric system performs similarly to the conventional system
- With the off road input, the symmetric system provides slightly lower level of response in term of RMS acceleration at the speed below 50 km/h, which is considered as the typical off road vehicle speed range. And the asymmetric system provides similar performances to that of the conventional system.
- With various road inputs, the asymmetric system provides very similar performances to those of conventional system in terms of DLC and PSD of dynamic tire loads. The results also indicate that the symmetric system provides lower level of DLC in the relative low speed ranges, which are considered as the typical speed ranges for the vehicles moving on city road and off road respectively.

The energy regeneration characteristics of the regenerative damping systems are studied in function of speed and road roughness to judge the regenerative properties of the regenerative damping systems. The conclusions are drawn as following:

- The energy regeneration, the POG and the POB, steadily increase with the increase of the vehicle speed.

- The energy regeneration, the POG and the POB, steadily increase with the increase of the roughness of roads.
- On city road both the asymmetric system and symmetric system reveal a relatively high energy regeneration efficiency between 30 and 50 km/h, while the symmetric system reaches its peak of 16.8% at 30 km/h.
- In the case of off road input, the relatively high energy regeneration rates are achieved with the two regenerative systems, over low speeds from 10 to 40 km/h, and then decrease with the increase in speed. The symmetric system shows a better energy regeneration performance than the asymmetric system over the low speed range, reaching 14.6% at 20 km/h.

- A linear DC generator meets the requirements of energy regenerative damper. The regenerative damper is an adequate device for vibration damping.
- By employing the variable resistance to the regenerative dampers, steady nonlinear damping characteristics and a steady damping force output can be obtained.
- Vehicles with a regenerative damping system installed offer an ideal combined performance at low speeds, while the hybrid damping system offers an improved ride quality and road-friendliness and the energy regeneration efficiency at its the peak.
- The symmetric system is recommended, since it provides a better performance than the asymmetric system in all facets over a low speed range.

5.3 Recommendations for Future Work

The present research work offers a significant contribution to knowledge of the effects of the regenerative damping systems on the ride quality, road-friendliness and energy regeneration. Although the study clearly demonstrates the potential benefits of the proposed concepts, a further systematic investigation need to be undertaken to fully realize the benefits.

- It is recommended to formulate a half or full vehicle model so that its roll and pitch dynamics can be taken into account.
- Further studies should be undertaken to explore different vehicle models, such as a bus and a heavy truck, to investigate the mass effect offered by the regenerative damping systems.
- The potential performance benefits of the proposed concept need to be verified through a laboratory or field investigation.
- It is also recommended to investigate more comprehensive electric circuit to improve the battery charging efficiency. A large power decay circuit is recommended for consideration to protect the electric components from the self-induction of the DC generator.

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