ELECTROMAGNETIC
ENERGY REGENERATIVE
VIBRATION DAMPING

by

Kynan E. Graves
BEE (Hons)(Melb)

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DECLARATION

This thesis contains no material that has been accepted for the award of any other degree or diploma in any university or college of advanced education, and to the best of my knowledge and belief, contains no material previously published or written by another person, except where due reference has been made.

KYNAN E. GRAVES
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<td>$\sigma$</td>
<td>Copper Conductivity $(\Omega \cdot m)^{-1}$</td>
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<tr>
<td>$\rho_{\text{COND}}$</td>
<td>Copper Density $kg/m^3$</td>
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<tr>
<td>$\zeta$</td>
<td>Damping Ratio $(Ns^2)/(kg.m)$</td>
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<tr>
<td>$\omega_n$</td>
<td>Natural Frequency $rad/s$</td>
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<td>$B_{\text{URUR}}$</td>
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<td>Overall System Benefit $kJ/year$</td>
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<td>$B_{\text{URB}}$</td>
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<td>$d$</td>
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<td>$D$</td>
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<td>Motional EMF Voltage (DC Generator) $V$</td>
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<td>Magnetic Field Frequency (Synchronous Generator) $Hz$</td>
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<td>Normalised Damping Ratio</td>
</tr>
<tr>
<td>$f_n$</td>
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<td>$f_s$</td>
<td>Switching Frequency $Hz$</td>
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<td>-------------</td>
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<tr>
<td>$I$</td>
<td>Rotating Inertia</td>
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<tr>
<td>$i_{BATT}$</td>
<td>Battery Current</td>
</tr>
<tr>
<td>$i_D$</td>
<td>Driving Machine Current</td>
</tr>
<tr>
<td>$i_{\text{Drain}}$</td>
<td>Transistor Drain Current</td>
</tr>
<tr>
<td>$i_L$</td>
<td>Inductor Current</td>
</tr>
<tr>
<td>$i_{LB}$</td>
<td>Average Inductor Current</td>
</tr>
<tr>
<td>$i_{\text{STALL}}$</td>
<td>Stall Current (Required to Overcome Friction)</td>
</tr>
<tr>
<td>$I_{\text{Total}}$</td>
<td>Total Conductor Length (Synchronous Generator)</td>
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<tr>
<td>$L$</td>
<td>Internal Machine Inductance</td>
</tr>
<tr>
<td>$I_C$</td>
<td>Electromagnet Length (Synchronous Generator)</td>
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<td>Electromagnetic Machine Voltage Constant</td>
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<td>$k_r$</td>
<td>Modified Vehicle Spring - Spring Constant</td>
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<tr>
<td>$K_T$</td>
<td>Electromagnetic Machine Torque Constant</td>
</tr>
<tr>
<td>$k_t$</td>
<td>Effective Tire Spring Constant</td>
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<tr>
<td>$k_v$</td>
<td>Vehicle Spring Constant</td>
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</tr>
<tr>
<td>$m_t$</td>
<td>Unsprung Mass (Tire / Suspension System Mass)</td>
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<tr>
<td>$m_V$</td>
<td>Sprung Mass (Quarter Vehicle Mass)</td>
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<tr>
<td>$N$</td>
<td>Number of Windings (Synchronous Generator)</td>
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<td>Number of Stop-Start Cycles Per Year - Urban</td>
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<tr>
<td>$N_{\text{URB}}$</td>
<td>Number of Stop-Start Cycles Per Year - Non-Urban</td>
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<td>$P_{\text{DISS}}$</td>
<td>Power Dissipation</td>
</tr>
<tr>
<td>$P_{\text{IN}}$</td>
<td>Regenerative Interface Input Power</td>
</tr>
<tr>
<td>$P_M$</td>
<td>Electromagnetic Machine Output Power</td>
</tr>
<tr>
<td>$P_{\text{OUT}}$</td>
<td>Regenerative Interface Output Power</td>
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<td>Average Conductor Radius</td>
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<td>$r_{\text{GEAR}}$</td>
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<td>$r_l$</td>
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<td>$r_w$</td>
<td>Normalised Coil Width (Synchronous Generator)</td>
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<td>$S_{\text{ad}(\omega)}$</td>
<td>Road Surface Spectral Density</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
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<tr>
<td>$t_{\text{OFF}}$</td>
<td>Switch Off (Open) Duration</td>
</tr>
<tr>
<td>$t_{\text{ON}}$</td>
<td>Switch On (Closed) Duration</td>
</tr>
<tr>
<td>$T_S$</td>
<td>Total Switching Period</td>
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<td>Description</td>
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<td>---------</td>
<td>-----------------------------------------------------------</td>
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<tr>
<td>( \dot{X} )</td>
<td>Damper Velocity</td>
</tr>
<tr>
<td>( V )</td>
<td>Vehicle Velocity</td>
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<tr>
<td>( \mu )</td>
<td>Conducting Material Volume (DC Generator)</td>
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<tr>
<td>( V_{BATT} )</td>
<td>Battery Voltage</td>
</tr>
<tr>
<td>( V_D )</td>
<td>Diode Voltage</td>
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<tr>
<td>( V_L )</td>
<td>Inductor Voltage</td>
</tr>
<tr>
<td>( V_{REL} )</td>
<td>Relative Damper Velocity</td>
</tr>
<tr>
<td>( V_{RUR} )</td>
<td>Vehicle Velocity - Non-Urban</td>
</tr>
<tr>
<td>( V_{URB} )</td>
<td>Vehicle Velocity - Urban</td>
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<tr>
<td>( w )</td>
<td>Conducting Coil Width (Synchronous Generator)</td>
</tr>
<tr>
<td>( W_r )</td>
<td>Regenerated Energy</td>
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<tr>
<td>( xo )</td>
<td>Input (Roadway) Displacement</td>
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<tr>
<td>( xt )</td>
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<td>( xv )</td>
<td>Vehicle Body (Sprung Mass) Displacement</td>
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</tr>
<tr>
<td>AC</td>
<td>Alternating Current</td>
</tr>
<tr>
<td>ADC</td>
<td>Analog-to-Digital Converter</td>
</tr>
<tr>
<td>DC</td>
<td>Direct Current</td>
</tr>
<tr>
<td>DOF</td>
<td>Degree-Of-Freedom</td>
</tr>
<tr>
<td>EMF</td>
<td>Electromotive Force</td>
</tr>
<tr>
<td>EPA</td>
<td>Environmental Protection Authority</td>
</tr>
<tr>
<td>EV</td>
<td>Electric Vehicle</td>
</tr>
<tr>
<td>ICEV</td>
<td>Internal Combustion Engine Vehicle</td>
</tr>
<tr>
<td>I/O</td>
<td>Input / Output</td>
</tr>
<tr>
<td>ISO</td>
<td>International Standard Organisation</td>
</tr>
<tr>
<td>LCA</td>
<td>Life Cycle Analysis</td>
</tr>
<tr>
<td>MHD</td>
<td>Magneto-Hydro-Dynamic (Energy Conversion)</td>
</tr>
<tr>
<td>PWM</td>
<td>Pulse Width Modulation</td>
</tr>
<tr>
<td>RMS</td>
<td>Root Mean Square</td>
</tr>
<tr>
<td>VLT</td>
<td>Variable Linear Transmission</td>
</tr>
<tr>
<td>ZEV</td>
<td>Zero-Emission Vehicle</td>
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This thesis documents a PhD level research program, undertaken at the Industrial Institute Swinburne, Swinburne University of Technology between the years of 1997 and 2000. The research program investigated electromagnetic energy regenerative vibration damping; the process of recovering energy from damped, vibrating systems. More specifically, the main research objective was to determine the performance of regenerative damping for the application of vehicle suspension systems. This question emerged due to the need for continuous improvement of vehicle efficiency and the potential benefits possible from the development of regenerative vehicle suspension. It was noted, at the outset of this research, that previous authors had undertaken research on particular aspects of regenerative damping systems. However in this research, the objective was to undertake a broader investigation which would serve to provide a deeper understanding of the key factors.

The evaluation of regenerative vibration damping performance was achieved by developing a structured research methodology that began with analysing the overall requirements of regenerative damping and, based on these requirements, investigated several important design aspects of the system. The specific design aspects included an investigation of electromagnetic machines for use as regenerative damping devices. This analysis concentrated on determining the most promising electromagnetic device construction based on its damping and regeneration properties. The investigation then proceeded to develop an 'impedance-matching' regenerative interface, in order to control the energy flows in the system. This form of device had not been previously developed for electromagnetic vibration damping, and provided a significant advantage in maximising energy regeneration while maintaining damping control. The results from this analysis, when combined with the issues of integrating such a system in vehicle suspension, were then used to estimate the overall performance of regenerative damping for vehicle suspension systems.

The methodology and findings in this research program provided a number of contributing elements to the field, and provided an insight into the development of regenerative vehicle systems. The findings revealed that electromagnetic regenerative vibration damping may be feasible for applications such as electric vehicles in which energy efficiency is a primary concern, and may have other applications in similar vibrating systems.
"...concerned environmentalists nowadays point to what may be the most profound threat of all over the long term: the prospect that human economic activities are creating a dangerous "greenhouse effect" of global warming, with consequences for the earth's entire ecosystem...

According to the U.S. Environmental Protection Agency, to stabilize atmospheric concentrations of CO$_2$ at the present level, carbon emissions must be reduced by 50 to 80 percent, back to the level of the 1950s.

The obstacle here is not so much industrial emissions. Eliminating CFC and CO$_2$ emissions from factories and supermarkets is costly,...The real issue is the need to cut the emissions by vehicle engine combustion...by measures including stiff rises in gasoline prices, heavy investment in fuel-economy combustion, (and) severe penalties on "gas-guzzling" automobiles..."^

Paul Kennedy
Professor of History
Yale University

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1.1. **Overview**

This thesis is the result of a Ph.D. level research program undertaken at the Industrial Research Institute Swinburne, Swinburne University of Technology, Australia, between the years of 1997 and 2000.

This thesis documents research undertaken in the field of regenerative vibration damping or, more specifically, the process of recovering energy from damped, vibrating systems. The main research objective was to determine the performance of regenerative damping in relation to vehicle suspension systems. This question emerged due to the need for continuous improvement of vehicle efficiency, and the potential benefits from the development of regenerative vehicle suspension systems.

During the course of the literature review, undertaken during the progress of this Doctoral research program, it became evident that a number of researchers had previously conducted engineering and scientific research into the recovery of energy from vehicle suspension systems. However, it was the intention of this research to provide, for the first time, a comprehensive study of the overall energy recovery process, as it pertained to vehicle damping, including both engineering and economic performance factors.

In relation to the estimation of both economic and practical performance of regenerative damping for a vehicle system, the energy-saving, or regeneration, methodology is dependent on a number of factors, including:

- The technical complexity of the proposed methodology.
- The development and production cost of the devices based upon the methodology.
- The value/cost of energy (as a global commodity).
- The energy expended for the production of the recovery devices.
- The energy expended for the inclusion of such recovery devices.
- The reliability and safety issues associated with the operation of the recovery device.
• The perceived market/environmental need for (and marketing potential of) the proposed recovery device.

Each of these factors can be complex in its own right and, moreover, some of the factors are interdependent and cross traditional disciplinary boundaries (including Engineering and Economics, for example).

Due to the nature of the research objectives the research documented in this thesis begins with a broad overview of the field, with the aim of answering two fundamental questions:

• What are the main design requirements for the recovery system?
• What type of regenerative damper has the most potential for implementation in a vehicle system?

As the thesis develops, the documented research concentrates on more specific areas that are uncovered as major determinants of the system performance. An example of this is the analysis presented in Chapter 5, which specifically investigates the regenerative interface required between a regenerative damper and a storage device. Another example of the specific research, covered in this thesis, is the issue of integrating a regenerative damper in a vehicle system, which is covered in Chapter 6. The final analysis, which is presented in Chapter 7, specifically investigates the overall performance of regenerative damping in vehicle systems, and draws on the analysis and results of all previous thesis chapters.

In this (Introduction) chapter, Section 1.2 provides the reader with the background of, and motivation for, undertaking this research. The concept of how regenerative vibration damping is related to vehicle systems, and how this form of damping may lead to the benefits of improved vehicle energy efficiency is discussed. The thesis 'problem statement' is then defined in Section 1.3, in order to provide a concise description of the specific objectives of the research project. The perceived contributions to knowledge for each thesis chapter are outlined in Section 1.4.
1.2. **Background of Problem**

In the latter decades of the 20th Century, the overall value of passenger vehicle transportation was measured according to several performance factors. These factors included cost, reliability, power, road handling and passenger comfort. However, following on from a perceived energy crisis in the 1970s, another vehicle parameter, which was increasingly considered as a determinant of a vehicle's quality, was energy efficiency [1]. By the 1990s, this was especially true for electric vehicles, in which efficiency was a primary concern. At the time this Doctoral research commenced, considerable research, development and commercialisation of electric vehicles had already taken place. For example, prototype and commercial electric vehicles had already been produced by manufacturers including:

- General Motors,
- Honda,
- Peugeot,
- Ford, and
- Daimler-Chrysler.

However, one of the greatest impediments to consumer acceptance of such vehicles was the potential vehicle range between recharges [2]. This, in turn, was governed by the overall energy efficiencies of the vehicles.

Improved vehicle energy efficiency can be achieved either by reducing the energy required to propel the vehicle, or by more efficiently producing the energy required for propulsion [3]. For efficient vehicle propulsion, energy needs to be delivered to the driving wheels as efficiently as possible. This efficiency is brought about by the minimisation of vehicle power-plant losses, and losses associated with the transmission and drive-train. Moreover, matching between a vehicle engine and wheels via a
transmission is needed to ensure that an engine is working at an efficient operating point.

The energy required to propel a vehicle is governed by aerodynamic resistance and rolling resistance [3, 4]. Aerodynamic resistance may be reduced by streamlining the external vehicle body to reduce the aerodynamic drag. This leads to energy savings due to a reduction in the drag coefficient. The reduction of tyre rolling resistance and the development of efficient road surface textures has also been investigated in order to reduce energy loss [4].

In some instances, however, it is not possible to minimise energy loss due to the inherent operational nature of the particular vehicle system. A method of improving the energy efficiency in these systems is to regenerate, or recover, the energy that is ordinarily dissipated to the external environment. The transference of the regenerated energy back into the vehicle system then leads to an improvement in the overall energy efficiency.

One example of the use of energy regeneration in vehicle systems is regenerative braking. Regenerative braking uses a form of power transmission to direct energy that is normally dissipated in frictional brakes, into a power storage device [5]. It had been shown that, ideally, for electric or hybrid vehicles in urban traffic, almost 65 per cent of a vehicle's propulsion energy was available for regeneration using this method [6]. More realistic estimates of the total energy recovered using regenerative braking were around 20 per cent for urban driving and 6 per cent for highway driving [7]. These estimates included the inefficiency of the regenerative braking process (in this example around 65 per cent efficiency) and inefficiency of the battery and electrical conversion process. Because the effectiveness of regenerative braking reduces with vehicle speed, more recent investigations focused on combining electromagnetic regenerative braking with conventional frictional braking. The aim was to both maximise regeneration efficiency, as well as preserve the "pedal feel" of conventional brakes [8].
Another example of energy regeneration in vehicle systems is the use of heat energy from the internal combustion engine. The energy efficiency of internal combustion engines is typically in the order of 10 - 15 per cent [6], with a significant proportion of the unused energy lost as heat [9]. This heat energy can be recovered and used within the vehicle system for applications such as cabin warmth and preheating fuel. Therefore, energy efficiency is improved due to regenerating the otherwise wasted energy.

Another vehicle component, which was the specific subject of investigation in this Doctoral research, and had potential for energy regeneration, was the vehicle suspension system. A conventional vehicle suspension system contains the passive elements of a viscous damper and spring. The purpose of the damper is to limit the vertical wheel motion caused by a vehicle traversing an uneven roadway surface. However, the inherent nature of the damper, as a non-conservative element, results in energy being wasted. In the case of a conventional hydraulic vehicle damper, vibrational energy is converted into heat due to the movement of the damping fluid. Traditionally, the heat energy was dissipated to the external environment, and not practically applied elsewhere. In a regenerative vehicle suspension, a regenerative damper replaces the hydraulic damper, and recovers the otherwise wasted energy.

There are several benefits from improving the energy efficiency of a vehicle system. An increase in fuel economy is a major improvement for Internal Combustion Engine powered Vehicles (ICEVs). For the vehicle owner, a major advantage is the reduction in vehicle running costs due to lower fuel consumption. From an environmental perspective, however, lowering fuel consumption generally results in reduced vehicle emissions. This leads to a reduction in 'green house' gases as well as the lowering of atmospheric pollution levels.

At the time this research commenced, regenerative vehicle suspension had had relatively little research recognition. This was, in part, due to the assumption that suspension loss was negligible in comparison to energy usage for combustion engine vehicles. However, for applications such as electric vehicles, the opportunities for recovery were potentially more significant. It was, therefore, the objective of this Doctoral research to address the
issue of regenerative vibration damping, and assess the technical and economic performance of such a system for use in vehicle suspension.

1.3. Objectives

The broad objective of this research program was to investigate the use of a regeneration scheme in vehicle suspension systems. More specifically, however, this research was undertaken according to the following problem statement:

"To investigate various means of recovering energy from damped, vibrating systems; to select the most promising alternative and to assess the performance of such a system for use in vehicle suspension systems."

As a starting point for the research, it was noted that there had already been several previous research investigations into regenerative vibration damping and its application for vehicle suspension systems. However, when applied to vehicle systems, previous regeneration research had mainly focused on energy regeneration as a means of improving the efficiency of semi-active damping systems, rather than for regenerative dampers in their own right. Some researchers have previously speculated as to the potential feasibility of these systems. For example, when experimentally analysing energy loss in vehicle suspension systems, Browne and Hamburg [10] stated that:

"...the interest in this type of application (recovering suspension energy) would be especially high in those special applications such as electric cars where fuel economy/vehicle range are primary concerns."

However, when referring to the use of regenerative vehicle suspension, Hedrick [11], editor of Vehicle System Dynamics, stated that:
"...not much power is dissipated in the dampers for reasonable speeds and roughness and that considering the practical difficulty of constructing devices to recover a portion of this power there is little chance that this line of thought will lead anywhere."

By undertaking an objective performance analysis in this field, this research program evolved into what was believed to be the first attempt at specifically investigating regenerative suspension with the objective of establishing its potential feasibility (or lack thereof) for vehicle systems.

The application of electromagnetic devices for use as regenerative damping elements formed a significant proportion of the regenerative damping analysis undertaken in this Doctoral research. Compared to other regeneration systems, electromagnetic devices offered several advantages, such as efficiency and damping response. This issue is described in greater detail (Chapter 3).

The method of analysing the performance of regenerative suspension was to perform an 'energy analysis'. In this analysis, the benefits of recovered suspension energy were weighed against the negative aspects (or costs) of the system. These costs included factors such as added vehicle weight and material production costs. The objective was to establish the parameters under which regenerative suspension would be deemed feasible.

It was recognised that there could be no absolute answer to the feasibility question, however. This was due to the large number of dimensions to the problem, including the type of regeneration scheme used, the type of vehicle, vehicle drive cycle, and so on. The objective, therefore, was to establish a framework from which emerging trends would be derived (in terms of economic benefit) and to identify where (or if) improved system design could have an influence on viability.

Although a major application for regenerative vibration damping was for vehicle systems, there were other fields that could benefit from such technology. Any vibrating mechanical system in which damping was required could benefit, especially if energy
efficiency was considered an important aspect of the design. Examples of such processes included vibrating machinery, engines or other equipment which, because of their operation, did not run smoothly. Wave power generation had also been suggested as a possible beneficiary of regenerative damping systems [12].

The field of semi-active suspension may also benefit from regenerative damping. A semi-active damper has a continuously variable damping coefficient in order to improve vehicle response and passenger comfort. The research conducted in this project, together with findings from other researchers, have indicated several advantages of combining regenerative, and semi-active damping systems.

At the time of preparing this dissertation, the research presented here, was considered to be of significance for several reasons. Firstly (and as earlier noted), recent efforts had been undertaken to develop electric vehicles for the commercial market. This was largely a result of legislation, such as imposed by the California Air Resources Board, which governed the production of zero-emission vehicles (ZEVs) [13]. Under this legislation, five per cent of new cars delivered for sale by major manufacturers in California had to be zero-emission vehicles by 2001. This increased to ten per cent by the year 2003. Another aspect of undertaking the research was due to the improvements in the technology associated with regenerative vibration damping. Areas such as semiconductor based power-electronic devices and permanent magnet technology were continually improving. This potentially led to the improved viability of electromagnetic regenerative vibration dampers, due to improved reliability, and lower device mass, volume and cost.
1.4. Perceived Contributions of this Research

Following on from an extensive literature review into the field, and following a critical self analysis of the research, it was perceived that a number of contributions to knowledge had arisen from this Doctoral research. Where practical, these contributions were submitted for independent peer review through publication in international refereed forums. The arising publications are listed in Appendix A and are also cited herein where relevant.

Specifically, the perceived contributions of this research are:

(i) Performance Analysis of Regenerative Damping (Chapter 7)

A significant outcome of this research was the investigation into the performance of regenerative damping for vehicle suspension systems. Although previous research investigations analysed particular aspects of regenerative vibration damping, it is believed that this work was the first documented attempt at providing an objective measure of the performance of regenerative damping in vehicle systems.

A methodology was proposed to obtain an objective performance measure, and was based on a cost-analysis for the vehicle system, and regenerative damping system. The results of this analysis were included in a paper published in the Proceedings of the 1999 Society of Automotive Engineers Australasia, Young Engineers Conference, entitled: "Cost Function Analysis of Regenerative Vehicle Systems".
(ii) Electromagnetic Damping Devices (Chapter 4)

An analysis of the damping and regeneration potential of two fundamental electromagnetic designs, the DC generator and the synchronous generator was undertaken. It is believed that this was the first generalised analysis of these two forms of electromagnetic devices with respect to their damping and regeneration characteristics. The results of this analysis led to a paper published in the Journal of Sound and Vibration, entitled: "Theoretical Comparison of Motional and Transformer EMF Device Damping Efficiency".

For the first time, a generalised topology structure for the DC generator was published specifically for the analysis of electromagnetic damping devices. The results of this analysis were included in a paper published in the International Journal of Vehicle Design, entitled: "Electromagnetic Regenerative Damping in Vehicle Suspension Systems".

(iii) Regenerative Energy Interface (Chapter 5)

It is believed that an original contribution was made in the field of electromagnetic regenerative vibration damping due to the development of a new type of energy interface. The importance of impedance matching was previously revealed for mechanical, regenerative dampers. However, this was the first such system specifically for use with electromagnetic devices.

(iv) Vehicle Integration of Regenerative Damping (Chapter 6)

A contribution was made for the analysis into energy dissipation in vehicle suspension systems. A theoretical analysis of vehicle suspension energy dissipation was undertaken by modelling the vehicle dynamic system and the road
surface. The theoretical results were compared against experimental energy
dissipation obtained in other research investigations. The comparison revealed that
the dissipation estimates between theory and experiment coincided reasonably
accurately. The results of this analysis were included in the proceedings of the 32\textsuperscript{nd} International Symposium on Automotive Technology and Automation (ISATA), entitled: "Regenerative Vehicle Suspension".

Another contribution of knowledge arose from an analysis of the advantages and
disadvantages of using rotating dampers in vehicle suspension. Previous research
indicated that the use of rotating dampers of reasonable mass was not suitable for
use in vehicle suspension. However, a theoretical investigation undertaken for this
research revealed that, by using a slightly more complicated vehicle model or by
modifying how the rotating damper was placed in the suspension, the use of
rotating dampers of reasonable mass could be feasible. The results of this analysis
were also included in the paper published in the International Journal of Vehicle
Design, entitled: "Electromagnetic Regenerative Damping in Vehicle Suspension
Systems".

Moreover, an assessment of the maximum rotating mass for a rotating damper was
given. This analysis was determined by analysing the acceleration characteristics
of the suspension system, and the impact the acceleration had on the vehicle
occupants. The result had significant implications for electromagnetic regenerative
suspension due to inherent advantages of rotating damper designs.
"The scientist takes off from the manifold observations of predecessors, and shows his intelligence, if any, by his ability to discriminate between the important and negligible, by selecting here and there the significant stepping-stones that will lead across the difficulties to new understanding." 

Hans Zinsser

2.1. Overview

This Literature Review presents a study of investigations undertaken by other researchers in fields relevant to this research dissertation. The objectives were to acquire an understanding of the context of the research and provide an impetus for, and background to, the project methodology.

1 Source: Hans Zinsser, As I Remember Him, Macmillan, London, 1940.
Section 2.2 gives a review of regenerative suspension research, which covers an overview of research on the design of regenerative dampers, including electromagnetic regenerative dampers. Section 2.3 documents the survey of the field of electromagnetic damping in general, with a particular emphasis on electromagnetic dampers used in vehicle suspension systems. Section 2.4 reviews recent investigations on vehicle rolling resistance, and includes a survey of energy loss in both the vehicle tire and suspension systems. Section 2.5 analyses the methods other researchers have used in developing a measure of the system performance, in particular cost-analysis of vehicle systems. Section 2.6 contains a review of patent applications in the area of regenerative suspension systems. Finally, Section 2.7 summarises the overall literature review findings, including limitations of the reviewed research investigations. In this section, suggestions for further research in the field of electromagnetic regenerative vibration damping are provided, which gave the impetus for the research conducted here.

The sources used for the review of regenerative vibration research included both journal and conference publications as well as patent applications. However, the majority of information was obtained from journal articles. A search was undertaken for information in the following areas:

- Regenerative vibration damping systems.
- Electromagnetic dampers.
- Regenerative suspension systems.
- Electromagnetic semi-active and active suspension systems.
- Energy loss in vehicle suspension systems.
- Cost-analysis (and Life Cycle Analysis) of vehicle systems.

Although this thesis refers to many authors who have contributed to regenerative vibration research, and vehicle suspension research, there are a number of key authors that must be specifically mentioned. Suda and Shiiba [18] and Suda et al. [22, 23] investigated active vibration control using regenerated vibration energy. These investigations utilised electromagnetic devices to provide both damping and energy
regeneration. Okada and Harada [19] and Okada et al. [20, 21] analysed a similar system for active vibration control and energy regeneration. Fodor and Redfield [14] and Mahajan and Redfield [15] also described regenerative damping schemes using a theoretical analysis. These investigations had the objective of describing energy-efficient, active damping systems, with the possibility of energy regeneration. These investigations have particular significance to the research presented in this thesis, due to the specific investigation of regenerative damping systems.

Karnopp [16, 25, 28, 32] made a substantial contribution to the field of vehicle dynamics. This led to research contributions in the fields of suspension system power requirements and to the study of active and semi-active suspension systems. Ryba [26, 33, 63-65] also contributed to suspension system research. Ryba's research concentrated on the improvement of vehicle ride comfort using either passive or semi-active means. Segel and Lu [30, 31] made a large contribution to the theoretical measurement of vehicle suspension losses, and Browne and Hamburg [10] made a major contribution with a detailed experimental investigation of energy loss in vehicle suspension systems.

A key journal identified in this review and referred to in this thesis was the Journal of 'Vehicle System Dynamics'. This was due to the large amount of research covered in this journal in the areas of suspension systems, and the efficiency of vehicle systems. Other key journals include the 'International Journal of Vehicle Design' and the 'Journal of Dynamic Systems, Measurement, and Control'.
2.2. Regenerative Vibration Damping

2.2.1. Overview

The objective of this section is to gain an understanding of current knowledge in the field of regenerative vibration damping by reviewing previous investigations in the field. The findings of this section will, therefore, establish the direction of the research methodology.

2.2.2. Mechanical Regenerative Vibration Damping

Fodor and Redfield [14] stated that the primary action of a regenerative damping element is to transfer vibration energy to an energy-storing device as efficiently as possible while maintaining acceptable vibration control. Their research emphasised the importance of impedance matching between the source and the storage device to maximise the regeneration efficiency. Impedance matching referred to the ratio of 'efforts' and 'flows' in an energy transfer system [15]. The 'efforts' referred to force and voltage in mechanical and electrical systems, respectively. The 'flows' referred to velocity and current in mechanical and electrical systems, respectively [16]. Fodor and Redfield [14] described how energy storage devices exhibit a 'barrier potential' which must be overcome before storage is realised. In the instance of electrical devices, the barrier potential was the voltage across the storage battery or capacitor that must be met for charge accumulation.

A device called a Variable Linear Transmission (VLT) was proposed by Fodor and Redfield [14] as a means of achieving vibrational energy regeneration. The VLT used a mechanical method of energy regeneration, using a mechanical lever and moveable fulcrum mechanism, as shown in Figure 2.1. A mathematical model was constructed...
using bond-graphs to theoretically analyse the system. The aim was to improve a vehicle's dynamic characteristics by re-using the otherwise dissipated energy for an active suspension system, which had previously required an external energy source. The VLT was analysed using a two degree-of-freedom, quarter car model, as shown in Figure 2.2.

![Figure 2.1 Variable Linear Transmission [14].](image)

![Figure 2.2 Quarter Car Model with VLT System [14].](image)

It was shown that the VLT had the ability to store all power that would otherwise be dissipated; to provide linear damping, and required no power to drive the fulcrum actuators. However, the VLT had some limitations when analysed with a non-ideal fulcrum actuation mechanism. Firstly, the effect of a limited actuator bandwidth revealed that the VLT damping was bounded by two limiting cases. For ideal actuation, the VLT behaved as an ideal viscous damper, but for low bandwidth actuator operation, the VLT behaved as a coulomb friction damper.

The non-ideal VLT model was also analysed with respect to the operating efficiency. The fulcrum motion of the proposed VLT was broken into two degrees of freedom, with each degree of freedom requiring some input power. For this situation, with non-ideal actuators, the VLT was analysed for a typical vehicle model with random input. The
results indicated that the VLT consumed more power than it absorbed. Fodor and Redfield stated that this result did not nullify the feasibility of the VLT as a regenerative damper. They suggested that a more advanced fulcrum control strategy was needed, or the reduction of the fulcrum motion from two to one degree of freedom was needed to improve the feasibility \[14\].

Mahajan and Redfield \[15\] investigated the power requirements of active suspension systems. In this analysis an optimal active damping system was investigated, with the objective of evaluating the feasibility of such a system with energy regeneration. A one degree-of-freedom model was investigated which had an active damper in parallel with a passive damper. It was shown that, for an optimal active system, the net actuator energy was zero. In reference to the energy flows in the damping model, Mahajan and Redfield stated that, "...strictly from the performance viewpoint, there is a potential for developing regenerative (vibration damping) systems \[15\]."

It was stated that the actuator should allow efficient regeneration while providing the appropriate damping control. The process of 'impedance matching' was proposed to achieve this, and it referred to the ratio of 'efforts' and 'flows' in the energy transfer system, which was also previously described by Fodor and Redfield \[14\]. To analyse the energy flows and damping performance of the system, a regenerative actuator was modelled using a bond-graph representation.

It was stated that the feasibility of whether a system could employ a regenerative actuator (damper), would depend upon how much energy was absorbed by the actuator. The energy that could be absorbed was defined as the 'absolute regeneration potential' of the system, which ultimately depended on the damping control law that was implemented. The system was analysed using either 'isolation control' or suspension 'stroke control'. Using the theoretical model, it was found that the absolute regeneration potential was higher for isolation control, compared to stroke control \[15\].

The investigation by Mahajan and Redfield \[15\] was undertaken using a theoretical model with harmonic disturbances. It was stated that further research was being
undertaken to extend the formulation for random disturbances and, also, that more investigations were needed for such a damping model to be reliably implemented in an actual system [15].

A similar analysis to Mahajan and Redfield [15] was undertaken by Mahajan and Redfield [17]. In this analysis the objective was to investigate the energy flows in active suspension systems, and did not consider regenerative systems in detail. However, it was found that it was possible to modify the energy flows in the system by adjusting the control parameters. It was described how an approach to maximise the ratio of delivered to absorbed power could be used in the design of regenerative damping systems.

2.2.3. Electromagnetic Regenerative Vibration Damping

There have been several proposals in recent years for regenerative vibration dampers using electromagnetic devices.

Suda and Shiiba [18] investigated a regenerative damping system using an electromagnetic device. In this investigation, vibrational energy was converted to electrical energy via a rotating DC motor. Suda and Shiiba used the principle that the motion of a mass caused a change of magnetic flux density through a closed electrical circuit. The change in flux density also caused the Lorentz force to affect the mass, in the same way that it would be affected by an ordinary viscous damper.

From the theoretical analysis, Suda and Shiiba described some of the trade-offs involved with the design of such a regenerative damper. It was revealed that there was a direct trade-off between energy regeneration efficiency and the system damping coefficient, which was affected by the internal DC machine resistance and external circuit resistance. Suda and Shiiba also showed that the damping coefficient was increased by an increase in the gear ratio of the regenerative system. However, a large gear ratio also led to an increase of the moment of inertia of the rotating motor and an increase in machinery friction [18].
An experiment was undertaken by Suda and Shiiba to gain further insight into the regenerative damper [18]. Their experimental test bed is shown in Figure 2.3.

![Experimental Test Bed](image)

**Figure 2.3** Regenerative Damper Experimental Test Bed [18].

From the experimental results, Suda and Shiiba concluded that the regenerative damper had an adequate damping effect. The calculated damping coefficient was 110 (Ns/m), which nominally correlated with the theoretical damping coefficient of 99.44 (Ns/m). According to Suda and Shiiba, the efficiency of the regenerative damper was also quite good. The regeneration efficiency as a function of input frequency revealed that the maximum regeneration efficiency was close to the theoretical maximum efficiency which, in the given example, was 24.3 per cent. The efficiency was shown to reduce at high frequencies due to the small amplitude of the disturbance which led to a strong influence of machinery loss, such as gear backlash [18].

Suda and Shiiba [18] continued their investigation for a 'hybrid' suspension system. This system featured frequency-dependent regeneration and damping control. For low frequencies, active control was used to improve vibration isolation. For high frequencies, in which the energy requirements for active suspension was high, passive
control, with energy regeneration, was applied instead. The advantage of this system was that improved vibration control was achieved without a large energy consumption.

Energy regeneration and vibration isolation of the proposed hybrid suspension was analysed using numerical simulations. Suda and Shiiba demonstrated a trade-off between isolation performance and energy consumption in the damper by the change in the control function. It was also demonstrated that, for an ideal system at high frequency, the regenerated energy exceeded the consumed energy, which would theoretically enable the regenerated energy to be an energy source for active control.

The numerical investigations of the hybrid system were then followed by experimental investigations. The results revealed a trade-off between isolation and energy consumption for different control function characteristics. The amount of energy regenerated in the damper was shown to be less than the energy consumed in the actuator for low frequencies. However, the ratio was large enough at high frequencies for the energy regenerated to supplement energy required by the control actuator [18].

Although Suda and Shiiba gave a comprehensive analysis of both the efficiency and the isolation response of the system, there were several aspects of their analysis that required further investigation. One aspect was the use of the DC Motor. Suda and Shiiba conceded that the rotating DC Motor was not the most suitable device for use in vehicle suspension systems. However, the research paper did not document any attempts to determine what factors would constitute a more suitable design. Suda and Shiiba also demonstrated some of the trade-offs involved in the design of a regenerative damping system. However, there was no documentation on the optimisation of design parameters, such as external resistance and gear ratio.

One limitation of this analysis was due to the employed regeneration model. The energy regenerated was assumed to be 'consumed' in an external resistance. This model may be adequate if the energy is used directly, such as for lighting or heating. However, for a system in which energy is stored, the associated 'potential barrier' of the battery or capacitor was not investigated.
Okada and Harada [19] proposed an energy regenerative vibration damper for use in active suspension systems. A linear DC electromagnetic motor was used for the damper. Figure 2.4 shows a 'double-voltage' charging circuit used to regenerate electrical energy during high speed motion of the actuator. For low speed actuator motion, the actuator voltage was lower than that of the storage batteries. The disadvantage of this was that no regeneration and no damping force was produced for low speed motion. Therefore, passive damping was used to overcome this problem by switching the actuator terminal to a resistance for low actuator velocities.

![Figure 2.4 'Double Voltage' Charging Circuit [19].](image)

A simulation was undertaken by Okada and Harada [19] to determine the performance of the system. It was revealed that, for the low frequency range, the isolation response was close to a purely active system, but the system consumed more energy than it regenerated. For the high frequency range, the isolation response was close to the passive case and some energy was regenerated. It was also concluded that, for adequate damping and improved energy regeneration, a larger damping ratio would be required.

An experimental analysis was undertaken by Okada and Harada [19], with measurements taken from the decay response of an input transient, step function. This analysis showed reasonable agreement with the simulation, although no energy was regenerated for the experimental analysis. However, according to the simulation analysis it was expected that energy regeneration would occur for higher excitation frequencies.

Okada et al. [20] continued the investigations by Okada and Harada [19]. In this investigation, Okada et al. introduced an electric resonant circuit to improve the energy
regenerative efficiency and damping response. This circuit is shown in Figure 2.5. An optimal tuning condition for the resonant circuit was analytically derived to maximise the damping performance.

![One DOF Regenerative Vibration System with Resonant Circuit](image)

**Figure 2.5** One DOF Regenerative Vibration System with Resonant Circuit [20].

A simulation of the suspension system revealed that the system provided adequate frequency response when the optimal tuning condition was satisfied. An experimental investigation was also undertaken, with measurements taken from the decay response of an input transient, step function. The results differed from the theoretical analysis, mainly due to a finite actuator resistance, which resulted in lower damping than expected. For the system with the resonant circuit, the isolation response was improved in comparison to the system without the resonant circuit.

There were some limitations of the analysis by Okada et al. [20]. For simplicity in deriving the analytical model, the analysis did not include the voltage potential of the regenerative storage device. Also, the tuning of the resonant circuit was only undertaken with respect to the suspension isolation response and not with respect to regeneration efficiency. Furthermore, both simulation and experimental investigations did not give any quantitative measurements of the regenerated energy.

Okada et al. [21] also continued the investigations by Okada and Harada [19]. In this investigation the actuator terminals were switched to a resistance for low actuator velocities, for the same reason put forward by Okada and Harada [19]. The experimental results were undertaken using an input sinusoidal function, with different forcing
magnitudes. It was shown that the equivalent damping ratio for a system that included passive control was greater than the system without passive control ($c_{eq}=1.55$ compared to $c_{eq}^{\prime}=0.2443$ for the passive control design). Energy was regenerated near the suspension resonant frequency, and was similar for both regeneration designs.

Suda et al. [22], also investigated regenerative damping for the purpose of self-powered active vibration control. The damping system utilised two electromagnetic DC linear motors in a two degree-of-freedom suspension system. One linear motor, called the 'regenerative damper', was placed in the primary suspension, and could transfer vibration energy to a storage capacitor using relay switches. The damper could also be connected to a short-circuit, in which it would act as an ordinary passive damper. The regeneration circuit is shown in Figure 2.6.

The other linear motor, called the 'actuator' was placed in the secondary suspension. The actuator functioned as a passive damper, or using the energy stored in the capacitor, provided active control. The vibration control system is shown in Figure 2.7.
Experiments undertaken by Suda et al. [22] revealed that the damping response of the regenerative damper was reduced when regenerating energy. However, to prevent an unwanted reduction in the suspension damping, the electromagnetic damping coefficient was controlled such that it was always at least greater than 50 per cent of the passive damping coefficient.

An experimental analysis was undertaken by Suda et al. [22] to analyse the performance of the system. A random vibration response was used as an input disturbance, and either semi-active or active control was used to control damping and regeneration. In the active control scheme, the actuator energy was supplied by the storage capacitor. In the semi-active control scheme, a short circuit was connected to the actuator when the required damping coefficient was larger than the maximum attainable damping coefficient.

The results indicated that the performance of the system was adequate. The output force of the actuator damper approximately followed the force required for active control. It was found that the average energy regeneration efficiency was 15.0 per cent, and the damper produced 85.0 per cent of the maximum damping coefficient. It was also shown that the isolation performance of the self-powered active vibration control was better than the semi-active or passive control system. This is shown in Figure 2.8, with the thick curve representing the self-powered damping situation.

Suda et al. [23] used a similar system to their earlier investigation [22]. In this later study, the self-powered active suspension was incorporated into the cabin of a heavy-duty truck. The regenerative damper (similar to the damper shown in Figure 2.6) was installed in the truck's chassis suspension and energy was regenerated into a storage capacitor. The electromagnetic actuator was installed in the cabin suspension and provided active control, using energy from the storage capacitor. It was expected that the vibration energy of the chassis would be larger than that of the cabin, because the mass of a typical truck chassis was greater than that of the cabin.
A simulation of the system was undertaken with the truck travelling at a velocity of 10 (m/s), on a road surface classified as poor. It was shown that the actuator had the ability to generate a damping force against the vertical velocity of the cabin, which is not possible with passive or semi-active suspension. It was shown that, with the self-powered damping, the vertical velocity of the cabin was reduced compared to both the passive and semi-active systems.

To confirm the isolation performance of the system, Suda et al. [23] also undertook an experimental investigation. A two degree-of-freedom model was used, with the regenerative damper and actuator placed in the primary and secondary suspension, respectively. Random vibration was used as an input. It was shown that the isolation performance, compared with passive and semi-active suspension, was improved.

In a further investigation of regenerative vibration systems, a generalised actuator model was theoretically analysed by Heinzmann et al. [24], using a bond-graph representation. The objective was to optimise energy regeneration in a number of dynamic, regenerative systems. It was found that, if the control objective was to maximise energy regeneration, the motion was determined by the damping that allowed the actuator to operate at maximum efficiency. For a linear system, it was shown that the maximum efficiency
occurred for an actuator acting as a linear viscous damper [24]. Although model simulations and experimental testing showed good qualitative agreement, the experimental energy efficiency of an electromagnetic regenerative damper was lower than predicted. It was, therefore, stated that the source of additional losses should be further investigated [24].

The review of electromagnetic damping research, documented in this section highlighted several significant findings in relation to this Doctoral research program. Fodor and Redfield [14] stated the importance of maximising regenerative efficiency. In this research an 'impedance matching', mechanical damper, called a VLT, was theoretically analysed. However, the use of impedance matching, to maximise the regeneration efficiency of electromagnetic regenerative systems, was not documented. It was believed that an impedance matching design could also restrict the problems observed by Okada and Harada [19] with respect to the lack of damping and regeneration for low-speed damper velocities for a passive regeneration circuit. This Doctoral research, therefore, sought to analyse the regeneration system, with the objective of improving the regeneration efficiency and damping response.

The research documented in this section revealed that, although primarily investigated for their use as semi-active dampers, electromagnetic devices were able to provide damping as well as regenerative vibrational energy. However, the research documented by Suda and Shiiba [18] indicated that the rotating DC Motor was not the most suitable device for use in vehicle suspension systems. The research paper did not document any attempts to determine what factors would constitute a more suitable design. A focus of this Doctoral research was, therefore, to analyse these factors in order to improve the design of an electromagnetic regenerative damper.

The research by Suda and Shiiba [18] also demonstrated some of the trade-offs involved in the design of the overall regenerative damping system. However, there was no documentation on the optimisation of design parameters, such as external resistance and gear-ratio. In this research, therefore, an objective was to improve the effectiveness of
regenerative damper design by analysing, in further detail, the trade-offs associated with electromagnetic regenerative damping systems.

One issue raised from the review of regenerative vibration damping research, in Section 2.2, was that there had been a small number of investigations specifically analysing regenerative damping systems in their own right. The previous analysis concentrated on analysing regenerative systems for the purpose of improving the performance of semi-active or active suspension systems. The research documented in this thesis, however, specifically investigated regenerative suspension as a method of improving the energy efficiency of vehicle systems, without reference to improved vehicle dynamics through the use of semi-active, or active suspension.

Another issue raised from this section was that Fodor and Redfield [14] emphasised the importance of impedance matching between the source and the storage device to maximise the regeneration efficiency. However, the investigations of electromagnetic regenerative dampers did not analyse the use of impedance matching as a method of improving regenerative efficiency. A major objective of this research was to determine the performance of regenerative vehicle suspension, and it was believed that this would be largely influenced by the efficiency of the overall system. Therefore, the research documented in this thesis analysed the issue of impedance matching, as a method of improving the performance of regenerative damping. This analysis is presented in Chapter 5.

### 2.3. Electromagnetic Damping

In this section, an analysis of research undertaken in the area of electromagnetic damping is provided. Although the previous section documented research undertaken using electromagnetic dampers, these systems were analysed specifically for their regenerative damping properties. This section differs, in that the aim of the reviewed
research, was to analyse the design of the electromagnetic damper itself. The aim of this section is to document the state of knowledge in the field of electromagnetic damper design (as it existed at the time of this research) and observe how the design of these devices relate to electromagnetic regenerative dampers used in vehicle suspension systems.

Kamopp [25] examined permanent magnet linear dampers, specifically for use in semi-active suspension systems. The analysis identified several advantages of using these devices as linear actuators in semi-active suspension systems. One advantage was that the damping coefficient could be rapidly varied by changing the external resistance connected to the damper. The devices had low static friction, and they were inherently linear devices. Kamopp also recognised some limitations of electromagnetic linear dampers for use in suspension systems. These limitations included the finite coil resistance, device mass and the achievable magnetic field produced by permanent magnets.

The analysis evaluated how the device mass affected the damping force of the linear damper. The mechanical time constant of the coil mass was evaluated to determine the physical limitations of such a system. The time constant represented the fastest rate of decay of a coil that was decelerated at its maximum rate. It was shown that electromechanical dampers could only be effective as long as the vibration period was long in comparison to the mechanical time constant. The results of this analysis indicated that reasonable damping was achievable with lightweight coils, as the typical coil time constant was much lower than the period of vibration in typical vehicle suspension.

Kamopp [25] proceeded to evaluate the electrical time constant of the electromagnetic damper due to the internal coil inductance and resistance. It was concluded that the response delay would not be a limitation at the typical vibration frequencies encountered in vehicle suspensions. Kamopp revealed that when using high-energy magnetic material in linear dampers, only a short magnet length was needed and, as such, the weight and size of the complete damper were mainly functions of the iron pole pieces.
Therefore, the effective use of such magnets was only achieved by designs which minimised or eliminated the use of pole pieces. A toroidal design, which is shown in Figure 2.9, was suggested as a method of increasing the efficiency of permanent magnet linear dampers. It was designed in such a way as to minimise the use of the magnetic pole pieces. From a theoretical analysis of the damper mass, Karnopp concluded that the design had the potential to greatly reduce the size and weight of voice-coil type dampers.

Two limitations were given for the device shown in Figure 2.9. Firstly, a long magnetic field had to be generated, and was only partially utilised. Secondly, because of the parallel connection of the conductors, a very low output voltage, and a high current was produced. The use of several, stationary current-collecting brushes was suggested as a possible solution. The brushes would be located only in the region of magnetic field, and would only collect current in the area of the magnetic field. This design is shown in Figure 2.10. The brushes could also be wired in such a way as to link groups of conductors in the air-gap in series, which would increase the output voltage.
Ryba [26] undertook an analysis of a vehicle vibration isolation. In order to design an optimal force generator, both frictional electromagnetic devices and electric motors were possible alternatives to replace a conventional damper. It was proposed that linear electromagnetic devices were not readily available for this type of application, and should be designed for this purpose. Ryba analysed the linear motor suggested by Karnopp [25] (reviewed at the start of this section). However, the preliminary calculations led to the result that the mass of this linear motor would be quite considerable for the application of a sprung seat [26]. Therefore, the linear electromagnetic damper was not analysed further by Ryba.

A rotating motor was suggested by Ryba as a method of reducing the damper mass. A rack-and-pinion drive converted linear to rotating motion and provided a form of mechanical amplification. This amplification led to a lower device mass. However, Ryba [26] suggested that the use of a rotating mass, as a controlled force element, may cause problems due to a finite amount of high frequency transmission. A one degree-of-freedom suspension model was used to demonstrate this. This model is shown in Figure 2.11.
The transmission response is shown in Figure 2.12, in which the isolation response was defined by,

\[ H^2 = \left| \frac{X}{Y}(j\omega) \right|^2, \tag{2.1} \]

in which \( X \) was the sprung mass displacement, and \( Y \) was the input displacement, as shown in Figure 2.11. The response for the rotating damper (solid line), has a finite response for high frequencies \( (H_0) \), whereas, the linear damping element (dashed line) provided zero transmission at high frequencies.

Ryba [26] demonstrated that the use of an electromagnetic clutch led to a lower mass than an electric motor, which would therefore minimise the amount of high frequency transmission. Therefore, Ryba's analysis was continued with an electromagnetic clutch.
Semi-active damping allows for the use of an electromagnetic clutch as a force element, with the dissipation of vibrational energy as heat. However, it is not appropriate for use as a regenerative damper. The results given by Ryba reveal some details that affect the design of electromagnetic regenerative damping elements. The rotating damper had an advantage due to mechanical amplification, however, it produced a degraded vehicle isolation response. Ryba did not attempt to investigate under what circumstances rotating dampers could be used in vehicle suspension systems.

The investigation of the vibration isolation characteristics of an ideal, active and electromagnetic force generator was investigated by Su et al. [27]. The electromagnetic device was modelled using first-order dynamics. Although the emphasis of the investigation was for active vibration control, it was found that the dynamics of the electromagnetic force generator had a considerable influence on the isolation characteristics of the system [27].

In the context of this research, there are two main findings obtained from the analysis of electromagnetic dampers, given in this section (Section 2.3). Firstly, as demonstrated by Karnopp [25], it is important to analyse the topology and overall design of an electromagnetic device to optimise its performance. However, this analysis did not include the development of a generalised topology structure, for the purpose of determining the most appropriate electromagnetic damping system. The review of electromagnetic regenerative dampers (Section 2.2.3) also revealed that there has been only a limited analysis into the design of the most appropriate electromagnetic dampers, and no analysis utilising the device topology. It was, therefore, the objective of this Doctoral research to develop a generalised topology structure, with the purpose of determining the most appropriate form of electromagnetic device for regenerative damping applications. This analysis (presented in Chapter 4) therefore assists in the resolution of the project objective of determining the most suitable regenerative damping device.

Another finding from this portion of the review was that there were trade-offs between the use of linear and rotating electromagnetic devices for damping applications. The use
of rotating dampers allowed for mechanical amplification, via a rack-and-pinion type mechanism. However, as analysed for a one degree-of-freedom system, the rotating damper led to high-frequency transmission between the input disturbance and sprung mass. The reviewed research did not determine the limitations of using rotating dampers or investigate the use of rotating dampers in more detailed vehicle system models, which have two or more degrees of freedom.

2.4. Vehicle Suspension Energy Dissipation

2.4.1. Overview

At the time this research was conducted, there had been several attempts to analyse the mechanism of vehicle rolling resistance and energy dissipation in suspension dampers. The motivation for these investigations primarily stemmed from the need to find the sources, and quantify the amount of energy dissipation for the entire vehicle system. The results led to a greater understanding of how the resistance could be minimised to design more efficient vehicles. In relation to this investigation, the function of rolling resistance and energy dissipation in vehicle dampers plays a major role in the improvement of overall vehicle efficiency and, therefore, the performance of regenerative dampers.

2.4.2. Theoretical Investigations

Karnopp [28] undertook a theoretical analysis to determine the energy loss mechanisms of a suspension damper in a vehicle traversing an uneven roadway. Shown in Figure 2.13 is the theoretical, one degree-of-freedom (DOF) suspension model that was analysed. The suspension model was assumed to traverse a deterministic, periodic roadway, with either a sinusoidal or triangular profile.
Using the theoretical model, the average propulsive force was evaluated as a function of vehicle velocity. The response for a sinusoidal profile is shown in Figure 2.14. The response gave the normalised, average propulsive force ($\bar{F}$) verses the normalised vehicle velocity, $\beta$ (relative to the resonant velocity). The response was given for three reasonable values of the suspension damping ratio ($\zeta = 0.25, 0.50, 0.707$). It was shown that only a small propulsive force was required for speeds much less than the resonant velocity. For high vehicle velocities, the average propulsive force was proportional to both vehicle velocity and suspension damping coefficient. However, for a small damping ratio and a vehicle velocity near the resonant velocity, a large suspension motion was induced which required a large propulsive force.

Karnopp's research also investigated very 'stiff suspensions', in which a periodic loss of contact occurred between the tire and the road surface. This could happen for rough roads and large vehicle velocities. Karnopp concluded that the energy dissipation would be quite large for very 'stiff suspensions', and that the propulsive force could be much larger than air drag and rolling friction. However, the detailed findings of the stiff suspensions are not presented here as they are not deemed relevant to regenerative vehicle suspension systems for conventional vehicles.

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2 The resonant velocity occurred when the temporal frequency of the sinusoidal roadway profile equalled the natural suspension frequency.
Although the results from Karnopp's investigation gave an indication of the process of energy dissipation in vehicle suspension, the road profiles were periodic and would not be typical of a real situation. The analysis also utilised a one degree-of-freedom model which neglected tire dynamics, such as tire deflection. Also, the model was only valid for road profiles with a large wavelength relative to a typical tire radius. This is due to the 'enveloping' effect, in which small roadway irregularities are smoothed out due to the finite tire radius.

Velinsky and White [29] also employed a deterministic road roughness model to analyse energy dissipation in vehicle suspension due to road roughness. The model included both the tire and suspension system. It was represented by a four degree-of-freedom, rear axle suspension model. The energy dissipation was analysed by measuring the relative velocity across both the suspension damper and the damper representing the dynamics of the tire. The energy loss was analysed with respect to the parameters of tire pressure, vehicle speed, and road roughness. Figure 2.15 shows some results of this analysis.
It was shown that, for a sine wave input, the percentage of energy dissipation in the suspension damper, with respect to total energy dissipation in the tire-suspension system, was seen to decrease rapidly with increasing vehicle speed. For an input wavelength of 0.6 (m), and vehicle velocity of 5.55 (m/s), the suspension system contributed to approximately 95 per cent of the energy loss in the tire-suspension system. The tire became the dominant dissipative energy component for frequencies above 20 (Hz), or 13.89 (m/s). These results were expected as the suspension only responded to the low frequency excitations due to the relatively low spring rate in comparison to the tire system. Although the percentage of power dissipation between the tire and suspension damper was presented by Velinsky and White [29], the actual amount of energy dissipated in each element was not given.

![Graph showing tire contribution to system energy loss](image)

**Figure 2.15** Tire Contribution to System Energy Loss [29].

Velinsky and White also performed an experimental investigation of energy dissipation. The investigation measured and analysed the vertical acceleration of the tire-axle system in response to various types and roughness of road surfaces. The aim was to define road roughness in a quantitative sense, as it is directly linked to power dissipation in the tire-suspension system. On-road measurements of vertical acceleration were compared with a deterministic road roughness model. The deterministic model was devised using
randomly spaced ramp inputs to represent the road surface. This model is shown in Figure 2.16.

![Figure 2.16 Ramp Input Function [29].](image)

According to Velinsky and White, the deterministic model was a successful representation of an actual road profile as the predictions were of the same order, and exhibited the same trends as those experimentally attained for all test cases. The measured results found that the root-mean-square (RMS) vertical axle acceleration (which was a measure of the energy lost in the tire-suspension system) increased with increasing vehicle speed. The vertical axle acceleration also increased with an increase in tire pressure. The roads subjectively evaluated as 'rough' also exhibited increased RMS axle accelerations over those judged as "smooth". Velinsky and White also determined that rough roads led to an increase in rolling loss as high as 20 per cent in comparison to traversing smooth roads.

The deterministic road profile, shown in Figure 2.16, was defined using randomly spaced ramp inputs. The justification for using the deterministic model was that the "bump" length was in the same order of rough road disturbances such as pot holes. Other deterministic road profiles, such as semi-circular or rectangular disturbances, were not mentioned.

A detailed theoretical analysis of rolling resistance was undertaken by Segel and Lu [30]. The analysis examined the affect of road surface roughness on vehicular resistance to motion caused by energy loss in the tire and suspension system. The resistance to motion was evaluated using a theoretical, one-quarter, two degree-of-freedom vehicle model. This model was assumed to traverse an irregular road surface with a stationary random process describing the profile [30]. The tire system was modelled as an assembly of many independent, radial springs, with each spring behaving as a hysteretic elastic element. The suspension damper was modelled as an ideal linear viscous damper.
The vehicle dynamic model is shown in Figure 2.17, in which \( y_v, y_t \) and \( y_e \) were the sprung mass, unsprung mass and road surface elevations, respectively.

![Two DOF Quarter Car Model](image)

**Figure 2.17** Two DOF Quarter Car Model [30].

The total impeding force \((F_x)\), derived from the tire and suspension models, was given as a function of the spectral density of the random elevation of the roadway. Shown in Figure 2.18 is the vehicle resistance to motion as a function of the road surface condition. Shown in Figure 2.19 is the vehicle resistance to motion as a function of vehicle velocity for a 'worst case' road. \( F_{xs} \) and \( F_{xt} \) refer to the motion resistance from the vehicle suspension and vehicle tires, respectively. The 'roughness' level of the road surface was defined by a constant \((A)\).

![Motion Resistance as a Function of Road Roughness](image)

**Figure 2.18** Motion Resistance as a Function of Road Roughness [30].
The results indicated that there was a substantial increase in rolling resistance due to the suspension system when the surface condition of the roadway became sufficiently rough (classified as 'poor' or 'very poor'). Also, the resistance deriving from suspension loss was predicted to be substantially greater than the incremental energy lost in the tire [30]. The results from Figure 2.19 indicated that the suspension loss increased considerably in the range from a velocity of 2 (m/s) up to 10 (m/s) and, although still the dominant loss mechanism above 10 (m/s), the suspension loss became less significant, and the tire loss began to increase.

Segel and Lu [30] indicated that many of the assumptions made in the analysis could be violated when the road surface became exceedingly rough or the vehicle velocity was high. The vehicle model, analysed by Segel and Lu, included both the tire dynamics and the finite tire radius enveloping the small scale road irregularities. The road model used in the analysis was a random stochastic process, and was based on the measured elevation profiles of a large number of roads. This model therefore analytically represented a road surface to a high order of accuracy [30].

The research of Lu and Segel [31] continued their earlier investigation, documented in [30]. In the later investigation, an experiment was undertaken to determine the degree to which a linear analysis (as analysed in [30]) was a valid model of the road traversal process in terms of power dissipation. To achieve this, Lu and Segel compared a linear simulation, in which energy loss only occurred due to the stress-strain hysteresis losses
in the tire and viscous damper loss, with non-linear simulations in which energy loss also occurred due to loss of road-tire contact and the suspension limits being encountered.

The results from Lu and Segel [31] indicated that, for vehicle velocities up to 15 (m/s), there was a very good correlation between power dissipation predicted by a non-linear simulation and that predicted by a linear model. The results also indicated that a non-linear vehicle model traversing surfaces classified, under an ISO Standard (1972) [30], as 'average', or better, appeared to be more than adequately described by a linear model. These results were important as they suggested that the previous theoretical linear analysis given by Segel and Lu [30] could be related to a more realistic non-linear model, as long as the velocity was not too high and the road was not exceedingly rough.

Lu and Segel also demonstrated that the additional energy losses, attributable to the traversal of a typical primary road at a speed of 13.33 (m/s), were of the order of 10 per cent of the rolling energy loss occurring on an absolutely smooth surface.

Kamopp [32] theoretically analysed the power requirements of vehicle suspension systems, neglecting the effects of air and tire rolling resistance. To do this, an arbitrary suspension model was utilised. The vehicle model traversed an uneven roadway surface at a constant velocity. The results revealed that the average propulsion power was equal to the average power dissipated in the suspension [32]. This result was important in the context of this doctoral research because it can, therefore, be assumed that any energy regenerated from vehicle suspension (and put back into the vehicle system) will result in a direct reduction in the vehicle propulsion energy. This is important for applications such as electric vehicles, in which the reduction of vehicle propulsion energy provides a significant benefit to the vehicle system.

Kamopp [32] suggested that there were two cases in which the propulsion power could be made to diminish to zero. One case was for very stiff suspension. However, the vehicle velocity must remain low in this situation to prevent the vehicle wheel from leaving the road surface. The other case was for very soft suspension. In this case, the
suspension filtered out the effects of the surface unevenness while contributing little to the required vehicle propulsion power. However, this situation caused another design limitation, due to a large relative motion induced between the sprung mass and the wheel [33].

2.4.3. Experimental Investigations

As opposed to the theoretical analysis previously given, Browne and Hamburg [10] investigated the amount of energy dissipation in automotive shock absorbers using two independent experimental methods. The first method used the thermal change within a vehicle shock absorber to give a measure of the amount of energy dissipated while traversing an irregular road surface. This method was only moderately successful at predicting energy loss due to variable heat loss to the external environment in the field experiments. This method, therefore, only provided a lower bound on energy dissipation rates, and only provided gross trends.

A second, more direct, measure of energy dissipation was also undertaken. This method involved measuring the axial components of applied force and relative velocity between the two end attachments of a conventional shock absorber. The product of the instantaneous velocity and force measurements provided the instantaneous rate of energy dissipation. Measurements were recorded from two different test vehicles which were driven over different test tracks, or under normal traffic conditions on interstate highways and rural and urban main arteries.

The loss rates per vehicle damper, using this method, varied from an average of 3 to 57 (W) depending on the road surface. The typical energy dissipation rates were between 10 and 15 (W) per damper [10].

The experimental data recorded by Browne and Hamburg [10] also provided a strong indication of the influence of operating factors on the amount of energy that was dissipated in the vehicle damper. The power dissipation was shown to be linearly
dependent on the vehicle speed (within the experimental operating conditions). The results also suggested that an increase in power dissipation occurred with increasing road roughness. Energy loss in the suspension damper was also shown to increase with increasing tire inflation pressure. However, this change was insignificant in comparison to the increases of energy loss with vehicle velocity and road surface roughness. Browne and Hamburg [10] also found that shock absorber losses on relatively smooth surfaces, where the speed-roughness combination was insufficient to excite motions of the sprung mass, appeared insensitive to a change of vehicle mass.

Browne and Hamburg concluded that between 40 and 60 (W) of energy was dissipated in all four vehicle suspension dampers when travelling on good to average road surfaces.

From their experimental investigation, Browne and Hamburg [10] concluded that it was not practical to regenerate suspension energy for major vehicle power systems, as the regenerated power would have little effect on fuel economy. However, Browne and Hamburg concluded that the regeneration of this power may have a particular interest for special applications such as electric vehicles where fuel economy and vehicle range were important factors. Another application of suspension energy regeneration, suggested by Browne and Hamburg, included the storage of the energy for systems that only needed energy on an intermittent basis, such as windshield wipers and defrosters.

In relation to the present research program, this portion of the review highlighted several important findings. One limitation of the then current research was that, although there had apparently been theoretical estimates of rolling resistance due to energy loss in suspension systems, there has not been a theoretical analysis specifically investigating energy loss in vehicle suspension systems. Furthermore, there had not been a comparison between theoretical models and experimental measurements of suspension energy loss. A comparison could reveal any limitations with either the experimental or theoretical investigations. Therefore, as the determination of suspension energy loss was a large determinant of regenerative suspension performance, it was determined that such
an analysis should be performed within the context of this Doctoral research. This analysis is presented in Section 6.2.2.

2.5. Cost-Analysis of Vehicle Systems

A fundamental determinant of the performance of any energy recovery system was the relative cost. In this section, an analysis of research undertaken in the area of vehicle system cost analysis is presented. The objective of this analysis was to gain an understanding of how to evaluate the performance of a particular vehicle system.

Schuckert et al. [34] undertook a Life-Cycle Analysis (LCA) of electric and conventional vehicle systems to evaluate the overall environmental impact of vehicle systems. In this LCA study, material and energy flows for all system processes were investigated. The energy flows included material production, assembly, maintenance, disposal, transportation and operating energies. The scope of LCA began with the extraction of raw materials from the earth and ended with the disposal of wastes back to it. Within the context of automotive production environments of the 1950s to 1980s, such an analysis may have been excessive in as much as there was little regard for the disposal of redundant products. However, with the introduction of environmental accounting in the 1980s, this analysis became important.

An analysis was performed to evaluate the life-cycle energy components, as a function of vehicle weight, material use and material recycling, for an electric vehicle (EV) and a functionally equivalent internal combustion engine vehicle (ICEV). The results indicated that, although the EV was 227 (kg) heavier than the ICEV, it had 25 per cent less life cycle energy (551.4 GJ for the EV compared to 729.1 GJ). This was mainly due to the greater operating efficiency of the EV. It was also shown that the largest energy reduction could be achieved by reducing the vehicle operating energy, or the efficiency of energy required to propel the vehicle over its operating lifetime. It was also shown
that material use affected the overall life cycle energy of a vehicle system. For example, if the steel in the ICEV was replaced by aluminium, the weight saving improvement in life cycle energy was 7.5 per cent. If 50 per cent recycled aluminium was used, the benefit became 12 per cent.

Although Schuckert et al. [34] stated that the results of the LCA approximated the life cycle energy for vehicles, they conceded that the results were not exact. The main sources of error were attributed to the variations in production, assembly, and operating energies.

Dhillon [35] investigated life-cycle analysis, with an emphasis towards large projects, such as the budgeting and costing for the acquisition of major defence systems for the military, or costing of large organisation infrastructure. The investigation by Dhillon did, however, include important information that applied to any general cost analysis and, therefore, could be related to the performance analysis of a vehicle system such as regenerative damping.

The investigation by Dhillon [35] included the procedure for the development of a life-cycle cost model. A discussion of a "break-even" analysis was presented, in which, at the break-even point, total revenue equalled total cost. The analysis included the development of a generalised model to determine what conditions must be met in order to break-even. An example was given for the determination of how many particular items, for a manufacturing system, must have been produced before the break-even point was reached. In relation to this research, the break-even analysis is directly related to the feasibility of a particular system, in which the revenue was related to the benefits attributed to the system, and costs were the losses associated with the system.

In relation to the specific information required for performing life cycle costing studies, Dhillon [35] included the following parameters:

- Useful operational life of the item in years.
- Development costs.
• Annual maintenance cost of the item.
• Salvage value or disposal cost of the item.
• Procurement cost of the item.
• Transportation (delivery) and installation costs.
• Annual operating cost of the item. Which includes:
  • Energy cost.
  • Cost of supplies.
  • Labor cost.
  • Cost of materials.

Dhillon revealed several aspects that related to the performance analysis undertaken in this research. For example, for effective life cycle cost estimates of products or items, Dhillon stated that the availability of reliable historical cost data on similar items or products was vital. Without data sources of adequate quality, the accuracy of the cost estimate was sacrificed.

The investigation by Dhillon also analysed cost-indexes. These indexes related to a dimensionless number used for the purpose of adjusting an item's cost from one period to another. The reasons for an item's cost to change continuously included inflation, changing technology, and changing availability of materials and labor. These issues are important for a technological system development methodology, such as for a regenerative damper, that may rely on improving technology, materials and labor.

Sassone and Schaffer [36] described several criteria for evaluating the worth of a particular project, which may also apply to a regenerative vehicle system. The 'Net Average Rate of Return' (NARR) was defined as the sum of the net benefits over the life of the project divided by the number of years over which such benefits were incurred. This criteria had shortcomings for the purposes of comparing two projects with differing life-cycles, however [36]. Another criteria was called the 'Cutoff Period'. According to this criteria, a specific time was chosen in the future, and a project was acceptable only if it covered all its costs by that time.
An analysis of cost-analysis, with respect to the production of articles or rendering services was undertaken by DeGarmo et al., [37]. The cost-analysis was based on an economic (monetary) measure, although, the methodology was applicable to many systems such as regenerative dampers. The costs were attributed to materials, labor and overheads, and a 'breakeven chart' was described that gave a measure of the overall cost of the system. In this analysis, it was shown, using a linear 'price and demand' relationship that, for an increase in the output units for a particular item, the overall benefit increased (increase in profit).

The analysis by DeGarmo et al. [37] also investigated the effect of material selection on the overall cost of the system. It was stated that, when making selections between two or more materials, the proper cost criterion must be used. For example, if selecting between steel and aluminium, both rigidity and strength might be of importance. It would, therefore, be necessary to consider both these parameters to arrive at a proper selection decision. It was also stated, however, that the economic selection between materials depended on other cost parameters such as production and transportation.

An investigation of cost-analysis by Humphreys and Katell [38], indicated that, depending upon the particular process, material cost could constitute a major portion of the overall system costs. For this reason, it was suggested that a complete list of all raw materials needed to be developed for the system cost-analysis. Also, the material unit cost, available material sources, as well as the quantity and quality of the materials should be determined for the purposes of developing an accurate cost-analysis model [38].

The overall findings of this review indicated that a cost-analysis of a regenerative vehicle system will be able to provide a measure of the system performance. Also, each particular system under investigation will have its own cost-analysis criteria [36-38] and, therefore, a detailed knowledge of the system under investigation is needed for the identification of the costs and benefits [36]. Another finding was that the determination of the costs and benefits, for such a regenerative vehicle system, must account for the overall energy flows in the system. This will include the energy associated with the
regeneration process, and all energy associated with materials, production and system operation [34].

2.6. Patent Applications

In order to conduct a broad ranging study of previously conducted work in the field, it was necessary to examine relevant patents in the field. This section reviews some of the patent applications that were made in the area of regenerative suspension systems. The aim was to find any commercially orientated regenerative research that had not been documented in journal articles.

The review of patent articles concentrated on several key areas, which included patents (classified under the International Patent Classification) related to:

- Vehicle suspension arrangements having dampers accumulating utilisable energy (classified as B 60 G 13/14).
- Vehicle suspension combined with energy-absorbing means (classified as B 60 G 13/18).
- Mechanisms recovering energy derived from swinging, rolling, pitching, or like movements, e.g. from the vibrations of a machine (classified as F 03 G 7/08), and
- The suppression of vibrations in systems using electromagnetic means (classified as F 16 F 15/03).

In 1977, Starbard [39] suggested a device called a 'shock absorber generator' which is shown in Figure 2.20. The device consisted of a pair of electric alternators connected to a movable shaft via a rack and pinion, and a one-way clutch mechanism. The movement of the vehicle under-carriage drove the alternators which then produced electricity. The alternators also provided sufficient drag to produce a shock-absorbing action. The rack and pinion mechanism converted the linear motion of the suspension into rotating motion to drive the alternators. The gears provided a relatively high speed drive for the
alternators, and the clutch mechanisms enabled the alternators to only rotate in one direction.

![Figure 2.20 Shock Absorber Generator [39].](image)

Although the 'shock absorber generator' would behave as a regenerative suspension element, there were several inadequacies of the design which were not described in the patent application. Firstly, there was no mention of the effect of the device on vehicle dynamics. The device shown, although no dimensions were given, had elements such as gears and alternators that would have had a reasonably large mass. This mass, together with the large gear ratio would have provided a large rotating inertia. The rotating inertia may have affected the vehicle dynamics, as well as producing large forces on the gear and clutch mechanisms.

The alternator regeneration efficiency or damping response was not given. Also, the clutch mechanism did not appear to serve an important function. The alternators, although rotating in only one direction, would still have become stationary when the rack was stationary. Therefore, the clutch mechanism would not have produced any advantages with respect to a reduction in mechanical wear due to stopping and starting the alternator rotation.
In 1990, a proposal by Gaseidnes [12], suggested some energy regeneration devices with several objectives. To transform energy from vibrating form to another more useful form of energy, to use the energy to regulate the vehicle level, and to design a device that could store the converted energy for later use. Three different suggestions were proposed for the energy transformer. A pump that transformed movement energy to pressure energy in an hydraulic or pneumatic system, an electrical generator which had linear or rotating movements, or a heat pump powered by heat produced by friction.

Gaseidnes [12] suggested the inclusion of a turbine in the path of the moving oil or gas, in a conventional damping shock absorber. The turbine would drive an electrical generator or hydraulic or pneumatic pump while providing a restrictive damping force. The force would be provided instead of the openings in a conventional damper piston. The author also suggested a membrane pump as a means of energy conversion.

Suggestions were given for the application of regenerative damping in relation to vibrating machinery. Energy for regeneration could be produced by modifying the clamping equipment used for motion devices such as combustion engines that vibrate and lose energy. Proposals included the addition of elastic pillows filled with air or liquid that is forced through a one-way valve system, which would then drive a turbine.

Takubo [40] suggested a regenerative energy device similar to that by Gaseidnes [12]. Fluid pressure was produced in a vehicle damper due to the vertical motion of the vehicle body. A 'fluid pressure motor' then drove a generator to produce electrical energy. The main difference between this design, and Gaseidnes' suggestion, was that only one generator was necessary for all four damping cylinders. The design is shown in Figure 2.21.

The outputs of all four vehicle dampers entered into one centrally located tank via a one-way valve, and the generator was fed off the central tank. The obvious advantage of this system was that only one, instead of four generators were required.
A major difficulty with undertaking a systematic review of patent applications was that there was no supporting evidence for the particular designs, and little information given as to why the authors had made particular design decisions. Furthermore, there was no information given as to the successful practical implementation of the designs. If this information was given, the understanding of the design performance would have been greatly enhanced. It is self-evident, however, that the purpose of a particular patent application is for the protection of new inventions and innovative ideas and, hence, patents will not generally provide a sufficiently detailed design description, or provide answers as to why a researcher or inventor has pursued a particular design path.

Although not contributing directly to the technical content of the present research program, the patent articles gave an important insight into the current ideas, and methodologies proposed in the area of regenerative damping research. The patent articles reviewed contribute to the overall thesis methodology by giving a broader perspective of the approach towards integrating regenerative dampers for vehicle suspension systems.
2.7. **Conclusions**

Following the review of previous research in the field of regenerative damping, vehicle systems, and electromagnetic devices, a number of issues were resolved. However, in order to achieve the overall objectives in this research program, the review also indicated that a number of areas required further investigation.

As stated in the introduction, the overall objective of this Doctoral research was to analyse regenerative vibration damping, and assess the viability of such a system for use in vehicle systems. One major finding of this literature review was that there had not been an overall performance analysis of regenerative damping for use in vehicle systems. It was, therefore, an objective of this research to investigate the process of regenerative damping, with a particular emphasis towards developing a methodology to determine the performance for use in a vehicle system.

The analysis presented in Section 2.2, reviewed several regenerative damping systems. It was found that the primary action of a regenerative damping element was to transfer vibration energy to an energy storing device as efficiently as possible while maintaining acceptable vibration control. The importance of impedance matching, or meeting the 'barrier potential' of the storage element was emphasised by Fodor and Redfield [14] and Mahajan and Redfield [15].

A number of investigations utilised electromagnetic devices as regenerative vibration dampers [18-23]. The objective of these investigations was to regenerate the vibrational energy and to improve the dynamic characteristics of the particular system. Although these investigations showed the ability for energy regeneration as well as the improvement of the system dynamic response, there were several limitations to the investigations. The most notable of these was the 'barrier potential' issue, previously referred to by Fodor and Redfield [14]. The electrical regeneration devices used either diodes or relays for energy transference to the storage device (see Figure 2.4 and Figure...
2.6). The disadvantage of these regeneration circuits was that they were not designed to provide impedance matching between the electromagnetic damper and the storage device and, therefore, did not maximise regeneration efficiency.

A further limitation of these devices, also caused by the 'barrier potential' issue, was the effect on the device damping response. When the electromagnetic device potential was lower than the storage device potential no current flowed in the circuit. This resulted in no force (and, therefore, no damping) produced by the electromagnetic device. It was, therefore, the objective of this Doctoral research to investigate the 'barrier potential' issue in relation to electromagnetic regenerative dampers; both with respect to maximising regeneration efficiency and maintaining adequate damping forces.

The review of electromagnetic regenerative vibration damping articles in Section 2.2 was unable to uncover a published analysis investigating the most appropriate form of electromagnetic device for use as a regenerative damper. For instance, Suda and Shiiba [18] conceded that the rotating DC Motor was not the most suitable device for use in vehicle suspension systems. However, the research paper did not document any attempts to determine what factors would constitute a more suitable design. With respect to achieving the overall project objectives in this research project, the optimisation and analysis of the most appropriate form of electromagnetic regenerative system, was then a research area that needed to be investigated in order to make a contribution to knowledge.

The review in Section 2.3 revealed the importance of the design of electromagnetic devices with respect to the device mass and effects on the vehicle system dynamic response. It was shown that the weight and size of a complete electromagnetic damper were primarily functions of the iron pole-pieces. The minimisation of damper mass in vehicle suspension was, therefore, an important design consideration. Hence, an effective electromagnetic regenerative damper design would only be achieved by minimising or eliminating the use of the pole pieces [25]. The electromagnetic regenerative dampers previously reviewed in Section 2.2 did not take these factors into consideration. It was, therefore, a further objective of this Doctoral research to
investigate the design of electromagnetic regenerative dampers with respect to their integration in a vehicle system. This analysis is presented in Chapter 4, with an analysis into the design of an electromagnetic regenerative damper.

It was also found that, although rotating dampers had an advantage of mechanical amplification, they could affect the dynamics of the vehicle system [26]. Another objective of the Doctoral research was, therefore, to investigate the use of rotating electromagnetic devices in vehicle suspension. This analysis is presented in Chapter 6, with an analysis of the integration of rotating electromagnetic regenerative dampers in vehicle systems.

In Section 2.4, Vehicle Suspension Energy Dissipation, a number of theoretical and experimental investigations were undertaken to analyse the process of energy dissipation, and rolling resistance of vehicle suspension systems. The experimental findings indicated that between 40 and 60 (W) of energy was dissipated by all four dampers of a vehicle suspension system for vehicle travelling on good to average road surfaces. It was also suggested that the regeneration of the dissipated suspension energy could have a particular interest for special applications such as electric vehicles in which fuel economy and vehicle range were important factors [10]. The theoretical results, however, only analysed the process of rolling resistance in vehicle suspension systems, and did not directly analyse power dissipation. As such, another objective of this research was to compare power dissipation rates between experimental and theoretical models, to highlight any inconsistencies between the investigations. This analysis is undertaken in Section 6.2.

The review in Section 2.5 investigated previous investigations in the area of cost-analysis. Although, there were only a limited number of investigations specifically related to vehicle systems, the analysis revealed several important aspects of developing a cost-analysis model for a vehicle system. Schuckert et al. [34] analysed a vehicle system life-cycle analysis, and gave several important findings in relation to the overall energy flows in a vehicle system. Important aspects of the life-cycle analysis included the material production, assembly, maintenance, disposal, transportation and operating
energies of the vehicle system. An example of the life-cycle analysis of a vehicle system was given, and revealed that the operating efficiency and material use were major determinants of life-cycle energy. This investigation also revealed that factors such as the useful operational life of the item, development costs, maintenance costs and disposal costs all should be considered when developing a cost-analysis model for a particular system. It was, therefore, the purpose of this research program to develop a vehicle cost-analysis model, using the factors identified in this review, for the purpose of determining the performance of regenerative vehicle suspension and achieve the overall thesis objectives.

The patent review, detailed in Section 2.6, revealed other forms of regenerative vibration damping designs. However, due to the lack of experimental evidence, and the limited design information, the patent applications did not provide a large amount of information for use in this research thesis. The review did provide an overview of different methodologies, and technologies proposed for regenerative damping systems, however.
"...the two operations of our understanding, intuition and deduction, on which alone we have said we must rely in the acquisition of knowledge."\(^1\)

René Descartes

3.1. Overview

This Chapter gives an introduction into the overall principles of energy regenerative vibration damping. The purpose is to establish a foundation for research documented throughout the thesis by discussing, in general terms, the operation and principles of a regenerative damping system.

There were several examples of regenerative damping systems investigated by other researchers. Examples included a mechanical regenerative system such as the Variable Linear Transmission (VLT) proposed by Fodor and Redfield [14] in Section 2.2.2, and electromagnetic regenerative systems investigated by Suda and Shiiba [18] or Okada and Harada [19] in Section 2.2.3. However, during the course of the review there was no published literature recovered on a generalised regenerative system, or an analysis of what type of regenerative system had the most potential for this application. An objective of this chapter, therefore, is to present an analysis of a generalised regenerative damping system and define the operational requirements and operation of the system as they emerged from the research. From this analysis, it was possible to propose a form of device that had the most promise as a regenerative damper. The results of this analysis led on to investigations that analysed specific aspects of the regenerative system which are documented in later chapters.

In this chapter, Section 3.2 gives a brief overview of the requirements of the regenerative damping system investigated in this research program. This section is referred to throughout the thesis, as it states the more important aspects of the design of the regenerative vibration damping system. Section 3.3 and Section 3.4 discuss the process of regenerative damping with respect to the energy flows in the damping system, and the energy conversion process, respectively. Section 3.5 discusses the process of energy conversion to electrical energy and the reasons for an interest in conversion to electrical energy in this project. Also presented is a discussion of the alternative processes and technologies available for a regenerative damper for the conversion from vibration energy to electrical energy.
3.2. Regenerative Damping Requirements

The purpose of this section is to put forward some of the main requirements for the regenerative system investigated in this research. These requirements served to focus the research effort.

There were several design considerations required for the regenerative vibration system analysed in this research. As an illustration, Fodor and Redfield [14] stated that "...the primary action of a regenerative damping system is to transfer vibrational energy to an energy storage device as efficiently as possible...while maintaining acceptable vibration control". Sharp and Crolla [41] also proposed several design considerations. These considerations included:

- Capital cost,
- Space requirements,
- Component weight, and
- Noise transmission and generation.

An extension to these requirements for the regenerative damping system analysed in this thesis is presented in the following list:

(i) Damping Response

The damping characteristic of the regenerative damper should follow that of a viscous damper. This requirement can be compared to the "acceptable vibration control" stated by Fodor and Redfield [14]. Two factors contribute to the requirement of acceptable vibration control. Firstly, the response of the damper must reasonably follow the dynamic characteristics of a conventional viscous damper. A conventional viscous damper exerts a force with a magnitude
proportional to the relative velocity across the damper [42]. This is illustrated in Figure 3.1, which shows an ideal viscous damper, and the relationship between the relative damper velocity ($V_{\text{REL}}$), and damper force ($F_D$). The constant of proportionality ($C$), is the viscous damping coefficient.

![Figure 3.1 Viscous Damper Characteristics.](image)

In many practical situations, however, due to the inherent non-linearity of the particular damping system, the relationship between damper force and velocity may not be exactly linear. In fact, for many vehicle damping applications, non-linearity may be considered beneficial, such as for producing damping forces in the rebound phase (when the damper rod moves out of the damper body) larger than the compression phase (when the damper rod moves into the damper body). The 'bilinearity' effect is used to optimise stability and passenger comfort [43]. However, the requirements of the regenerative damper in this research is for it to behave as much like a viscous damper as possible. This objective was also adopted by Fodor and Redfield [14], in an investigation of regenerative damper requirements.

(ii) Robust Performance

The damping performance of the overall system must be robust. Herein, robustness implies that it must be able to maintain adequate damping over an extended period. In applications such as vehicle suspension damping, it is critical that vibration control be maintained. The result of a total loss, or reduction in damping force may result in system damage or perhaps human injury for the applications that this particular system is designed.
(iii) Energy Conversion Efficiency

The energy conversion efficiency between the vibrational and regenerated energy should be maximised. This is to maximise the performance of the regenerative damping system.

(iv) Damping Force

The damping coefficient of the regenerative damper should be large enough to provide adequate damping for applications such as vehicle suspension or vibrating machinery. This is because these applications are believed to be the main potential beneficiaries of regenerative damping. For vehicle suspension systems, this will typically be in the order of 100 to 1000 (Ns/m).

(v) Input Disturbance

The overall system must be designed for random, non-deterministic and bi-directional mechanical vibration input. This is due to the nature of the vibrating systems that the regenerative system is designed for. Applications such as vehicle suspension, or vibrating machinery will have this form of random input disturbance.

The following sections analyse, in more detail, the requirements and operation of the regenerative damper with respect to vibration damping and energy regeneration.
3.3. *Regenerative Damping - An Energy Flow Issue*

Two main operations of a regenerative damper are to maintain adequate damping and to provide energy regeneration. The relationship between these operations and energy flow are outlined in this section.

The inherent nature of a viscous damper, as a non-conservative element, is that energy is consumed within the damper. In a conventional viscous damper, this generally results in the vibration energy dissipated to the external environment. A conventional hydraulic damper has a cylinder filled with a viscous fluid and a piston with holes or other passages by which the fluid can flow from one side of the piston to the other. The mechanical energy is converted into heat due to the movement of the damping fluid through the damper.

The regenerative damper performs the same damping process as the conventional damper, however, the energy process is modified. Instead of all of the mechanical energy being converted to heat, a portion of this energy is converted into a more useful form of energy that can be used for other applications. This is illustrated in Figure 3.2.

![Figure 3.2 Regenerative Damping Energy Conversion.](image)

It can be seen from Figure 3.2, that there is an "energy flow" issue involved with the design of a regenerative damping system. Also, the damping process is affected by
energy flow within the regenerative vibration damping system. The damping refers to the relationship between the force produced by, and the relative velocity across the particular damping element. However, the product of the instantaneous force and velocity is a measure of the power dissipated within the damper.

Referring to Figure 3.1, for a conventional viscous damper the power dissipated \( P_{\text{Diss}} \), is given by:

\[
P_{\text{Diss}} = F \dot{X} [\text{W}],
\]

\[
P_{\text{Diss}} = C \ddot{X}^2 [\text{W}],
\]

where: \( C = \) damping coefficient, \( \dot{X} = \) relative damper velocity, and \( F_D = \) damper force.

Equation 3.2 reveals that energy is inherently dissipated in a conventional viscous damper due to the actual damping process.

### 3.4. Regenerative Damping - An Energy Conversion Issue

There are five kinds, or forms, of energy; mechanical, electrical, chemical, photon and heat [44]. In regenerative vibration damping, the mechanical energy of the vibrating system is converted to a more useful form of energy. Table 3.1 presents a summary of energy conversion phenomena and technologies in which mechanical energy is the primary source [44]. For example, there are three main methods to convert mechanical energy to heat energy; via friction, a collision process, or through the use of metalhydride reactions.
Mechanical | Electrical | Chemical | Photon | Heat
---|---|---|---|---
• Torque Converter<br>• Fly wheel | • Generator<br>• Piezo<br>• Electricity<br>• M.H.D. | • Mechano chemical effect<br>• Metalhydride | • Tribo-luminescence | • Friction<br>• Collision<br>• Metalhydride

Table 3.1 Energy Conversion Phenomena from Mechanical Energy [44].

Ohta [44] stated that there are two kinds of energy conversion; direct and indirect energy conversion. Direct energy conversion occurs when the initial form of energy is directly converted to the final one without any intermediate form of energy. Examples include a solar cell (photon energy to electrical energy) or an electrical motor (electrical energy to mechanical energy). Direct energy conversion has, in principle, relatively high efficiency [44] and it is, therefore, considered an important aspect of the design of a regenerative damper.

The other form of conversion is indirect energy conversion, in which more than one energy conversion stage is required between initial and final energy forms. An example of indirect energy conversion occurs with a typical coal fired power station. The chemical energy of the fossil fuel is changed to heat, the heat energy produces water vapour which rotates a turbine to produce electricity. The conversion proceeds from chemical to heat energy, from heat to mechanical and, finally, from mechanical to electrical energy.

In order to achieve high conversion efficiency, a regenerative damper should theoretically use a direct energy conversion process. However, there are some practical circumstances where it may be preferred to use an indirect method of energy conversion. An example for a regenerative vibration system was given in the Literature Review Chapter, Section 2.6. The patent article by Starbard [39] suggested a method of energy regeneration using four separate hydraulic pumps (converting mechanical energy into another form of mechanical energy: hydraulic pressure/velocity). These pumps drove a single centralised fluid pressure generator (converting mechanical energy to electrical energy). This is shown in Figure 2.21. The advantage of this indirect energy conversion
system, was that there was one centralised fluid pump rather than four separate direct electrical energy generating devices in the four suspension locations.

The design of a regenerative damper relies on both the appropriate energy conversion process as well as the most appropriate form of secondary (or recovered) energy. The latter issue is discussed in the following section.

3.5. Conversion to Electrical Energy

3.5.1. Overview

The focus of the research documented in this dissertation was the conversion of vibrational energy into electrical energy. There were several reasons for considering electrical energy as a final energy form. Firstly, electrical energy storage was a relatively straightforward process. Secondly, electrical energy was useful for many subsequent applications, especially for vehicle system applications.

At the time of this research, the technology associated with electrical storage was relatively mature, especially with devices such as secondary (rechargeable) batteries and capacitors, which provided relatively practical and efficient energy storage. For instance, high power, short-term energy storage could be achieved using capacitors or ultra-capacitors, or longer-term energy storage could be realised with secondary (rechargeable) batteries. Although secondary batteries utilise a chemical process to store charge, for the purpose of simplicity in this discussion, they were considered as an electrical form of energy storage.

Other alternatives to electrical storage included either mechanical, thermal or chemical storage [45]. Each of these storage forms had advantages and disadvantages. Although
chemical storage could have the advantage of a high energy density (as for the example of fossil fuels), the disadvantage of this energy form occurred due to the difficulty of controlling the energy conversion process. This was especially true for the conversion between mechanical and chemical energy, as was needed for a regenerative damping application. Another alternative was thermal energy storage. This technology, however, was generally less efficient, and less mature than electrical storage technology [45]. There was also a problem for long-term storage, in which the storage efficiency was compromised due to energy (heat) loss [45].

Other possible storage methods included flywheels (kinetic energy storage) or mechanical springs (potential energy storage). Although these devices had the potential for high storage efficiency, there were several reasons why these devices were not considered further for this research. This included the problems associated with the incorporation of the elements in vehicle systems. Also, as noted in Section 3.2, it was necessary to consider the effect on the damping properties of the system. The force-velocity relationship of spring and flywheel systems did not have a linear force-velocity relationship as needed for regenerative damping. Therefore, an intermediate power-transmission stage would have been needed between the damper and energy storage device. The practical implementation of this system would not have been straightforward\(^2\). Fodor and Redfield [14] described some of the limitations associated with a mechanical power-transmission device, in an investigation into the Variable Linear Transmission (VLT) regenerative damping system. These limitations included non-ideal actuation, limited frequency response, and degraded efficiency (see Section 2.2.2).

The other important aspect in the decision to use electricity as a secondary energy, was due to the usefulness of this energy for subsequent applications. There were many applications that could directly use regenerated electrical energy. Examples included

\(^2\) A form of power-transmission was proposed as part of this Doctoral research for an electrical storage system (see Chapter 5). The basis of this proposal was that it would be more straightforward to implement a power-transmission system using power-electronic devices to modify the voltage-current relationship in an electrical system, rather than modify the force-velocity relationship in a mechanical system, such as for the VLT system.
lighting, electric motors and household appliances. As discussed in the Introduction Chapter, Section 1.2, electric vehicles may benefit from the use of regenerative damper technology. This was true, especially if the vibrational energy, otherwise lost in the vehicle suspension, was used directly to propel the vehicle.

If a direct energy conversion system was utilised for the purpose of a regenerative damper, from Table 3.1, it could be recognised that the choice of energy conversion was either a generator (electromagnetic), piezo device or magneto-hydro-dynamic (MHD) device. If an indirect energy conversion was utilised, any combination of energy conversion could be used in any number of stages. A direct method of energy conversion would be preferred for two main reasons. Firstly, as previously discussed, direct energy conversion systems have, in principle, a relatively high efficiency [44]. The other advantage of direct energy conversion was due to simplicity. Generally, the fewer energy conversion stages there were, the simpler the overall system would have been. The following sections discuss the possible processes that could be used to convert mechanical energy into electrical energy, using a direct energy conversion system.

3.5.2. Piezoelectric Materials

Piezoelectric materials are suitable for many applications in which the conversion between mechanical and electrical energy is required. These applications include:

- High Voltage Generators (for ignition purposes):
  - Gas Appliances
  - Cigarette Lighters
- Ultrasound Applications:
  - Sonar
  - Microphones, Loudspeakers
- Resonators and Filters:
  - Radio
  - Telecommunications
- Miscellaneous:
  - Fine Movement Control
  - Flow Meters

A piezoelectric device (crystal) may be defined as [46]:

"...a crystal in which 'electricity or electric polarity' is produced by pressure; or, more briefly, as one that becomes electrified on squeezing; or as one that becomes deformed when in an electric field."

The disadvantage of piezoelectric devices is that they have very low compliance. This means that the voltage generated by a small force is very low and, conversely, the displacements obtainable with these transducers are far too small for many applications. Generally, the voltages and forces required to produce the displacements are very high [47].

One solution to the problem of low compliance is to use a flexure element such as a bi-laminar strip. An example of a bi-laminar strip is shown in Figure 3.3. These devices are suitable for fine movement control, small vibratory motors and keyboard buttons.

![Figure 3.3 Bi-Laminar Piezoelectric Strip [47.](image)](image)

Even though devices such as the bi-laminar strip have been developed, the use of piezoelectric devices were not considered for the purpose of regenerative damping elements in this research. This was due to the low-compliance issue, in that the force-
displacement characteristic was not suitable for the forces and displacements required by a damping application as stated in Section 3.2.

3.5.3. Magneto-Hydro-Dynamic (MHD) Energy Conversion.

The Magneto-Hydro-Dynamic (MHD) power generator is a device that uses the movement of a conducting fluid within a magnetic field to provide energy conversion [44]. An example of this device is given in Figure 3.4. In this example, a channel, which has conductive sides and an insulated bottom, contains a liquid conductor. A magnetic field is applied to the channel in an upward direction. If the fluid conductor flows in the channel, then a voltage will be observed between the side conductive plates. In concept, the MHD device is no different from conventional electrical generators where the conductor is a solid metal, usually copper, but in detail it is very different because the conductor is a fluid and, therefore, is usually compressible [48, 49]. MHD generators, generally, use either partially ionised gas or liquid metal [49, 50].

![Figure 3.4 Magneto-Hydro-Dynamic (MHD) Energy Converter [44].](image)

At the time of this research, the main terrestrial generator applications of MHD systems were for large scale power generation using ionised gas systems [48, 49]. Smaller scale generation, with liquid-metal generators had been proposed for applications such as space auxiliary power requirements [49]. The partially ionised gas or plasma for gas...
MHD systems, required both high temperatures (in the order of 2000 - 3000 K) and high pressures to become sufficiently conducting. Therefore, due to the practical implementation issues, ionised gas MHD devices were not considered a viable alternative for such an application as regenerative vehicle dampers.

The liquid-metal, MHD generator has the advantage of high electrical conductivity at all temperatures (approximately $10^6$ times that of an ionised gas) [49], and the utilisation of liquid lithium [49, 50], aluminium [49] and mercury [44] had also been proposed. However, for the metal to be in a liquid form (apart from mercury), temperatures in the order of 1200 (K) are needed [49]. The use of non-metal, conducting liquids such as electrolyte solutions could also be used in MHD devices. A study of the conductivity of electrolyte solutions was presented by Murrell and Jenkins [51], in which it was described that, for strong electrolytes, the liquid conductivity is roughly proportional to the concentration of salt in the solution. The molar conductivities for electrolyte solutions in water, such as Sodium-Chlorine (NaCl) and Lithium-Chlorine (LiCl) was presented. For the purposes of determining the optimal regenerative damping system, it would be important to compare the performance of MHD devices operating with electrolyte solutions, or liquid-metal, and conventional electromagnetic devices. However, this analysis was considered beyond the scope of this research program.

The analysis by Womack [49] indicated that liquid metal MHD generator systems appeared to be more suited for space, rather than for terrestrial applications due to the high temperatures required. Also, at the time of this research, the conversion efficiencies were in the order of 5 to 14 per cent [49], which was considerably lower than the typical energy conversion efficiencies of conventional electromechanical power generating devices which were close to unity [45].

There are several disadvantages to the use of MHD generators for the application of regenerative damping. If a liquid-metal is used (apart from mercury), high temperatures and pressures are required to provide a sufficiently conducting liquid-metal. This would not be a problem for conducting electrolyte solutions, however. The utilisation of substances such as mercury or lithium may be unsuitable due to the potential toxic
nature of these substances. It was also indicated that the energy conversion efficiency of liquid-metal MHD devices was considerably lower than devices such as conventional electromechanical energy conversion devices. The conversion efficiency of MHD electrolyte devices was not given. Further investigation should be undertaken in this area, however.

It was recognised that there could be advantages of a MHD regenerative damping system. This could occur if an MHD system was constructed in a similar arrangement to a conventional vehicle damper (shock-absorber). In this arrangement, the conducting liquid would replace the damper fluid, and would allow the system to directly replace a conventional damper in the suspension system.

3.5.4. Electromagnetic Devices

Electromagnetic devices provide a conversion between mechanical and electrical energy using the medium of a magnetic field. There are several forms of electromagnetic devices including linear and rotating DC machines, synchronous and induction machines. The operation of the particular devices, with respect to their potential for regenerative vibration damping is discussed in more detail in the following chapter (Chapter 4). There are many applications for electromagnetic devices, with a large range of both size and power capabilities.

The operation of all electromagnetic devices is governed by the same fundamental principles and laws [52, 53]. The difference in device operation is due to the details of the mechanical construction [52]. The relationship between the developed force (or torque) and current in an AC or DC electromagnetic machine, is governed by the same basic formula, which is derived directly from Ampere's law, as shown in Equation 3.3 [54].
\[ \oint \mathbf{H} \cdot d\mathbf{l} = I_{\text{ENC}} \quad [\text{A}], \]  
\text{where:} \quad \mathbf{H} = \text{magnetic field intensity around a closed path,} \\ d\mathbf{l} = \text{differential length vector, and} \\ I_{\text{ENC}} = \text{current enclosed by the path.} \]

The relationship between machine velocity and generated EMF in machine windings is governed by the same basic formula which is derived from Faraday's Law. For an EMF induced in a circuit with contour \( (C) \), defined as,

\[ V_{\text{EMF}} = \oint \mathbf{E} \cdot d\mathbf{l} \quad [\text{V}], \]  
\text{where:} \quad \mathbf{E} = \text{electric field vector.} \]

Faraday's law predicts that the EMF induced in the circuit with contour \( (C) \) and surface \( (S) \) is given by [54],

\[ V_{\text{EMF}} = -\oint_\mathbf{s} \frac{\partial \mathbf{B}}{\partial t} \cdot d\mathbf{s} + \oint_\mathbf{C} (\mathbf{u} \times \mathbf{B}) \cdot d\mathbf{l} \quad [\text{V}], \]  
\text{where:} \quad \mathbf{B} = \text{magnetic field,} \\ s = \text{differential area vector, and} \\ u = \text{velocity of conducting circuit.} \]

The first term on the right of Equation 3.5 represents the transformer EMF due to the time variation of \( \mathbf{B} \), and the second term represents the motional EMF due to the motion of the circuit in the magnetic field. The division of the induced EMF between the motional and transformer parts depends on the chosen frame of reference [54].

It was shown in the Literature Review that, in many semi-active damping and regenerative damping situations, electromagnetic devices were utilised. For instance, it was revealed in Section 2.3, that Karnopp [25] proposed a linear electromagnetic device called a 'Moving Coil' for a semi-active application. For the analysis, Karnopp used
derivations of both Faraday's and Ampere's law, as a fundamental stage in the evaluation. The analysis undertaken by Karnopp [25] revealed that the moving-coil device provided a viscous damping response, with a force-velocity characteristic given in Equation 3.6.

$$ F_D = \frac{B_0^2 V}{\sigma} \dot{X} \quad [N], $$

(3.6)

where:

- $F_D =$ force produced by the damper,
- $B_0 =$ magnitude of the magnetic field,
- $V =$ volume of conducting material,
- $\dot{X} =$ relative device velocity, and
- $\sigma =$ conductor resistivity.

Karnopp explained how the damping properties of electromagnetic devices may be feasible for mechanical dampers and vehicle suspension systems in general. Also, by transferring some of the vibrational energy to an external system, as analysed by Suda and Shiiba [18] (in Section 2.2.3), it may be feasible to use electromagnetic devices for regenerative vibration dampers. It was, therefore, considered that these devices should be analysed further to determine their performance with respect to regenerative damping applications.

Electromagnetic devices have several advantages over the previous energy conversion devices discussed in this chapter. Firstly, in comparison to piezo devices, electromagnetic devices generally have a much higher compliance than piezo devices, and are, therefore, more suited to large-scale applications such as vibration damping.

In comparison to the MHD energy conversion devices, electromagnetic devices offer an advantage due to simplicity of implementation and operation. Electromagnetic devices rely on the mechanical interaction of conductors moving through a magnetic field. MHD devices also use this principle, however, the energy conversion process relies on a fluid conducting medium. It was shown that MHD devices had several disadvantages due to high pressures and temperatures required to have a sufficiently conducting fluid. This
would add significantly to the complexity, reliability and cost of such a system relative to electromagnetic devices. Also, it was indicated that the energy conversion efficiency of MHD devices, relative to conventional electromagnetic energy conversion systems, was quite low. Therefore, electromagnetic devices show the most promise as regenerative damping devices, and are the focus for further investigation for this Doctoral research.

3.6. Conclusions

In this chapter it was revealed that the two main requirements of the regenerative damping system were to maximise energy regeneration while maintaining adequate damping. Also described in this analysis were the potential advantages of converting the vibrational energy into electrical energy. The advantage of electrical energy mainly stemmed from its use for subsequent applications, as well as its ability to be conveniently stored for long periods of time through the use of devices such as secondary batteries.

A number of devices were also discussed, with respect to the potential for converting the mechanical, vibrational energy into electrical energy, especially for a regenerative damping application. At the time this research was undertaken, it was found that, due to their energy conversion efficiency, high compliance and viscous damping properties, electromagnetic devices showed the most promise as regenerative damping elements.

One finding, given in the Literature Review chapter (Chapter 2), was that, although electromagnetic devices had been used for regenerative damping, no published research was uncovered on the design optimisation or selection of the most suitable form of device for use as a regenerative damper. The following chapter, Chapter 4, documents the investigation on this topic.
"Why does this magnificent applied science which saves work and makes life easier bring us so little happiness? The simple answer runs: Because we have not yet learned to make sensible use of it."

Albert Einstein

4.1. Overview

The analysis documented in this chapter specifically investigated electromagnetic devices, with the objective of determining the important design characteristics for their use as regenerative damping devices. This investigation was undertaken due to the previous analysis (documented in Chapter 3) that indicated the potential benefits of these devices, and the limited published research of the most suitable electromagnetic devices for applications such as vehicle damping.

1 Source: Albert Einstein, Address, California Institute of Technology, February 1931.
In the Literature Review, a number of electromagnetic damping systems were analysed (refer to Sections 2.2.3, and 2.3). However, the analysis revealed that there needed to be further research into the most appropriate form of electromagnetic device for applications such as semi-active and regenerative vehicle damping. An example of this was documented in an article by Suda and Shiiba [18]. In this investigation, a rotating DC motor was used as a semi-active, regenerative damper. In reference to the use of this motor, Suda and Shiiba stated that "...it is not the most suitable one for automobile suspensions [18, p. 644]". The authors stated that a further investigation was needed to determine the applicability of these devices for vehicle suspension.

4.2. **Electromagnetic Damper Design**

4.2.1. Overview

Electromagnetic devices are machines that perform the function of exchanging energy between a mechanical and electrical system through the medium of a magnetic field. For the design of a regenerative vibration damper, the primary interest is for a generating action; the exchange of mechanical to electrical energy. Electromagnetic devices can be characterised by the operation of the electrical system. Generally, if the electrical system has alternating current, the devices are referred to as AC machines and, conversely, if the electrical system is characterised by direct current, the devices are referred to as DC machines [52].

A basic energy transfer model of a generalised electromagnetic machine is shown in Figure 4.1. It can be seen that energy transfer takes place in the device through the medium of a coupling magnetic field. For an electromagnetic machine operating as a
regenerative damper, the input mechanical energy corresponds to a force-velocity relationship, and output electrical energy corresponds to a voltage-current relationship.

The damping response is determined by the relationship between the force-velocity characteristic of the machine and, as discussed in Section 3.2, this response should follow that of a linear, viscous damper. The energy regeneration in the electromagnetic damper is determined by the voltage-current relationship from the electrical system. The interdependence between the force-velocity and voltage-current of the electromagnetic device means that the construction, or the type of electromagnetic device used, determines to what extent the electromagnetic machine performs as a regenerative damper. The following sections document an analysis of the three fundamental electromagnetic devices: the induction, synchronous and DC machine, to determine their potential as regenerative dampers.

4.2.2. Induction Machines

Induction machines provide conversion between mechanical and electrical energy with the use of a moving magnetic field. These machines represent a class of apparatus that include induction motors, induction generators, induction frequency converters and electromagnetic slip couplings [53]. For the case of rotating induction motors under normal operation, an AC energy source is connected to one winding, usually the field winding, to cause a rotating magnetic field. These machines are, therefore, defined as
AC machines. Currents are made to flow in the armature winding by the process of induction then, according to Lenz's law, the conductors develop a mechanical force in the same direction as the rotating magnetic field. The difference between the speed of the rotating magnetic field and the speed of the rotor is called the slip speed. For a motor action the slip speed is always greater than zero, i.e., the field speed is always greater than the rotor speed. For generator action to occur, the slip speed is less than zero. An illustration of a three-phase induction motor is given in Figure 4.2, in which the stator winding has three coils, $aa'$, $bb'$ and $cc'$, 120 electrical degrees apart.

![Figure 4.2 Three-Phase Induction Motor](image)

Shown in Figure 4.3 is a diagram of the developed torque (and line current) as a function of rotor speed for a four-pole, 60 (Hz), line connected, induction machine. It can be seen that the torque is not linearly related to the rotor speed. For the purposes of electromagnetic damping for vehicle systems a line connected system would not be suitable due to the isolation of the electromagnetic device from a grid system. Therefore, an isolated induction generator would be required.
Isolated (self-excited) induction generators are increasingly used in renewable energy systems which employ wind or mini-hydro power [55]. As such, investigations of the steady-state behaviour of both unregulated [55-57] and regulated [55, 57] isolated induction generator systems have been undertaken. An isolated induction generator requires the use of capacitors in parallel with the stator windings to provide the magnetising current necessary for the build up of magnetic field [53, 55]. The build up of magnetic field is illustrated in Figure 4.4. It is a gradual process in which the residual magnetism in the generator causes an initial current ($I_0$) to flow in the capacitor. This current also flows in the (magnetising reactance of the) stator which causes the voltage across it to rise (to $V_1$). This voltage causes a higher current ($I_1$) in the capacitor, and so forth, until an intersection of the capacitance line and magnetisation curve\(^2\).

---

\(^2\) The non-linearity in the magnetisation curve is caused by magnetic saturation effects in the stator and rotor iron.
There is a value of capacitive reactance that causes the capacitance line to be tangent to the magnetisation curve. A reactance below this will prevent the build-up of voltage. The capacitive reactance \( X_c \) is shown in Equation 4.1.

\[
X_c = \frac{1}{2\pi f C_A} = \frac{V_{op}}{I_{op}} \quad [\Omega],
\]

where:
\( C_A \) = value of capacitance,
\( V_{op}, I_{op} \) = operating point for voltage and current, respectively.

For a self-excited induction generator rated at a particular frequency \( f \), there is a critical capacitance \( C_A \) required for the voltage build-up [53]. Conversely, for a given value of capacitance, there is a critical frequency required for voltage build-up and, as indicated in Equation 4.1, the critical frequency increases as the capacitance value decreases.

For operational frequencies (related to mechanical input velocity) below the critical frequency, there would be no damping and no energy transfer to an external load. Also, due to the random, bi-directional input velocities expected in a regenerative damping application, this would result in a non-linear force-velocity relationship. These factors may limit the potential for the self-excited induction generator for applications such as
electromagnetic damping of vehicle systems, in which random, bi-directional vibration input is expected.

Further analysis of induction generators for the application of regenerative damping was not undertaken in this investigation. It was understood, however, that the analysis presented here was by no means exhaustive and, therefore, further research should be undertaken to develop a full understanding of induction generators for this application. Although the analysis indicated a non-linear force-velocity relationship due to the self-excitation process, it did not eliminate the potential for induction machines to operate as regenerative dampers. Further research is considered important, especially considering the significant advantages of induction machines. They are rugged, relatively inexpensive and require very little maintenance; all considered significant factors for the successful implementation of regenerative vibration dampers.

4.2.3. Synchronous Machines

The rotating, synchronous machine is similar to the induction machine described previously, in that the stator produces a rotating magnetic field through an AC electrical input connected to the field windings. However, in the synchronous machine, the rotor consists of an electromagnet supplied with a DC source or, in the case of small-scale machines, permanent (rare-earth) magnets. Similar to the induction machine a mechanical force is produced in the same direction as the rotating magnetic field. An example of a synchronous machine is shown in Figure 4.5.
Synchronous machines may be operated either as motors or generators. In generator operation, synchronous generators are commonly used as constant-speed machines due to the direct relationship between speed and the line frequency. However, synchronous generators may be used for variable speed operation in isolation. To demonstrate the ability of the synchronous machine to act as an electromagnetic damper, a modified version of the synchronous machine design in Figure 4.5, is shown in Figure 4.6. The difference in the representations is due to a change in reference frame of the conductors relative to the magnetic field. The device in Figure 4.6 has the conductors located in the rotor and the magnetic field supplied by permanent magnets located in the stator\(^3\).

To conceptualise the operation of the synchronous generator, the device is shown with a stationary, but alternating, magnetic field. The rotor, which consists of electromagnets, moves within the magnetic field and causes a change in magnetic polarity across the electromagnets, as a function of rotor position. An alternating EMF (transformer EMF) is induced in electromagnet coils due to Faraday's Law.

\(^3\) This representation of the synchronous generator is given for the purposes of explanation of device operation, rather than to present a typical machine configuration.
During the course of this Doctoral research, there was no published literature recovered that analysed the force-velocity relationship of synchronous generators for the purpose of electromagnetic damping. Therefore, this analysis was undertaken, and was based on the operation of the synchronous generator shown in Figure 4.6. To generalise the analysis, the force-velocity relationship was evaluated for one 'electromagnet' section of the synchronous generator. The results for one electromagnet could then be scaled for a device with more electromagnet elements. The force-velocity relationship was given relative to the device construction, conducting volume and magnetic field strength.

The electromagnet of the synchronous generator consisted of a number of conductor coils coupled magnetically through a ferromagnetic core. This is also illustrated in Figure 4.7, in which \( A_c \) was the cross-sectional area of the core, \( r_c \) was radius of the core, \( w \) was the width of the conducting coil, and \( l_c \) was the length of the electromagnet.
Equation 4.2 gives the 'transformer EMF' \( E_{\text{TRAN}} \) induced in the stationary closed circuit as described by Faraday's law of electromagnetic induction [54].

\[
E_{\text{TRAN}} = -N \frac{d\Phi}{dt} \quad [\text{V}],
\]

(4.2)

where: 
- \( N \) = number of conductor coils,
- \( \Phi = \int B \cdot ds \) = magnetic flux linking the circuit,
- \( ds = a_s ds \) = differential surface area, and
- \( a_s \) = unit vector directed longitudinally along the core.

It was assumed that the magnetic field \( B (= a_s B(t)) \) was uniform within the core, directed longitudinally along the core and had a magnitude varying sinusoidally, as shown in Equation 4.3.

\[
B(t) = B_0 \sin(2\pi f_s t) a_s \quad [\text{T}],
\]

(4.3)

where: 
- \( f_s \) = frequency of the varying magnetic field, and
- \( B_0 \) = maximum amplitude of the magnetic field.

For the purpose of explanation, a linear synchronous generator was used for this analysis. It was also assumed that the magnetic field was produced with permanent magnet devices. The device is illustrated in Figure 4.8, in which the magnetic field
varied as a function of the relative position between the permanent magnets and ferromagnetic cores.

**Figure 4.8 Core and Permanent Magnets in the Generalised Synchronous Generator.**

The model in Figure 4.8 established the assumption that the time-varying magnetic field had an amplitude of \( B_0 \). This was because the maximum magnetic field was only supplied to the core when the magnets and cores lined up. Using these assumptions, the magnetic flux density, as a function of time, within each core was,

\[
\Phi(t) = A_C B_0 \sin(2\pi f_b t) \quad [\text{Wb}].
\]  
(4.4)

The induced EMF \( (E_{\text{TRAN}}) \), was then evaluated using the magnetic flux density (Equation 4.4) and Faraday's law (Equation 4.2),

\[
E_{\text{TRAN}} = -N A_C B_0 2\pi f_b \cos(2\pi f_b t) \quad [\text{V}].
\]  
(4.5)

Therefore, the total power dissipated for the device \( (P_{\text{TRAN}}) \) connected to an external resistance, is given by,

\[
P_{\text{TRAN}} = \frac{(N A_C B_0 2\pi f_b \cos(2\pi f_b t))^2}{R_{\text{INT}} + R_{\text{EXT}}} \quad [\text{W}],
\]  
(4.6)

where: \( R_{\text{INT}} \) = internal coil resistance, and \( R_{\text{EXT}} \) = external resistance.
The average the power dissipation ($\overline{P}_{\text{TRAN}}$), is given in Equation 4.7, where $T (= \frac{1}{f_B})$ was the period of the time-varying magnetic field.

$$\overline{P}_{\text{TRAN}} = \frac{1}{T} \int_{t=t}^{t+T} P_{\text{TRAN}} dt \ [	ext{W}]. \quad (4.7)$$

Also, the time-averaged cosine-squared function was given by,

$$\frac{1}{T} \int_{t=t}^{t+T} \cos^2(2\pi f_B t) \ dt = 0.5. \quad (4.8)$$

Therefore, the average power dissipation was given by,

$$\overline{P}_{\text{TRAN}} = \left( \frac{N A_c B_0}{R_{\text{INT}}+R_{\text{EXT}}} \right)^2 \frac{2\pi^2}{f_B^2} \left[ \text{W} \right]. \quad (4.9)$$

Referring to Figure 4.8, the frequency of the varying magnetic field could be given by,

$$f_B = \frac{\dot{X}}{d} \ [1/\text{s}], \quad (4.10)$$

where: $\dot{X}$ = relative velocity between the permanent magnets and core, and $d$ = distance between two 'same-polarity' permanent magnets.

It was then possible to evaluate the power dissipation with respect to the relative device velocity. This was given by,

$$\overline{P}_{\text{TRAN}} = \frac{2\pi^2}{d^2} \left( \frac{N A_c B_0}{R_{\text{INT}}+R_{\text{EXT}}} \right)^2 \dot{X}^2 \ [\text{W}]. \quad (4.11)$$

In a dissipative, physical system, the dissipated power is related to the force and velocity by,
\[ P_{\text{Diss}} = \mathbf{f} \cdot \mathbf{x} \quad [W], \quad (4.12) \]

where: \( P_{\text{Diss}} \) = dissipated power, \( \mathbf{f} \) = force vector, and \( \mathbf{x} \) = velocity vector.

Therefore, the force produced by the synchronous generator \( (F_D) \) was proportional to the relative velocity across the device, as shown in Equation 4.13 and, as the power was dissipated, the force was directed in the opposite direction to the velocity.

\[ F_D = \frac{2\pi^2 \left( N A_c B_o \right)^2}{d^2 \frac{R_{\text{INT}} + R_{\text{EXT}}}{X}} \quad [N]. \quad (4.13) \]

The force-velocity relationship given in Equation 4.13 indicated that the synchronous generator operated as a viscous damper and could, therefore, be used as a variable damper for applications such as regenerative damping systems. It was possible to evaluate the equivalent, average damping coefficient \( (C) \), of the device from Equation 4.13.

\[ \overline{P}_{\text{Diss}} = \overline{C} \overline{X}^2 \quad [W]. \quad (4.14) \]

Using Equations 4.11 - 4.14, the equivalent damping coefficient of the synchronous generator \( (C) \), was defined as the time-averaged damping coefficient, and is shown in Equation 4.15.

\[ C = \frac{2\pi^2 \left( N A_c B_o \right)^2}{d^2 \frac{R_{\text{INT}} + R_{\text{EXT}}}{X}} \quad [\text{Ns/m}]. \quad (4.15) \]

The maximum damping coefficient \( (C_{\text{MAX}}) \), occurred for an external resistance of zero, and was given by,

\[ C_{\text{MAX}} = \frac{2\pi^2 \left( N A_c B_o \right)^2}{d^2 R_{\text{INT}}} \quad [\text{Ns/m}]. \quad (4.16) \]
Therefore, it was shown that the synchronous generator had the ability to transfer electrical energy to an external system (in this example the external resistance), and had a viscous damping response. Both of these factors were important determinants of regenerative damper design.

In order to simplify this analysis, several assumptions were made. It was assumed that the frequency of the generated EMF was low enough, such that the coil inductance effect was negligible. The effect of the coil inductance on the system model, is that an equivalent discrete inductor \((L)\) is placed in series with the internal resistance \((R_{\text{INT}})\), as shown in Figure 4.9. For low-frequency operation, the damping results were independent of the inductive effect because the evaluation of the synchronous generator damping coefficient was directly related to the real power dissipated in the coil (see Equations 4.12 - 4.15). However, for high-frequency operation, the coil inductance will affect these results, and lead to a reduced damping coefficient (due to reduced current flowing in the circuit). The coil inductance will also affect other aspects of the device operation. For instance, to maximise power transfer to the load, a capacitive reactance should be added to the load to counter the internal coil inductance.

![Figure 4.9 Equivalent Synchronous Generator Circuit with Added Inductance.](image)

The effect of the AC resistance due to the skin effect was also not included in this analysis. This effect will increase the effective internal resistance \((R_{\text{INT}})\) for high frequencies and small diameter conducting coils. This will, subsequently, reduce the damping coefficient of the machine. It was also assumed that the magnetic field within the electromagnet core was uniform, and that the magnitude of the magnetic field varied...
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sinusoidally. If synchronous generators are to be further analysed for the application of regenerative damping, these assumptions need to be analysed in order to determine the limits of this analysis.

4.2.4. Direct-Current Machines

The review of previous investigations in the field of regenerative and electromagnetic damping was documented in the Literature Review (Chapter 2). This analysis revealed that there had been several previous investigations of DC electromagnetic machines used both as variable dampers for semi-active and active damping situations [25, 26], as well as regenerative dampers for use in semi-active suspension systems [18-23]. The operation of DC machines is based on the production of 'motional EMF' within the machine windings.

The investigation by Karnopp [25] provided an analysis of the damping properties of a 'moving coil', DC generator (refer to Section 2.3). An analysis was undertaken in this section for a general DC generator, however. The reason for this analysis was to determine if the results by Karnopp were applicable for any given DC machine construction. Figure 4.10 shows a generalised device, in which \( V \) was the volume of conducting material, \( \dot{X} \) was the relative device velocity, \( B_0 \) was the stationary magnetic field strength, \( i \) was current flowing in each conducting element, and \( h \) was the length of each conductor.

The electrical equivalent circuit for the device in Figure 4.10 is shown in Figure 4.11. In this diagram, \( N_c \) was the number of identical conductors, \( r \) was the resistance of each conductor, \( E_c \) was the 'motional EMF' potential produced in each conductor, \( R_{\text{ext}} \) was the external resistance, and \( I \) was the external device current.
The voltage around any loop, which included the external resistance, was given by,

$$E_c = i r + I R_{\text{EXT}} \ [V], \ (4.17)$$

which was equivalent to,

$$E_c = I \left( \frac{r}{N_c} + R_{\text{EXT}} \right) \ [V], \ (4.18)$$

and, the internal device resistance ($R_{\text{INT}}$), was given by
\[
\frac{r}{N_c} = R_{\text{INT}} \quad [\Omega]. 
\] (4.19)

The total force produced by the device, and the EMF generated in each conductor is given in Equations 4.20 and 4.21, respectively [54].

\[
F_D = N_c B_0 h i \quad [N]. \quad (4.20)
\]

\[
E_C = B_0 h \dot{X} \quad [V]. \quad (4.21)
\]

The damping coefficient of the DC generator (C), was, therefore, given by,

\[
C = \frac{F_D}{\dot{X}} = B_0^2 h^2 \frac{1}{R_{\text{INT}} + R_{\text{EXT}}} \quad [\text{Ns/m}]. \quad (4.22)
\]

An important design consideration for an electromagnetic damper is the maximum damping coefficient. This is because the maximum damping coefficient is an inherent limit to the ability for the device to damp vibrations, and places a limit to the overall performance of the damper. The maximum damping coefficient \(C_{\text{MAX}}\) occurred for an external resistance of zero, and was, therefore, given by,

\[
C_{\text{MAX}} = \frac{B_0^2 V}{\sigma} \quad [\text{Ns/m}], \quad (4.23)
\]

where: \(\sigma = \) conductivity of the conducting material.

The damping coefficient for a generalised device was, therefore, given with respect to the total volume of the conducting material and the magnetic field. This result was the same as for the 'moving coil' device, given by Karnopp [25]. It was also shown that DC generators had the ability to transfer electrical energy to an external system (in this example the external resistance). This indicates that general DC generators are suitable for use as regenerative dampers.
There were several simplifying assumptions made for the analysis of DC generator devices in this section. It was assumed that the devices fully utilised the available magnetic field, and that the conductors were always within the maximum available magnetic field. This, however, would not always be the case, as demonstrated by Karnopp [25], for an analysis of linear DC electromagnetic devices. Although Karnopp gave some suggestions as to a solution to this problem, the overall efficiency and damping coefficient of many DC generator devices would be reduced due to this effect. Although many rotating DC generators have the conductors totally within the magnetic field, a further problem would arise due to the magnetic field, current and velocity vectors not being mutually perpendicular for all situations.

4.2.5. Discussion

The analysis of electromagnetic devices in this chapter focussed on three main types of electromagnetic devices; the induction, synchronous and DC machines. A subsequent limitation, therefore, occurred due to the exclusion of other forms of electromagnetic machines, including reluctance, hysteresis, stepper motors, universal motors [53], and servomotors [52]. Therefore, to gain a further knowledge in the area of electromagnetic damping, future research is suggested for other forms of electromagnetic devices, including further analysis of the self-excited induction generator.

The objective of this analysis was to determine the most promising electromagnetic device for use as an electromagnetic damper. From this analysis, as well as the investigations reviewed by this author (documented in the Literature Review, Chapter 2), it was believed that the DC machine was the "most promising" electromagnetic device for regenerative damping applications. The two main reasons for this conclusion were:
1. DC machines provide a linear force-velocity characteristic.

The preliminary analysis of regenerative dampers (documented in Chapter 3) indicated that a linear force-velocity relationship was an important damper characteristic. Although the analysis of synchronous generators indicated that the average force was proportional to velocity, the analysis also indicated that, for a constant device velocity, the power dissipation varied proportional to a cosine-squared function (Equation 4.6). This indicated (from Equation 4.12) that the damping force produced by the generator was also proportional to a cosine-squared function. The variable damping force may be detrimental for the application of regenerative damping, especially for low-amplitude input excitation, in which, a smaller than average damping force may be produced. The analysis of induction machines also indicated a potential disadvantage for damping applications in a similar manner. This was because no force was produced below a finite machine velocity. Conversely, the analysis of DC generators indicated an inherently linear force-velocity relationship (Equation 4.22).

2. DC machines are more suited to an ‘impedance-matching’ interface.

As stated in the Literature Review (Chapter 2), an impedance-matching interface between a regenerative damper and storage device offers advantages with respect to damping and regeneration efficiency. Further investigation of the impedance-matching device was, therefore, undertaken in this investigation (documented in Chapter 5). It was considered that the implementation of an impedance-matching interface would be more straightforward with a DC machine-based damper rather than for an AC machine. The main reason for this was due to the linear voltage-velocity response of the DC machine. For an AC machine, the frequency and amplitude of the output voltage change as a function of damper velocity. This
characteristic would make the analysis of an impedance matching device (matched to a constant battery impedance) more complicated than for the DC machine.

Therefore, the remaining analysis documented in this thesis focuses on DC machines rather than other electromagnetic devices for use as regenerative dampers. The analysis in Section 4.3 investigates the topology of DC machines, with the objective of maximising their performance for the purpose of regenerative damping.

4.3. Optimisation of DC Generator Topology

4.3.1. Overview

In Section 4.2, the manner in which the mechanical construction of an electromagnetic device affected the relationship between the mechanical force-velocity, and electrical voltage-current relationships was documented. The manner in which the mechanical construction ultimately determines the particular type of electromagnetic device (e.g., the difference between induction, synchronous and other electromagnetic machines) was also noted. However, the mechanical construction also affects the specific operation of any particular machine. An example of this is the difference in operation between a linear and rotating induction machine. Although the operational principles between force-velocity and voltage-current are the same, the operation of the two devices is different due to their mechanical construction.

This led to the issue of regenerative damper performance of DC generators as influenced by their construction. This investigation is documented in the following section.
4.3.2. DC Generator Topology

A diagram of a generalised DC generator was given in Figure 4.10. In this diagram, a conducting volume was moving within a uniform magnetic field. The velocity, current, and magnetic field vectors were represented in a Cartesian coordinate system, and were mutually perpendicular. However, for the purposes of analysing more realistic electromagnetic models, it was more appropriate to represent the electromagnetic device in cylindrical coordinates. The same analysis could also be undertaken in a spherical coordinate system, however it was not as straightforward to physically represent a device in these coordinates.

The generalised cylindrical topology of the DC generator electromagnetic device is shown in Figure 4.12. In this diagram, the three vectors of current, velocity and magnetic field are placed either radially, axially or longitudinally, which led to six possible combinations of vector placement. Only four of the six possible combinations are shown in Figure 4.12 because a radial velocity vector was not realistic in a physical system. The diagram is shown with a vector representation, as well as a physical diagram for each topology. An example of the 'Moving Coil' design is the design that is most often used in loudspeakers, in which a coil of conducting material moves linearly within a radial magnetic field. An example of the 'Rotating Motor' design is the design that is most often used in DC motors. In this design, a number of conducting wires run longitudinally, and rotate around a central axis within a radial magnetic field. However, the conventional DC motor is a modified version of the generalised structure.

The design of the most suitable damper for a regenerative suspension application depends upon the most appropriate device topology. The device topology is influenced by the magnetic and electrical circuits of the device, together with the velocity component of the conductors. Therefore, the analysis documented in the following sections investigate the magnetic circuit (Section 4.3.3), electrical circuit (Section 4.3.4) and velocity component (Section 4.3.5) of the DC generator. This analysis is based on the generalised device topology presented in Figure 4.12.
a) Faraday Disk

b) Moving Coil

c) Radial Fin

d) Rotating Motor

Figure 4.12 DC Generator Device Topology in Cylindrical Coordinates.
4.3.3. Magnetic Circuit Design

The magnetic circuit of a DC generator commonly uses pole pieces to transport energy from the permanent magnet(s) to an air-gap. The pole-pieces consist of a ferromagnetic material, such as iron or mild steel, which provide a low reluctance path for the magnetic flux. The air-gap contains the moving conductors which, using the magnetic energy, are able to supply a damping force and provide regeneration.

One factor constraining the design of the magnetic circuit is that the pole-pieces must form a closed loop for the magnetic flux. If the magnetic circuit is broken by a high reluctance material, such as air, the damping coefficient will be reduced due to a reduction in magnetic energy supplied to the air-gap. Karnopp [25] analysed the magnetic circuit of an electromagnetic damper. This analysis described how it was not possible to arbitrarily reduce the volume of the pole-pieces, as the material saturated at a particular flux density. The saturation of magnetic flux would effectively increase the reluctance of the magnetic path and therefore reduce the damping coefficient.

Karnopp [25] concluded that, for a 'voice coil' device with high energy permanent magnets, the complete weight of the actuator was mainly a function of the iron pole-pieces. This was because only a short length of permanent magnet material was required, but the volume of the pole pieces could not be decreased arbitrarily (as a result of flux saturation). Karnopp concluded that the effective use of high energy magnets was only achieved by damper designs which minimised or eliminated the use of pole pieces. As a result, Karnopp proposed a device based on the 'radial fin' design, shown in Figure 4.12 (c). Figure 4.13 shows the magnetic circuit of this design. It can be seen that the pole pieces, shown as the shaded region, were minimised.

The 'Faraday Disk' design shown in Figure 4.12 (a) also minimised the pole piece mass. However, another longitudinal magnetic flux path would be needed to complete the circuit.
4.3.4. Electrical Circuit Design

The electromagnetic device topology shown in Figure 4.12 revealed that there are two electric circuit designs available for an electromagnetic damper, the series and parallel design. The series design consisted of one length of conductor with the two ends providing the output terminals of the device, and the parallel design consisted of more than one conducting length joined in parallel, with each element connected directly to the output terminals. From Figure 4.12, the 'Faraday Disk' and the 'Radial Fin' were examples of the parallel design, and the 'Moving Coil' and the 'Rotating Motor' were examples of the series design. However, it was also possible to have a combination of series and parallel designs for a modified version of these principle devices.

The performance of the electric circuit in an electromagnetic device was analysed by dividing the circuit into elementary segments. Each segment had an ideal voltage source \((E_c)\), in series with a finite resistance \((r)\). The voltage source represented the induced voltage, or the 'motional EMF', developed in the conductor due to the movement through the magnetic field. The resistive element represented the internal resistance of the conducting medium. The two designs are shown in Figure 4.14. \((V_s)\) and \((V_p)\) were the
output voltages generated across the external resistance \( R_{\text{EXT}} \), for the series and parallel designs, respectively.

Using a similar analysis to the investigation presented in Section 4.2.4, the maximum damping coefficient was evaluated to be the same for the series and parallel damper design. The damping coefficient \( C \), is shown in Equation 4.24. For zero external resistance, Equation 4.47 devolved to the same maximum damping coefficient shown previously in Equation 4.23.

\[
C = B_0^2 h^2 \frac{1}{R_{\text{INT}} + R_{\text{EXT}}} \quad \text{[Ns/m]},
\]

where:

\( h \) = length of each conducting element,

\( R_{\text{INT}}, R_{\text{EXT}} \) = internal and external device resistances, respectively, and

\( B_0 \) = magnetic field intensity.

The electromagnetic damper was able to provide energy regeneration by transferring energy to the output terminals of the damper. The power dissipation \( W_R \), was given by,

\[
W_R = \frac{V^2}{R_{\text{EXT}}} \quad \text{[W]},
\]

where:

\( V \) = voltage across the external resistance.

The Thévenin equivalent circuit for the circuit representations in Figure 4.14, has the parameters,
$R_{TH} = r$, and $V_{TH} = E_C$, for the series design, and

$R_{TH} = r/N$, and $V_{TH} = E_C$, for the parallel design,

where: $R_{TH} =$ Thévenin resistance, and

$V_{TH} =$ Thévenin voltage.

Therefore, maximum power transfer to the external resistance occurred for an external resistance equal to the equivalent Thévenin resistance. In this situation, the power dissipation in the external resistance for the series device ($W_{RS}$), and parallel device ($W_{RP}$), was the same. This is shown in Equation 4.28.

$$W_{RS} = W_{RP} = \frac{B_o^2 X^2 V}{4\sigma} \ [W]$$, provided that

$$R_{EXT} = R_{TH} \ [\Omega].$$

The equivalent internal resistance for the series design was, generally, larger than the parallel design. For example, if the $(N)$ conducting elements in Figure 4.14 (b), were rewired in series, the internal resistance of the equivalent series design would increase by a factor of $N^2$. One potential problem of the parallel design would occur because the equivalent internal resistance of the parallel design was generally low, and it would be difficult to realistically match an external resistance to the internal resistance. If the external resistance was larger than the internal resistance, it would not be possible to obtain maximum power transfer across the output terminals and, additionally, the damping coefficient would be reduced.

As previously noted in the Literature Review (Section 2.2.2), to provide regeneration, the device output voltage must be large enough to overcome the barrier potential of the storage device. For example, at the time of preparing this dissertation, the barrier potential of a typical vehicle battery was 14 (V), with a demand to increase to 48 (V) [58]. The output voltage for the parallel design ($V_P$), is shown in Equation 4.30.
The largest possible output voltage for the parallel design occurred for a large number of parallel conductors \((N)\), and was equal to the separate conductor voltage \((E_C)\), which was given by,

\[
E_C = B h \dot{X} \quad [V]. \tag{4.31}
\]

For the series design, however, the output voltage was generally much larger than for the parallel design due to the increased conductor length. The series output voltage \((V_s)\), was given by,

\[
V_s = \frac{R_{\text{EXT}} E_C}{R_{\text{EXT}} + R_{\text{INT}}} \quad [V]. \tag{4.32}
\]

Although the maximum damping coefficient was the same for both devices, the open-circuit output voltage produced by the device largely depended on the electrical circuit used to construct the device, and the length of the conducting elements. The performance of the 'Radial Fin' and 'Faraday Disk' designs could, therefore, be reduced due to the lower output potential.

It was previously mentioned in Section 4.3.2, that the conventional DC motor was a modification of the general 'Rotating Motor' in Figure 4.12 (d). This modification allowed the conductors to form a series, rather than parallel electrical design. This was achieved by changing the direction of the magnetic flux vector as a function of angular position around the central axis of the device. The direction of the current flow also changed as a function of angular position, such that a continuous length of conducting material could be used.
4.3.5. Linear and Rotating Electromagnetic Devices

The generalised electromagnetic topology of DC generators in cylindrical coordinates was shown in Figure 4.12. The generalised device topology revealed that the velocity vector could either be directed along the longitudinal or axial vector (as shown in Figure 4.12), referring to either linear or rotating electromagnetic devices, respectively. The analysis in this section outlines some of the advantages and disadvantages of using either rotating or linear dampers for the purpose of regenerative damping.

An investigation was undertaken to determine the relative damping performance of the two devices. It was shown that a rotating damper had the same force-velocity relationship as a linear damper if a mechanism such a rack-and-pinion was used to convert rotary to linear motion. To illustrate this, a one degree-of-freedom suspension system with either a linear (a) or rotating damper (b) was analysed, and is shown in Figure 4.15. The system was characterised by a sprung-mass \( (m_v) \), damping coefficient \( (c_v) \) and spring constant \( (k_v) \). The input and sprung-mass displacement were defined as \( (x_o) \) and \( (x_v) \), respectively.

![Figure 4.15 One Degree-of-Freedom Vehicle Models.](image)

For the rotating damper system, the gear radius \( (r_{GEAR}) \), was defined as the radius of the rack-gear, and the gear-ratio \( (\alpha) \), was defined as the ratio between the gear-radius to the average conductor radius \( (r_{COND-AV}) \). The relative velocity \( (V_{REL}) \), was given by,
\[ V_{\text{REL}} = r_{\text{GEAR}} \omega \quad [\text{m/s}], \]  
(4.33)

where: \( \omega \) = radial velocity of the damper.

The torque produced by damper was given by,

\[ \tau = r_{\text{GEAR}} F_D \quad [\text{N m}], \]  
(4.34)

where: \( F_D \) = longitudinal force across the damper mechanism.

The relative velocity was proportional to the angular velocity (Equation 4.33) and the torque was proportional to the longitudinal force across the damper (Equation 4.34), hence, the overall force-velocity relationship for the rotating damper was the same as for the linear damper. One advantage of rotating dampers was highlighted by Suda and Shiiba [18], in their analysis of regenerative, semi-active suspension. This advantage was that the damping coefficient was proportional to the gear-ratio squared\(^4\). This meant, for instance, that for a gear ratio of \( \alpha = 2 \), the electromagnetic damper produced four (two-squared) times the damping coefficient, as a linear device, for the same device mass and magnetic field intensity. This indicated that the use of rotating dampers may have a significant effect on the performance of such devices as regenerative vehicle dampers.

One disadvantage of using rotating electromagnetic dampers for applications such as regenerative vehicle suspensions, occurred due to their effect on the system dynamic response. This situation was analysed previously by Ryba [26], and a review of this article was given in the Literature Review, Section 2.3. This analysis revealed that, when analysing rotating dampers in a one degree-of-freedom representation of a vehicle suspension system, there were problems due to the finite transmission of high-frequency vibrations from the road surface to the vehicle body. This transmission did not occur for similar linear damper systems. Therefore, Ryba [26] then stated that "...the use of an electric motor may cause problems in this point."

\(^4\) The same advantage may be gained from the use of a linear device, such as a linear gear box, with linear damper. This investigation concentrated on the use of rack-and-pinion-type mechanisms with rotating dampers for providing mechanical amplification due to the relative prevalence of these devices.
The analysis given by Ryba was limited to a one degree-of-freedom model, which may not be totally representative of a realistic regenerative damping situation. Therefore, considering the possible benefits of using rotating electromagnetic dampers, a further investigation of the use of rotating dampers in vehicle systems is presented in Chapter 6.

4.4. Conclusions

The investigation documented in this chapter, analysed the use of electromagnetic machines for use in regenerative damping. In Section 4.2, three major types of electromagnetic devices (induction machines, synchronous machines and DC machines) were analysed, with the objective of determining which had the most potential for regenerative dampers. The results indicated that the DC generator had the greatest potential for use as a regenerative damper due to the inherent linearity of the force-velocity and voltage-current of this device.

A generalised topology structure was developed in Section 4.3 to evaluate the important design characteristics of the DC generator. It was revealed that, with respect to the magnetic circuit design, minimisation of the pole-piece mass was an important design consideration of DC generator devices. The two particular forms of device which achieved this were the 'Radial Fin', and 'Faraday Disk' designs. With respect to the electrical circuit design, a DC generator with a series circuit offered the advantage of increased internal resistance and high output voltage in comparison to parallel circuit designs. The two particular forms of device which had a series electrical circuit were the 'Moving Coil' and 'Rotating Motor' devices.

The generalised topology also revealed that either rotating or linear electromagnetic machines could be used as regenerative dampers. The theoretical advantage of rotating dampers was that an increase in damping performance could be achieved for a given
device mass, due to mechanical amplification with the rack-and-pinion mechanism. The disadvantage of rotating dampers occurred for applications, such as vehicle suspensions, in which a vehicle's dynamics could be degraded. Further analysis of rotating dampers used in vehicle suspension systems is presented in Chapter 6.

In relation to the purpose of regenerative damping, the issue of how the output voltage-current energy flow may be transferred to a storage device should also be investigated. Therefore, in Chapter 5, an analysis is presented of a proposed interface between the electromagnetic damper and the storage device.
"Efficiency... is doing better (than) what is already being done."¹

Peter F. Drucker

5.1. Overview

This chapter presents an analysis of the regenerative interface for an electromagnetic regenerative damping system. The interface provides the coupling between the electromagnetic machine and storage device, and controls the energy flows in the regenerative damper. It, therefore, governs both the damping and energy regeneration properties of the system.

It was revealed in the Literature Review, that there had been several 'interface' designs proposed for electromagnetic regenerative and semi-active suspension. Examples of these designs were given in Section 2.2.3, in which one design was proposed by Okada and Harada [19], and used a 'double-voltage' charging circuit (refer to Figure 2.4). Another design by Suda et al. [22]) used relay switching (refer to Figure 2.6). The operation of previous designs had performance limitations with respect to the damping and regeneration response. The main limitation was that, for passive designs such as the 'double-voltage' charging circuit, there was no regeneration and no damping for low machine velocities.

The development of a new regenerative interface stemmed largely from the impedance-matching issue highlighted by Fodor and Redfield [14] and Mahajan and Redfield [15] (refer to Section 2.2.2). This issue referred to the ability of a regenerative damper to match the impedance of the storage device, in order to maximise energy regeneration. Previous interfaces for electromagnetic dampers did not achieve this. Due to the objective of maximising energy regeneration in this research, this issue, in itself, was considered significant for the overall system design.

5.2. Regenerative Interface Proposal

In this section, a regenerative interface, providing energy transfer between the electromagnetic regenerative damper and electrical storage element, is introduced. The proposed device, which was referred to as an 'impedance-matching' regenerative interface, was an electrical circuit composed of switches (transistors), diode and capacitor elements. The interface was specifically designed for the purpose of electromagnetic energy regenerative damping. It was designed to control the energy flow between an electromagnetic machine and storage device, thereby, enabling energy transfer to the storage device, while maintaining adequate damping from the electromagnetic damper.
The main advantage of this interface was that it could transfer energy to a storage device and provide damping when the potential of the electromagnetic DC machine was less than the storage device potential. This would prove to be beneficial for applications which require a high energy efficiency. The regenerative interface was designed for use with DC machines because, as shown in Chapter 4, the electromagnetic device that demonstrated the most promise for use in the regenerative suspension system was either a linear or rotating electromagnetic DC machine.

The design of the impedance-matching interface was based on a 'step-up, DC-DC converter'. The operation of DC-DC converters, in general, was analysed by Mohan [59] and Mazda [60]. There were several reasons for basing the interface design on the step-up converter, rather than other converter types. One reason was that it was considered that the impedance-matching process would be more efficient for stepping-up the machine voltage to a battery potential, rather than stepping-down. Previous interface designs were limited to low storage-battery voltages, such as 1.26 (V) [19]. The reason for the low battery potential was that there was no regeneration and damping for electromagnetic damper voltages lower than the battery voltage. However, considering that at the time of this research, vehicle battery systems were generally 14 (V), with a large impetus to transfer to 42 (V) systems [58], passive designs would be considerably limited for these situations. However, by stepping-up the machine voltage, it was possible to match the impedance for machine voltages up to the battery potential, which meant that the system was not limited with respect to the battery potential. Another reason for basing the interface design on a step-up converter was that it was possible to utilise the internal machine inductance, rather than rely on a discrete inductor component.

A schematic diagram of a step-up DC-DC converter is shown in Figure 5.1. The converter used duty-cycle switching to control energy flow between the input (\(V_d\)) and output (\(V_o\)). The input inductance (\(L\)), inductor voltage (\(V_L\)), and inductor current (\(i_L\)), together with the capacitance (\(C\)), output resistance (\(R\)), and output current (\(i_o\)), are shown in this diagram.
The proposed regenerative interface is shown in Figure 5.2. The interface consisted of the DC machine, represented by an open-circuit voltage \( E_c \), in series with the internal resistance \( R_{\text{INT}} \), and inductance \( L \). The open-circuit voltage was due to the voltage induced in the machine windings from the relative movement of the windings and the magnetic field, and was proportional to the relative velocity across the damper. The internal resistance was due to the finite resistance of the machine windings, and the internal inductance was due to the magnetic energy storage effects of the windings. The storage battery is also shown in the regeneration circuit diagram. The battery was represented by an ideal voltage source \( V_{\text{BATT}} \), in series with the internal battery resistance \( R_{\text{BATT}} \).

There were several similarities between the DC-DC converter in Figure 5.1 and the interface design shown in Figure 5.2. The input voltage and inductance of the DC-DC
converter was equivalent to the machine voltage and internal inductance of the DC machine in the interface design. The output of the DC-DC converter \( V_o \) was equivalent to the battery output in the interface design. The main differences between the two designs were due to the internal machine resistance and battery resistance, not included in the DC-DC converter. The DC machine in the interface design is shown within an 'H-Bridge' section. The H-Bridge was similar to the circuits used in DC machine control and was used to rectify the bi-directional voltage produce by the DC machine.

The circuit had several advantages for use in regenerative damping applications. It was possible to change the voltage-current relationship (impedance) at the output terminals of the DC machine by varying the timing of the semiconductor switching devices (S1-S4). This led to direct control of the damping force produced by the DC machine and the regeneration of energy to the storage battery. The power losses within the circuit were relatively small, due only to the switching losses and the power loss within the diode. Also, the control of the circuit was not restricted by frequency limitations of the control devices. The frequency response of the switches could be made large compared with the suspension frequency response, by using semi-conductor devices.

In this research, it was proposed that the control of the switches (S1-S4) perform two interrelated functions. Firstly, the 'H-bridge' design of the power circuit enabled the rectification of the bi-directional DC machine voltage \( E_C \). The rectification could be performed by closing switches (S1) and (S4), and opening switches (S2) and (S3), when \( E_C \) was positive (as defined in Figure 5.2), and closing switches (S2) and (S3), and opening switches (S1) and (S4) when \( E_C \) was negative. In addition to the rectification control, 'pulse-width modulation' (PWM) was used to modify the output voltage-current relationship, and control the energy regeneration and damping response. The PWM and rectification functions could be readily combined with digital logic. The result was that the regenerative circuit could be in one of four possible states, \( E_C > 0 \), Closed-Circuit), \( E_C > 0 \), Open-Circuit), \( E_C < 0 \), Closed-Circuit), and \( E_C < 0 \), Open-Circuit). A current path diagram and state-diagram for the regenerative circuit is presented in Appendix B.1, Figure B.1 and Table B.1, respectively.
5.3. Theoretical Analysis of Regenerative Interface

5.3.1. Overview

The analysis documented in this section investigated the regeneration and damping properties of the impedance-matching regenerative interface with respect to the circuit parameters, and PWM control.

5.3.2. Interface Analysis

To simplify the analysis, the rectification of the input voltage ($E_c$), was ignored and the previous circuit shown in Figure 5.2 was simplified to the circuit shown in Figure 5.3. In the following analysis it was assumed that the highest frequency component of the machine voltage was negligible compared to the switching frequency. This assumption was based on a switching frequency in the order of 10 - 100 (kHz) compared to a maximum expected frequency component of the machine voltage ($E_c$) of around 10 (Hz). In this case, the PWM control and the rectification control process were independent. The rectification control was dependent only on the polarity of the open-circuit machine voltage, while the PWM control was dependent only on the magnitude of the open-circuit machine voltage.
As the operation of the circuit, shown in Figure 5.3, was independent of the rectification process it was, therefore, assumed that the machine voltage ($E_C$) was always greater or equal to zero. The circuit could be in either of two states:

- The switch was closed, and the DC machine output terminals were short-circuited. This allowed the input machine voltage ($E_C$), to supply energy to the internal machine inductance ($L$).
- The switch was open, and the energy stored in the DC machine inductor discharged to the battery, through the circuit diode.

The function of the capacitor ($C$) was to maintain a relatively constant output voltage at the battery terminals and to hold the cathode of the diode at a higher potential than ground. This ensured that, for the 'switch-off' condition, the diode was reverse biased and that the diode current was negligible. When the switch was open, the regeneration circuit supplied energy to both the load (battery) and the capacitor. When the switch was closed the capacitor discharged through the load.

In order to simplify the analysis, several assumptions were made concerning the ideal behaviour of the circuit elements. Firstly, it was assumed that the internal inductance of the electromagnetic machine was sufficiently large such that there was always a positive inductor current flowing. It was also assumed that an ideal switching device was used,
such that there was a negligible transition time and switch delay between switching states. Also, the 'on' and 'off' switch impedances were zero and infinity Ohms, respectively. It was also assumed that an ideal diode was used, such that the diode had a constant forward-bias potential \( V_d \), and was open-circuit for the reverse-bias condition. The validity of these assumptions are discussed further in the experimental analysis documented in Section 5.4.

The theoretical analysis of this circuit was performed for the circuit operating in steady-state. A diagram of the inductor current \( i_L \) as a function of time is shown in Figure 5.4.

When the switch was on (closed), the current increased to a peak of \( i_{L_{\text{max}}} \) until the switch was turned off, after which the inductor current dropped to \( i_{L_{\text{min}}} \). It was assumed that the minimum inductor current was always greater than zero (i.e., the circuit was operating in a continuous current mode).

![Figure 5.4 Inductor Current Under Steady-State Operation.](image)

For the purpose of simplifying the analysis, it was also assumed that the time constant of the electromagnetic damper \( \tau = \frac{L}{R_{\text{INT}}} \) was large compared to the switching period \( T_s \). Therefore, the derivative of the inductor current (with respect to time) was assumed to be constant, and the average inductor current \( i_{LB} \) was the mid-point between the maximum and minimum inductor current. A diagram of the inductor voltage is shown in Figure 5.5.
For the circuit operating in steady-state, the time integral of the inductor voltage \( V_L \), over one switching period was zero, therefore,

\[
V_{L_{\text{max}}} t_{\text{ON}} + V_{L_{\text{min}}} t_{\text{OFF}} = 0 \quad [\text{Vs}],
\]  

where:

\[
t_{\text{ON}} = \text{switch on (closed) duration}, \quad t_{\text{OFF}} = \text{switch off (open) duration}, \quad T_s (= t_{\text{ON}} + t_{\text{OFF}}) = \text{switching time period}. 
\]

The duty-cycle \( (D) \), was defined as,

\[
D = \frac{t_{\text{ON}}}{t_{\text{ON}} + t_{\text{OFF}}},
\]

which led to,

\[
(E_C - i_L R_{\text{INT}}) t_{\text{ON}} + (E_C - i_L R_{\text{INT}} - V_D - V_{\text{BATT}} - i_{\text{BATT}} R_{\text{BATT}}) t_{\text{OFF}} = 0 \quad [\text{Vs}],
\]

Assuming steady-state operation and applying the principle of conservation of energy, the power from the DC machine \( (P_M) \) was given by,

\[
(E_C - i_L R_{\text{INT}}) = (V_{\text{BATT}} + V_D + i_{\text{BATT}} R_{\text{BATT}})(1 - D) \quad [\text{V}].
\]
\[ P_M = P_{\text{OUT}} + P_{\text{Diss}} \quad [\text{W}], \quad (5.5) \]

where: \( P_{\text{OUT}} \) = output power, and \( P_{\text{Diss}} \) = power dissipated in the interface device.

The power dissipated in the interface was actually dissipated in the diode. The average current across the diode was equal to the battery current \( (i_{\text{Batt}}) \). Therefore, from Equation 5.5,

\[ i_L (E_C - i_L R_{\text{INT}}) = i_{\text{Batt}} (V_{\text{Batt}} + i_{\text{Batt}} R_{\text{Batt}}) + i_{\text{Batt}} V_D \quad [\text{W}], \quad (5.6) \]

From Equations 5.4 and 5.6, the inductor and battery current were given by,

\[ i_L = \frac{E_C + (V_{\text{Batt}} + V_D)(D - 1)}{R_{\text{INT}} + (D - 1)^2 R_{\text{Batt}}} \quad [\text{A}], \quad (5.7a) \]

\[ i_{\text{Batt}} = (1 - D) i_L \quad [\text{A}]. \quad (5.7b) \]

The inductor and battery current were only defined for the condition that they were greater or equal to zero. This ensured that the diode was forward biased. Given that the duty-cycle was constrained between 0 and 1, and that the internal and external resistances were always positive, from Equation 5.7a and Equation 5.7b, \( (i_L) \) and \( (i_{\text{Batt}}) \) were greater, or equal to zero provided that,

\[ E_C \geq (V_{\text{Batt}} + V_D)(1 - D) \quad [\text{V}], \quad (5.8) \]

When the condition, defined in Equation 5.8, was not satisfied, the inductor current became discontinuous. In this case the inductor current dropped to zero for at least part of the switching cycle. From the relationship between the circuit currents and the circuit parameters in Equation 5.7a and Equation 5.7b, it was possible to evaluate the damping
response of the system (given the condition in Equation 5.8). The damping coefficient of the electromagnetic machine ($C_{DAMP}$), was given by [19],

$$C_{DAMP} = \frac{K_E^2}{R_{INT} + R_{EXT}} \text{[Ns/m]},$$  \hspace{1cm} (5.9)

where: $K_E = \text{electromagnetic damper EMF constant}$.

The external interface resistance was the resistance presented to the electromagnetic damper across the damper terminals, which is also shown in Figure 5.3. Although the internal resistance was constant, the external resistance was the ratio of the voltage and current at the output terminals of the electromagnetic machine, and this relationship was dependent on the regeneration-circuit parameters and the duty-cycle. Given that the maximum damping coefficient ($C_{MAX}$), occurred for ($R_{EXT} = 0$), the damping coefficient could also be given by,

$$C_{DAMP} = \frac{C_{MAX} R_{INT}}{R_{INT} + R_{EXT}} \text{[Ns/m]}. \hspace{1cm} (5.10)$$

From Figure 5.3,

$$R_{INT} + R_{EXT} = \frac{E_C}{i_L} \text{[\Omega]}. \hspace{1cm} (5.11)$$

From Equations 5.7a, 5.7b, 5.10 and 5.11, the damping coefficient could be given by,

$$C_{DAMP} = C_{MAX} \left( \frac{R_{INT}}{E_C} \right) \left( \frac{E_C + (V_{BATT} + V_o)(D - 1)}{R_{INT} + (D - 1)^2 R_{BATT}} \right) \text{[Ns/m]}. \hspace{1cm} (5.12)$$

The parameters in Equation 5.12 that remained constant, for a given DC machine and storage battery, were ($C_{MAX}$), ($R_{INT}$), ($V_o$), ($V_{BATT}$) and ($R_{BATT}$). The damping coefficient could, therefore, be given as a function of the remaining variables, machine voltage...
(Ec), and duty-cycle (D). Figure 5.6 shows the damping coefficient (Cdamp), for an example in which the circuit parameters are given in Table 5.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal Machine Resistance</td>
<td>R_{INT}</td>
<td>3.0</td>
<td>(Ω)</td>
</tr>
<tr>
<td>Battery Resistance</td>
<td>R_{BATT}</td>
<td>0.2</td>
<td>(Ω)</td>
</tr>
<tr>
<td>Battery Voltage</td>
<td>V_{BATT}</td>
<td>6.0</td>
<td>(V)</td>
</tr>
<tr>
<td>Forward-Bias Diode Potential</td>
<td>V_{D}</td>
<td>0.7</td>
<td>(V)</td>
</tr>
</tbody>
</table>

Table 5.1 Regeneration Circuit Specifications.

Figure 5.6 Damping Response of the Electromagnetic Regeneration System.

It can be seen that for (D = 1), the DC machine provided maximum damping. In this case, the switch remained closed, ideally providing zero external resistance and, therefore, maximum damping. For a low duty-cycle (D), and low machine voltage (Ec), the current was not defined according to Equation 5.7a and 5.7b. Therefore, for this condition, the damping coefficient (as defined in Equation 5.12) was not defined either.

From the current relationships given in Equation 5.7a and Equation 5.7b, the output power of the system could be given by,

\[ P_{OUT} = i_{BATT}V_{BATT} + i_{BATT}^2 R_{BATT} \quad [W]. \]  

(5.13)
Substituting \((i_{BATT})\), from Equation 5.7b into Equation 5.13, gave,

\[
P_{\text{OUT}} = (D - 1) \frac{(E_C + (V_{BATT} + V_D)(D - 1))(D - 1)^{2} R_{BATT} V_D + (D - 1) R_{BATT} E_C - R_{INT} V_{BATT})}{(R_{INT} + (D - 1)^{2} R_{BATT})^{2}} \quad [W].
\]

(5.14)

The power dissipation was dependent on the circuit parameters, as well as the duty-cycle. Once again, however, all parameters were constant apart from the machine voltage \((E_C)\), and the duty-cycle \((D)\). The output power was, therefore, analysed with respect to these parameters. Figure 5.7 shows the output power \((P_{\text{OUT}})\), for the same example as given for Figure 5.6, in which the circuit parameters were given in Table 5.1.

**Figure 5.7** Output Power of the Electromagnetic Regeneration System.

Figure 5.6 and Figure 5.7 revealed that it was possible to develop a control function involving the duty-cycle \((D)\), and the machine voltage \((E_C)\), in order to control the output power and damping coefficient of the regeneration system. The machine voltage \((E_C)\), varied due to relative velocity across the DC machine which, in many regenerative damping situations, would randomly occur. Therefore, a control function was developed to give the duty-cycle \((D)\), as a function of the machine voltage \((E_C)\). The analysis
documented in Section 5.3.3 defined a control function to maximise the power transfer to the storage battery, while the analysis documented in Section 5.3.4 defined a control function to maintain a particular damping coefficient.

5.3.3. Maximum Power Transfer

The control of the circuit could be developed with respect to maximising the energy transfer to the battery storage element, for a given machine voltage \((E_c)\). In order to maximise the energy conversion, for a given input voltage, the partial derivative of the output power \((P_{\text{OUT}})\), with respect to the duty-cycle was evaluated to equal zero.

\[
\frac{\partial P_{\text{OUT}}}{\partial D} = 0. \tag{5.15}
\]

From Equations 5.14 and 5.15, to maximise the energy conversion, the relationship between input voltage \((E_c)\), and duty-cycle \((D)\) is given by,

\[
D = 1 + \frac{R_{\text{INT}}(V_{\text{BATT}} + V_D) - \sqrt{R_{\text{INT}}(V_{\text{BATT}} + V_D)^2 R_{\text{INT}} + E_c^2 R_{\text{BATT}}}}{E_c R_{\text{BATT}}}. \tag{5.16}
\]

To simplify the analysis, if the battery resistance \((R_{\text{BATT}})\), was considered to be negligibly small, the control function was given by,

\[
D = 1 - \frac{E_c}{2(V_{\text{BATT}} + V_D)}. \tag{5.17}
\]

One limitation of Equations 5.16 and 5.17 was that the duty-cycle \((D)\), was constrained between 0 and 1. For the example of the relation given in Equation 5.17, when the machine voltage \((E_c)\), became larger than \(2(V_{\text{BATT}} + V_D)\), the duty-cycle reduced to zero, in which the switch was continually open. In this case, the circuit given in Figure 5.3
approximated the passive regeneration circuit proposed by Okada and Harada [19] (refer to Section 2.2.3, Figure 2.4).

5.3.4. Damping Coefficient Control

In some instances such as semi-active damping, it may be beneficial to control the damping coefficient. A control law could be developed from Equation 5.12 to maintain the damping coefficient \( C_{DAMP} \) at a given fraction of \( C_{MAX} \). The normalised damping coefficient \( F_{DAMP} \), was defined as the damping coefficient produced, relative to the maximum possible damping coefficient of the DC machine.

\[
F_{DAMP} = \frac{C_{DAMP}}{C_{MAX}}. \tag{5.18}
\]

Substituting the normalised damping coefficient, Equation 5.18, into Equation 5.12 led to the following control function,

\[
D = 1 + \frac{R_{INT} (V_{BATT} + V_D) - \sqrt{R_{INT} (V_{BATT} + V_D)^2 R_{INT} + 4(1 - F_{DAMP}) F_{DAMP} E_C^2 R_{BATT}}}{2F_{DAMP} E_C R_{BATT}}. \tag{5.19}
\]

Once again, if it was assumed that the battery resistance was negligibly small, the analysis led to the control function,

\[
D = 1 - \frac{E_C (1 - F_{DAMP})}{(V_{BATT} + V_D)}. \tag{5.20}
\]

Figure 5.8 shows the output power \( P_{OUT} \), as a function of the machine voltage \( E_C \), and the normalised damping coefficient for the control function in Equation 5.19. The output power 'flattened out' for large machine voltages \( E_C \), and small values of
normalised damping ($F_{DAMP}$). This was because of the constraint that the duty-cycle ($D$), was always greater than zero.

![Graph of Output Power as a Function of Machine Voltage and Normalised Damping.](image)

**Figure 5.8** Output Power as a Function of Machine Voltage and Normalised Damping.

One factor that shed further insight on the operation of the regeneration circuit occurred due to the relationship between the normalised damping ratio and the condition of maximum power transfer. For a normalised damping coefficient of ($F_{DAMP} = 0.5$), it could be shown that the damping control function (Equation 5.19) devolved to the maximum power transfer relationship (Equation 5.16). Similarly, for the condition that the internal battery resistance was negligible, Equation 5.20 devolved to Equation 5.17. This finding indicated that maximum power transfer occurred for a normalised damping ratio of ($F_{DAMP} = 0.5$). From the relationship between the damping coefficient and the internal and external resistance given in Equation 5.10, this also indicated that maximum power transfer occurred under the condition that the circuit produced an effective output resistance ($R_{EXT}$) to match the source resistance ($R_{INT}$).
5.3.5. Energy Regeneration Efficiency

In addition to maximising the power transferred to the storage device, for a given input voltage \((E_C)\), it was also important to evaluate the overall regeneration efficiency. The efficiency was defined as the proportion of the input energy that was transferred to the storage battery. The input energy occurred due to the mechanical energy input into the DC machine, or the force-velocity input. Assuming an ideal energy conversion process between the mechanical and electrical systems, the input energy resulted in a voltage-current relationship produced by the DC machine. The open-circuit voltage \((E_C)\), was directly proportional to the relative velocity across the damper, and the force produced was directly proportional to the current induced in the machine windings \((i_L)\).

Therefore, referring to Figure 5.3, the regeneration efficiency \((E_{REG})\) could be defined according to Equation 5.21,

\[
E_{REG} = \frac{P_{OUT}}{P_{IN}},
\]

in which,

\[
P_{IN} = E_C i_L \quad [\text{W}],
\]

and the output power \((P_{OUT})\), was previously defined in Equation 5.13. If it was assumed that the battery resistance \((R_{BATT})\), was negligible, the power efficiency of the regenerative system could be given by,

\[
E_{REG} = \left(1 - F_{DAMP}\right) \frac{V_{BATT}}{V_{BATT} + V_D}.
\]

For a large battery voltage \((V_{BATT})\), relative to the forward diode potential \((V_D)\), the power efficiency could be given by,
The regenerative system power efficiency, as defined in Equation 5.23, is shown in Figure 5.9, with regenerative system parameters given in Table 5.1.

\[ E_{\text{REG}} = (1 - F_{\text{DAMP}}). \]  

\[ (5.24) \]

Figure 5.9 reveals that the power efficiency reduced below the efficiency given in Equation 5.23 for high machine voltages \( E_c \), and for a low normalised damping coefficient \( F_{\text{DAMP}} \). This was, as explained in previous sections, because the duty-cycle approached zero, and the switch shown in Figure 5.3 remained open.
5.4. Steady-State Experimental Analysis

5.4.1. Overview

The purpose of this section is to document an experimental steady-state analysis of the regenerative interface system. The performance and limitations of the system were evaluated under steady-state conditions such that the results could be compared directly with the previous theoretical analysis presented in Section 5.3.

5.4.2. Experimental Apparatus

A photograph and schematic diagram of the steady-state experimental apparatus are shown in Plate 5.1 and Figure 5.10, respectively.

Plate 5.1 Photograph of the Steady-State Experimental Apparatus.
For these experiments, a variable-voltage supply powered an electromagnetic DC machine called the 'Driving Machine'. The driving machine was operated at a constant angular velocity, and was coupled to the electromagnetic damper via a gear mechanism. The regenerative system specifications are presented in Table 5.2.

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Storage Battery (B38-6A)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internal Battery Resistance*</td>
<td>$R_{\text{BATT}}$</td>
<td>0.25 ($\Omega$)</td>
</tr>
<tr>
<td>Open-Circuit Battery Voltage</td>
<td>$V_{\text{BATT}}$</td>
<td>6.20 (V)</td>
</tr>
<tr>
<td>Electromagnetic Damper (S-19-3B)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internal Machine Resistance*</td>
<td>$R_{\text{INT}}$</td>
<td>1.06 ($\Omega$)</td>
</tr>
<tr>
<td>Internal Machine Inductance*</td>
<td>$L$</td>
<td>2.30 (mH)</td>
</tr>
<tr>
<td>Voltage Constant</td>
<td>$K_{E}$</td>
<td>0.155 (Vs/rad)</td>
</tr>
<tr>
<td>Torque Constant</td>
<td>$K_{T}$</td>
<td>0.155 (Nm/A)</td>
</tr>
<tr>
<td>Diode (RURP1560)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Forward Diode Voltage</td>
<td>$V_{D}$</td>
<td>0.85 (V)</td>
</tr>
<tr>
<td>Capacitor (AL-10A102BB100)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Capacitance</td>
<td>$C$</td>
<td>1000 ($\mu$F)</td>
</tr>
<tr>
<td>Transistor(s) (F15N06L)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Static Drain-Source On Resistance</td>
<td>$R_{\text{DS(ON)}}$</td>
<td>0.14 ($\Omega$)</td>
</tr>
<tr>
<td>Turn-On Delay Time (typ)</td>
<td>$t_{\text{ON}}$</td>
<td>16 (ns)</td>
</tr>
<tr>
<td>Turn-Off Delay Time (typ)</td>
<td>$t_{\text{OFF}}$</td>
<td>200 (ns)</td>
</tr>
</tbody>
</table>

Table 5.2 Regenerative System Specifications.

* Refer to Appendix B.3 for the experimental derivation.
The regenerative interface used in the experimental steady-state analysis was the circuit shown originally in Figure 5.3. Refer to Appendix B.2, Figure B.2 for a detailed circuit diagram, and Tables B.2, B.3 and B.4 for the component specifications.

The regenerative interface was controlled using a personal computer, according to the control function, Equation 5.20, evaluated previously in Section 5.3.4. A measure of the angular velocity of the electromagnetic damper was given by a 'Tacho-Generator', which was a 12 Volt DC electromagnetic device directly connected to the electromagnetic damper. A software algorithm converted the input analog signal from the tacho-generator into a digital duty-cycle output signal according to the control function.

In the following experimental investigation, the circuit voltages were measured directly using Tektronix Digital Phosphor Oscilloscope (TDS3052), and the current was measured using a Tektronix current probe (P6302). The results for the steady-state analysis were evaluated for machine voltages \(E_e = 2.0, 4.0 \text{ and } 6.0 \text{ V}\), and normalised damping ratios \(F_{DAMP} = 0.1, 0.2, \ldots 0.9\), except for \(E_e = 6.0 \text{ V}, F_{DAMP} = 0.8, 0.9\) due to limitations in the power delivered by the supply. The circuit switching frequency was \(f_s = 10 \text{ kHz}\).

### 5.4.3. Continuous Conduction Analysis

The theoretical analysis in Section 5.3 was undertaken using the assumption that the electromagnetic interface was operating in a continuous current condition. For this condition, the internal inductance of the electromagnetic machine was sufficiently large such that there was always a positive inductor current flowing. An experimental investigation was undertaken to test this assumption.

Figure 5.11 shows the inductor current for the boundary between continuous and discontinuous conduction. In this situation, the minimum inductor current \(i_{L,min} = 0\).
Theoretically, the inductor current \( i_L(t) \) at the boundary between continuous and discontinuous conduction when the switch was closed could be given by,

\[
i_L(t) = \frac{E_C}{R_{\text{INT}}} \left( 1 - e^{-\left(\frac{R_{\text{INT}}/L}{\tau}\right)t} \right) \quad [A]. \tag{5.25}\]

In Section 5.3, it was assumed that the derivative of the inductor current for this part of the switching cycle was constant. This was valid provided that the \( R-L \) time constant of the electromagnetic machine was large compared to the switching period \( (T_s) \). For the experimental regenerative system described in Section 5.4.2, the electromagnetic damper \( R-L \) time constant was \( \tau = 2.17 \text{ ms} \) and the switching period was \( (T_s = 0.1 \text{ ms}) \). Therefore, this assumption was valid. As such, the average inductor current \( (i_{L\text{avg}}) \), was half the peak inductor current \( (i_{L\text{max}}) \), and could be given by,

\[
i_L(t) = \frac{E_C}{2L} DT_s \quad [A]. \tag{5.26}\]

By equating Equation 5.26, and the inductor current relationship given in Equation 5.7a, it was possible to give the open-circuit machine voltage \( (E_C) \), as a function of the duty-cycle for the boundary of continuous and discontinuous conduction. This relationship is given by,

\[
E_C = \frac{2L \left( V_{\text{BATT}} + V_D \right) \left( D - 1 \right)}{\left( R_{\text{INT}} + (D + 1)^2 R_{\text{BATT}} \right) DT_s - \left( 2L \right)} \quad [V]. \tag{5.27}
\]
An experimental analysis was undertaken to verify the relationship in Equation 5.27. In this analysis, the open-circuit machine voltage \( (E_c) \) was measured at the continuous-discontinuous current boundary. At the boundary, the inductor current reduced to zero \( (i_{L_{\text{min}}} = 0) \) for at least part of the switching cycle. An example waveform of the inductor current for \( (D = 0.5) \), at the boundary of continuous and discontinuous conduction is shown in Figure 5.12.

![Figure 5.12 Experimental Inductor Current (\( E_c = 3.29 \text{ V}, D = 0.5 \)).](image)

<table>
<thead>
<tr>
<th>Vertical Scale:</th>
<th>Ch 2 (( I_L ))</th>
<th>50.0 mA per Division.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal Scale:</td>
<td>20.0 ( \mu \text{s} ) per Division.</td>
<td></td>
</tr>
</tbody>
</table>

The results of the theoretical and experimental analysis are presented in Figure 5.13. Refer to Appendix B.4, Table B.7 for the theoretical and experimental data.
It can be seen that there was reasonable agreement between the theoretical and experimental analysis for the boundary of continuous current operation. It was shown, however, that the experimental system reached the boundary of discontinuous conduction for lower values of machine voltage than indicated in the theoretical analysis. This was due to the non-ideal experimental conditions such as switching losses. Further analysis of these non-ideal effects are discussed in the following sections.

The open-circuit machine voltage at the boundary of continuous conduction was compared with the damping control function given previously in Section 5.3.4, Equation 5.20. From this it was shown that for the interface to be operating in continuous conduction,

\[
F_{DAMP} \geq \frac{\left( R_{\text{INT}} + (D+1)^2 R_{\text{BATT}} \right) DT_s}{2 L}.
\]

(5.28)

For the experimental regenerative system described in Section 5.4.2, the normalised damping ratio should be greater than or equal to \( F_{DAMP} = 0.045 \). This ratio was
considered relatively small and, therefore, for this regenerative system, the assumption that the circuit was operating in a continuous conduction region was valid.

5.4.4. Transistor Source-Drain Resistance Analysis

The theoretical analysis in Section 5.3 was undertaken using the assumption that the 'on' and 'off' switch resistances approached zero and infinity Ohms, respectively. An experimental investigation was undertaken to test this assumption. The switch used in the experimental investigation was a power Field-Effect Transistor (F15N06L). The parasitic diode in parallel with this transistor will not influence the operation of this circuit, however. This was because the transistor source-drain voltage ($V_{DS}$), was always greater, or equal to, zero, which meant that the parasitic diode was always reverse-biased. The effective source-drain resistance was estimated from the instantaneous source-drain voltage ($V_{DS}$), and drain current ($I_{DRAIN}$) of the transistor. Refer to Appendix B.4, Table B.8 for the voltage and current measurements.

The experimental waveforms for the transistor source-drain voltage ($V_{DS}$) and drain current ($I_D$) for a normalised damping ratio ($F_{DAMP} = 0.5$), and machine voltage ($E_C = 4.0$ V) are shown in Figure 5.14. The average current and voltage measurements for the transistor 'on' state were evaluated using an oscilloscope 'gating' function. The gating function enabled average measurements to be recorded for a particular segment of the overall waveform.

The measured source-drain 'on' resistance is shown in Figure 5.15. It was found that the resistance was largely independent of the machine voltage, and reduced to approximately ($R_{DS} = 0.10$ Ohms) for a large normalised damping coefficient.
The effect of the transistor resistance was to add an effective series resistance of \( (R_{DS,ON}) \) to the internal machine windings in the 'On State'. This is shown in Figure 5.16.
The transistor drain current in the 'off' state was also experimentally measured to estimate the source-drain 'off' resistance. However, the measured current was negligible so the effect of the source-drain 'off' resistance could be ignored.

5.4.5. Switching Loss Analysis

The theoretical analysis in Section 5.3 was undertaken using the assumption that the 'transition time' between switching states was negligible, and that there was no switching delay. The specified maximum turn-on delay of the transistor was 0.37 (μs), which was 0.37 per cent of the total switching period. The theoretical maximum turn-off delay of the transistors was 0.80 (μs), which was 0.80 per cent of the total switching period. From these estimates the assumption that the switching transition-time and delay was negligible, was valid.

It was also possible that losses due to the transition between switching states of the switching device may affect the regenerative interface performance. Therefore, the transition losses were experimentally measured. Example waveforms for the transistor source-drain voltage, drain current and power dissipation for a normalised damping ratio ($F_{DAMP} = 0.5$), and machine voltage, ($E_{C} = 4.0$ V), are shown in Figure 5.17.
The transition loss was evaluated by measuring the average switching loss over the total switching period. The oscilloscope gating function was used to isolate the transition interval. In the example given in Figure 5.17 a), the average switching loss was 3.53 Watts (= 17.65mVV \times 0.2 \text{ W/mVV}) over 800 (ns) of the 100 (\mu s) switching interval. This led to an average switching transition loss of 28 (mW) over the total switching period. The measured transistor switching transition loss is shown in Figure 5.18. Refer to Appendix B.4, Table B.9 for the measured data.
The switching transition loss is included in the analysis of the interface output power in Section 5.4.7.

### 5.4.6. Diode Loss Analysis

Although the theoretical analysis accounted for the power dissipation in the diode due to the forward voltage drop \( V_D \) and the diode current \( I_{\text{DIODE}} \) in the forward bias condition, the analysis did not account for possible transition losses in the diode. To test this assumption, an experimental analysis was undertaken. The diode voltage and current was measured at the transition between forward and reverse bias conditions. Examples of the experimental waveforms for the diode voltage, current and power dissipation for a normalised damping ratio \( F_{\text{DAMP}} = 0.5 \), and machine voltage \( E_c = 4.0 \) V, are shown in Figure 5.19.
a) Reverse to Forward Bias.  

<table>
<thead>
<tr>
<th></th>
<th>Ch 1 ($V_D$)</th>
<th>5.00 V per Division.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ch 2 ($I_{DIODE}$)</td>
<td>1.00 A per Division.</td>
<td></td>
</tr>
<tr>
<td>Math ($V_D \times I_{DIODE}$)</td>
<td>5.00 W per Division.</td>
<td></td>
</tr>
</tbody>
</table>

**Horizontal Scale:** 400 ns per Division.

**Figure 5.19** Experimental Diode Voltage and Current ($E_c = 4.0$ V, $F_{DAMP} = 0.5$).

It was found that in some circumstances, there was a finite loss due to diode changing state. An example of this is illustrated in Figure 5.19 b) in which a loss occurred due to the transient diode voltage during the switching transition. However, the magnitude of the transition loss was negligible for the range of experimental conditions.

5.4.7. Regenerated Output Power

The regenerated power was measured to give an indication of the regenerative interface performance. The power was evaluated both experimentally and by simulation.

The experimental output power was measured using the average battery voltage and current. Refer to Appendix B.4, Table B.10 for the experimental voltage and current measurements. Examples of the experimental waveforms for the battery voltage and current for the normalised damping ratio ($F_{DAMP} = 0.5$), and machine voltage ($E_c = 4.0$ V), are shown in Figure 5.20.
The simulation of the regenerative interface was undertaken using the MicroSim PSpice (Evaluation Version 6.3) circuit simulator. The circuit was simulated to analyse the effect of the transistor 'on' resistance, $R_{DS}$. The resistance values were taken from the experimental measurements in Section 5.4.4. Refer to Appendix B.7, Table B.21 for the simulation file used for this analysis, and Appendix B.4, Table B.11 for the simulation data. The output power for the theoretical, simulation and experimental analysis of the regeneration circuit is shown in Figure 5.21. The experimental results also include the transistor transition losses from Section 5.4.5. Error bars are included for the graphical representation of the experimental data. The error bars indicate the confidence limits as governed by the accuracy of current probe (P6302) and oscilloscope (TDS3052), which are given in Appendix B.2, Table B.4. The uncertainty of the output power ($U_{\text{out}}$) is given by [61],

$$U_{\text{out}} = \left( \frac{\partial P_{\text{out}}}{\partial V_{\text{batt}}} U_{V_{\text{batt}}} \right)^2 + \left( \frac{\partial P_{\text{out}}}{\partial I_{\text{batt}}} U_{I_{\text{batt}}} \right)^2 \quad [\text{W}], \quad (5.29)$$

where: $U_{V_{\text{batt}}}$ = uncertainty of the battery voltage, and $U_{I_{\text{batt}}}$ = uncertainty of the battery current.

\footnote{For the condition that the transistor 'on' resistance was zero, the output power measurements from the simulation analysis were consistent with the theoretical predictions.}
a) $E_c = 2.0$ Volts

b) $E_c = 4.0$ Volts
The results indicate that the output power predicted by the simulation was lower than the ideal theoretical analysis. The results indicated that up to 0.5 Watts (approximately 6.5 per cent of the output power) was lost due to the transistor 'on' resistance for a machine voltage \(E_C = 6.0\) V, and normalised damping ratio \(F_{DAMP} = 0.5\).

The results also indicated that there was reasonable agreement between the experimental and simulation output power results. The results were within the experimental uncertainty, except for larger values of the normalised damping ratios for all three machine voltages. It is believed that power loss may have been attributed to mechanical losses (friction) in the gear and bearing of the electromagnetic damper and drive motor. These losses would be more prominent for larger normalised damping ratios due to the larger forces produced by electromagnetic damper and drive motor under these conditions. It was not practical to further investigate the power associated with the gear and bearing system due to the difficulty of isolating these losses, however.
5.4.8. Damping Performance

The damping response was measured to give an indication of the regenerative interface performance. This was analysed by measuring the force-velocity (torque-angular velocity) characteristic of the electromagnetic damper.

To evaluate the effective damping response of the regenerative damping system, the force produced by the damper (for a given machine velocity) was compared with the force corresponding to the maximum damping coefficient. The open-circuit machine voltage was used as a measure of the angular velocity of the damper. The force (or torque) produced by the electromagnetic damper was evaluated by measuring the current in the driving machine ($I_D$). Refer to Appendix B.4, Table B.12 for the drive-current measurements. The driving machine current was proportional to the torque produced by the driving machine (via the machine torque constant, $K_E$). The torque produced by the driving machine was proportional to the torque produced by the electromagnetic damper (via the gear mechanism). For this analysis, it was not necessary to evaluate all the constants-of-proportionality to determine the relative damping performance.

The 'stall' current ($I_{STALL}$) was the current required to overcome friction in the system, without contributing to the force produced by the electromagnetic damper. The stall current was subtracted from the current measurements to obtain an effective drive current ($I_{D,EFF} = I_D - I_{STALL}$). The stall current was defined as the maximum current in the driving machine for zero machine velocity, and in this experimental analysis ($I_{STALL} = 0.23$ A).

Equation 5.30 gives the relationship between the system parameters and the normalised damping coefficient of the electromagnetic damper ($C_{EFF}$). The parameter ($R_{IE}$), was the current-voltage ratio for maximum damping, and was evaluated as ($R_{IE} = 1.39$ V/A).

$$C_{EFF} = \left( I_{D,EFF} \cdot \frac{R_{IE}}{E_C} \right).$$ (5.30)
The damping coefficient was compared against the maximum damping condition, which was the condition that the damper terminals were short-circuited. The damping coefficient ratio results are shown in Figure 5.22. Refer to Appendix B.4, Table B.13 for the normalised damping coefficient measurements.

![Figure 5.22 Normalised Damping Coefficient of the Electromagnetic Regeneration System.](image)

The results indicated that the damping coefficient reasonably agreed with the theoretical estimate. The damping coefficient was generally in the order of 0.85 to 0.90 of the theoretical damping coefficient. This may be attributed to the effective transistor 'on' resistance \( R_{ds} \), evaluated in Section 5.4.4. This resistance was generally around \( R_{ds} = 0.10 \) to \( 0.15 \) Ohms. A series resistance of this magnitude (relative to the internal machine resistance of \( R_{int} = 1.06 \) Ohms) corresponded to an damping coefficient reduction to between 0.87 to 0.91 of the theoretical estimate.
5.4.9. Power Efficiency

The power efficiency, or the regeneration efficiency, of the impedance-matching regenerative damping system was also experimentally evaluated. The power efficiency was previously given in Equation 5.21, and was defined as the ratio of output power \( P_{\text{OUT}} \), to the input power \( P_{\text{IN}} \), into the system. In the previous theoretical analysis, the input power \( P_{\text{IN}} \), was attributed to the machine voltage \( E_c \), and current \( i_i \) via \( P_{\text{IN}} = E_c i_i \). However, in the experimental analysis, it was difficult to isolate the machine voltage and current. Therefore, the input power was derived from the power supplied from the driving machine. This is illustrated in Figure 5.23.

![Figure 5.23 Driving Machine and Electromagnetic Damper Gear System.](image)

The input power at the electromagnetic damper is given by,

\[
P_{\text{IN}} = \tau_M \omega_M \quad [\text{W}], \text{ where}
\]

\[
\tau_M = \alpha \tau_D \quad [\text{Nm}], \text{ and}
\]

\[
\tau_D = K_{TD} i_{D-EFF} \quad [\text{Nm}].
\]

The torque constant of the drive motor was \( K_{TD} \), and \( \tau_D \) and \( \tau_M \) were the torque of the drive machine and electromagnetic damper, respectively. \( \omega_D \) and \( \omega_M \) were the
angular velocity of the drive machine and electromagnetic damper, respectively. The electromagnetic damper velocity is given by,

$$\omega_m = \frac{1}{K_{EM}} E_C \text{ [rads}^{-1}\text{]}, \quad (5.34)$$

where \((K_{EM})\) was the voltage constant of the electromagnetic damper. From Equations 5.31 to 5.34, the input power \((P_{IN})\), is given by,

$$P_{IN} = \alpha \frac{K_{TD}}{K_{EM}} E_C i_{D-EFF} \text{ [W].} \quad (5.35)$$

The output power is defined by,

$$P_{OUT} = V_{BATT} i_{BATT} \text{ [W].} \quad (5.36)$$

In this analysis, the gear ratio \((\alpha = 2.25)\), \((K_{TD} = 0.105 \text{ Nm/A})\), and \((K_{EM} = 0.155 \text{ Vs/rad})\). From Equations 5.21, 5.35 and 5.36, the power efficiency of the electromagnetic regenerative interface was evaluated. Refer to Appendix B.4, Table B.14 for the power efficiency measurements. The results are shown graphically in Figure 5.24.

The results in Figure 5.24 indicate a general agreement between the experimental and theoretical power efficiency estimates. It can be seen that the efficiency, generally, increased inversely to the normalised damping ratio \((F_{DAMP})\), and was largely independent of the machine voltage \((E_C)\). The power efficiency was generally in the order of 65 to 80 per cent of the theoretical estimate. This may be attributed to several factors. The transistor 'on' resistance \((R_{DS})\), would consume a portion of the input power. This resistance was evaluated in Section 5.4.4, and the effect on the output power was previously shown in Figure 5.21. The power efficiency would also be reduced due the mechanical losses from both the drive motor and electromagnetic damper. Although the stall current was included in this analysis, the dynamic losses due to bearings and gears was not included and would reduce the overall power efficiency.
Figure 5.24 Power Efficiency of the Electromagnetic Regeneration System.
5.5. Dynamic Experimental Analysis

5.5.1. Overview

The purpose of this section is to document an experimental dynamic analysis of the regenerative interface.

5.5.2. Experimental Apparatus

A photograph and schematic diagram of the dynamic experimental apparatus are shown in Plate 5.2 and Figure 5.25, respectively.

Plate 5.2 Photograph of the Dynamic Experimental Apparatus.
For the dynamic analysis, the electromagnetic damper was subjected to a sinusoidal disturbance within a one degree-of-freedom dynamic system. The dynamic system consisted of a spring and damper connected in parallel between an input disturbance and a sprung mass. There were several reasons for using the particular dynamic arrangement. This included its simplicity, and the ability to isolate the damping characteristics of the dynamic system. This model had also been used by other research groups in the field of suspension, and dynamic system design [18-20, 26]. The dynamic system parameters are given in Table 5.3.

<table>
<thead>
<tr>
<th>Description</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mechanical Spring</td>
<td>$k_v$</td>
<td>1.625 (kN/m)</td>
</tr>
<tr>
<td>Trolley (Sprung Mass)</td>
<td>$m_v$</td>
<td>21.19 (kg)</td>
</tr>
<tr>
<td>Natural System Frequency</td>
<td>$f_n$</td>
<td>1.39 (Hz)</td>
</tr>
</tbody>
</table>

Table 5.3 Experimental Dynamic System Specifications.

The electromagnetic damper was connected to a tacho-generator, and was controlled via an interface (I/O) card from a personal computer. In this analysis, the tacho-generator voltage provided both velocity magnitude and polarity (direction) information. A software algorithm converted the tacho-generator voltage into duty-cycle and polarity output signals. Similar to the steady-state analysis the control function previously evaluated in Section 5.3.4 (Equation 5.20) was used. This control function gave the output duty-cycle ($D$), as a function of the open-circuit machine voltage ($E_C$). In this
analysis a full-bridge regenerative circuit as previously shown in Figure 5.2 was used due to the bi-directional damper velocity. The duty-cycle and polarity signals were then converted (by digital logic) into four digital signals to control the switching of the power transistors.

In the following analysis, it was assumed that the current flowed through the body of the switching transistor, rather than the parallel, parasitic diode. Although the transistor source-drain voltage \( V_{DS} \) may be less than zero for part of the switching cycle\(^3\), this assumption was valid, provided that the magnitude of the source-drain voltage due to this reverse conduction was low enough such that the parasitic diodes did not conduct. If the diodes did conduct, however, the operation of the interface would be influenced due to the relatively low reverse recovery time of the diodes and would, subsequently, reduce the accuracy of the present analysis.

The power transistors were driven through two 'IR2110' high and low side drivers, which provided a floating gate voltage relative to the electromagnetic machine terminals. The circuit switching frequency was \( f_s = 10 \text{ kHz} \). Refer to Appendix B.2, Figure B.3 for a detailed circuit diagram, and Tables B.1, B.2 and B.3 for the component specifications. The personal computer, electromagnetic damper (Electro-craft Servo Products, S-19-3B), tacho-generator (Mabuchi, RS-540 SH) and storage battery (CNB, B38-6A) were the same as for the experimental steady-state analysis. The other component parameters and descriptions for the dynamic analysis (that were not used in the steady-state analysis) were two Spectrol Rotary Potentiometers, and a LEM Current Transducer in which the component specifications are given in Appendix B.2, Table B.2.

For the dynamic analysis, the experimental data was logged using the personal computer. This data included the input displacement \( (x_0) \), and sprung mass displacement \( (x_v) \) (using the Rotary Potentiometers) and battery current (using the 'LEM' Current Transducer), battery voltage, and the tacho-generator voltage. The data was

\(^3\) The source-drain voltage, \( V_{DS} \) was always greater, or equal to, zero for the previous, steady-state analysis documented in Section 5.4.
analysed using the mathematical software program MATLAB® for evaluating the average output power and dynamic response.

The input excitation was a constant-amplitude input sinusoidal displacement. In this analysis, the input amplitude was 28.4 (mm). The frequency varied continuously for a period of 100 (s), and for a frequency-sweep range of 1.0 to 2.2 (Hz). The sweep was undertaken for a period long enough, such that the effects of any transient response were negligible. The experimental parameters were sampled at a rate ($f = 30$ Hz). The frequency response of the dynamic system was analysed using the Fast-Fourier-Transform (FFT) function of the input excitation displacement ($x_o$), and the sprung mass displacement ($x_v$).

5.5.3. Regenerated Output Power

Another important parameter, for the dynamic analysis of the impedance-matching regenerative interface, was the regenerated power. In this experiment, the average output power was measured as a function of frequency. This was achieved by multiplying the battery current (using the LEM current transducer), and battery voltage, for a range of input frequencies, and for a constant input amplitude. Refer to Appendix B.5, Table B.15 for the output power measurements. The results of this analysis are also shown in Figure 5.26. In this analysis the amplitude of the input excitation was 18.9 (mm).
The output power results in Figure 5.26 indicate that the output power increased for frequencies up to the natural frequency of the dynamic system ($f_n = 1.39$ Hz). This was because the amplitude and frequency of the relative damper velocity increased in this range. The output power reduced to an almost constant level for frequencies between the natural frequency and up to 2.2 (Hz). This was because the amplitude of relative velocity was reducing in this range, which was counteracted by an increase in the frequency of excitation. Another aspect of the results, shown in Figure 5.26, was that the output power generally increased for lower normalised damping ratios. The reason for this was that the relative amplitude of the damper velocity increased as the normalised damping was reduced.

There were several, non-ideal, effects that may have modified the experimental results given in this section. One effect, that was mentioned, was due to friction in the system. The forces due to friction were minimised through the use of low-friction bearings and lubrication in the system. Rotary bearings were used on the trolley wheels, and a linear bearing was used for supporting the rack mechanism. Another method used to minimise the friction effects was to maximise the input amplitude. The amplitude was increased up to the allowable travel range of the trolley and rotary potentiometer. By doing this,
the relative damper velocity and, therefore, the damping forces were maximised. This meant that the friction forces, relative to the damping forces, were minimised.

The force produced by the rotary potentiometers had a slight effect on the forces generated in the dynamic system. However, this force was considered insignificant in comparison to the spring and damper forces in the system. The maximum starting torque of the potentiometers was $3.6 \times 10^{-4}$ (kg.m). The gear that was attached to the potentiometer had a diameter of 14.0 (mm), which referred to a maximum linear force (via a rack attached to the trolley) of approximately 0.25 (N). Considering that the forces produced by the damper and spring, were in the order of 10 to 100 (N), the potentiometer force was not considered to have a large effect on the system response.

Other non-linear effects, may have included a non-ideal spring response. To test this, the spring force-displacement response was measured using a Shimadzu, Universal Testing Machine (DSS-25T). For a displacement range exceeding the experimental conditions, the spring coefficient was linear within the accuracy of the testing machine accuracy of ±1.0 per cent.

5.5.4. Damping Performance

An experimental analysis of the dynamic damping performance was undertaken. This was undertaken by analysing the Bode-diagram of the one degree-of-freedom system. Figure 5.27 shows the dynamic response of the electromagnetic damper and impedance-matching interface as a function of input frequency. The dynamic response was measured for a normalised damping ratio ($F_{\text{DAMP}} = 0.2, 0.4, 0.6, 0.8$), and the maximum damping condition for zero external resistance (short-circuit condition).
From the dynamic response, shown in Figure 5.27, it was possible to estimate the actual damping coefficient of the damper. This was undertaken by comparing the maximum the experimental response ($\chi_{v}/\chi_{o}$) against an ideal, one degree-of-freedom, theoretical model. The estimate of actual damping compared to the maximum (short-circuit condition) is given in Table 5.4.

It can be seen, from Table 5.4, that the estimated damping coefficient varied from the ideal response. For the normalised damping ratios, ($F_{\text{DAMP}} = 0.6$ and $0.8$), the experimental damping reasonably agreed with the theoretical damping response (62.5 per cent instead of 60.0 per cent, and 76.0 per cent instead of 80.0 per cent, respectively). However, for the lower damping ratios of ($F_{\text{DAMP}} = 0.2$ and $0.4$) the experimental damping was higher than the theoretical estimate (40.6 per cent instead of 20.0 per cent, and 48.9 per cent instead of 40.0 per cent, respectively).

<table>
<thead>
<tr>
<th>$F_{\text{DAMP}}$</th>
<th>0.2</th>
<th>0.4</th>
<th>0.6</th>
<th>0.8</th>
<th>Short-Circuit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured Damping Coefficient (Ns/m)</td>
<td>39</td>
<td>47</td>
<td>60</td>
<td>73</td>
<td>96</td>
</tr>
<tr>
<td>Proportion of Maximum Damping (%)</td>
<td>40.6</td>
<td>48.9</td>
<td>62.5</td>
<td>76.0</td>
<td>100.0</td>
</tr>
</tbody>
</table>

Table 5.4 Experimental Dynamic Damping Properties.
The damping force (produced by the damper) was smaller for the low normalised damping ratios and, therefore, the effect of friction (due to the linear bearings and wheel friction) was more significant for the lower damping ratios. The effect due to sliding, or coulomb friction remained approximately constant for a given system velocity [62]. Therefore, the friction force would have altered the "effective" system damping coefficient more significantly at lower normalised damping ratios, and this would have contributed to the higher than expected damping force measured at low normalised damping ratios.

5.6. Comparison with Previous Regenerative Interface Designs

5.6.1. Overview

The analysis documented in this section compared the damping and regeneration performance of the impedance-matching regenerative system to previous regenerative interface designs. The aim was to improve the understanding of the relative performance of the impedance-matching interface. There had been two main regeneration interface designs previously proposed for regenerative vibration damping using electromagnetic devices. These two designs were reviewed in the Literature Review, Section 2.2.3.

One design was proposed by Suda et al. [22] for the purpose of self-powered active vibration control. The system used two electromagnetic DC motors in a two degree-of-freedom suspension system. One motor was placed in the primary suspension and transferred energy to a storage capacitor using relay switches. The capacitor could supply energy to the second electromagnetic motor, which provided active control. The regeneration circuit is shown in Figure 5.28.
Due to the frequency limitations of relay devices (typically in the order of milliseconds, rather than nanoseconds for semiconductor switches) it was believed that a form of duty-cycle switching was not used by Suda et al. [22]. The main objective of this regenerative system was for active control, rather than energy regeneration. As such, a comparison of this system performance was not undertaken.

A second form of regenerative system was also proposed as a method of providing regenerative vibration control. This system was proposed by Okada and Harada [19] for use in active suspension systems. This regenerative interface design had the objective of providing active control, as well as energy regeneration. Figure 5.29 shows the 'double-voltage' regenerative interface used to regenerate electrical energy during high speed motion of the electromagnetic damper.

The investigation documented in this section compared the impedance-matching interface design from this research to the 'double-voltage' interface design, shown in Figure 5.29. In this analysis, however, a full-bridge rectifier circuit was tested, instead of the 'double-voltage' interface. The difference between the designs was that a series of
two diode drops, instead of one, was necessary for the full-bridge design. However, the advantage of the full-bridge interface was that only one battery was needed. The passive, 'full-bridge' rectifier interface is illustrated in Figure 5.30.

![Figure 5.30 Passive Rectifier Interface Design.](image)

The experimental analysis of the regenerative damping systems was performed using steady-state (Section 5.6.2) and dynamic (Section 5.6.3) analysis. The electromagnetic damper (Electro-craft Servo Products, S-19-3B) and the storage battery (CNB, B38-6A) was the same as for the previous experimental analysis. The diodes used for the 'full-bridge', passive interface circuit were Fairchild Semiconductor, Model No. 1N5400, and the specifications are described in Table 5.5.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Rectified Current</td>
<td>3.0</td>
<td>A</td>
</tr>
<tr>
<td>Max. Forward Voltage Drop (@ 3.0 A)</td>
<td>1.2</td>
<td>V</td>
</tr>
<tr>
<td>Maximum Reverse Current (@ 25° C)</td>
<td>5.0</td>
<td>μA</td>
</tr>
</tbody>
</table>

Table 5.5 Regenerative Interface Diode Specifications (Fairchild Semiconductor 1N5400).

5.6.2. Steady-State Analysis

As previously stated, both the energy regeneration and damping performance of the impedance-matching regenerative interface and the passive 'rectifier' interface were compared. The steady-state results were obtained using the same experimental apparatus
used for the previous steady-state analysis (Section 5.4), in which the experimental apparatus was previously shown in Plate 5.1 and Figure 5.10.

The output power \( P_{\text{OUT}} \) as a function of the open-circuit machine voltage \( E_c \) is shown in Figure 5.31. The results of the output power from the passive 'rectifier' interface, and the impedance-matching regeneration interface at \( F_{\text{DAMP}} = 0.3, 0.5, \) and \( 0.7 \) are given. Refer to Appendix B.6, Table B.16 for the passive rectifier interface output power measurements. The impedance-matching regenerative interface power analysis was given previously (Section 5.4.7). Refer to Appendix B.6, Table B.17 for the measured data.

![Figure 5.31 Passive and Impedance-Matching Interface Output Power Comparison.](image)

It can be seen from Figure 5.31 that the impedance-matching regenerative interface had a greater output power than the passive interface for the normalised damping ratios \( F_{\text{DAMP}} = 0.3, 0.5 \) and \( 0.7 \), and machine voltages (at least) up to \( E_c = 8.0 \) V). The main difference between the two systems was the ability of the impedance-matching regeneration interface to transfer power to the storage battery for the condition that the machine voltage was less than the battery potential \( V_{\text{BATT}} = 6.20 \) V). The passive
interface did not transfer any power to the storage battery for this condition because the diodes were reverse-biased.

The relative damping performance between the impedance-matching regeneration interface, and the passive interface was also determined. The analysis was performed by measuring the current of the driving machine to determine the force produced by the electromagnetic damper. The derivation of this analysis has previously been undertaken, and was shown in Section 5.4.8. Refer to Appendix B.6, Table B.18 for the drive current \( I_d \), and damping coefficient ratio \( C_{EF} \), measurements for the passive rectifier circuit.

Figure 5.32 shows the relative damping performance between the impedance-matching regeneration interface, and the passive 'rectifier' interface. The impedance-matching interface drive current and normalised damping coefficient are given in Appendix B.6, Table B.19 and Table B.20, respectively. It can be seen that the impedance-matching regeneration interface had a greater damping performance compared to the passive interface for the normalised damping ratios \( F_{DAMP} = 0.3, 0.5 \) and 0.7), and machine voltages up to \( E_C = 10.0 \text{ V} \).

![Figure 5.32 Passive and Impedance-Matching Interface Damping Comparison.](image-url)
The damping of the passive interface was approximately 5 per cent of maximum damping when the machine voltage was lower than the battery potential. Because there was no current flowing in the machine windings for this condition, the 'damping' force was caused by mechanical friction in the damping system. It can be seen that the relative damping coefficient ratio ($C_{\text{Eff}}$), of the passive regeneration interface increased for machine voltages above ($E_C = 8.0$ V). This was because the diodes became forward-biased and allowed both the battery current ($I_{\text{BATT}}$), and the current in the damper windings to flow. The current flowing in the damper windings then allowed an electromagnetic damping force to be produced for voltages above ($E_C = 8.0$ V).

The problem of regeneration for low damper voltages (and, therefore, velocities) was acknowledged by Okada and Harada [19], and was described previously (Section 2.2.3). In this investigation, Okada and Harada used passive damping to overcome this problem by switching the damper terminals to a resistance for low actuator velocities. Okada et al. [20] continued the investigations by Okada and Harada [19], and introduced an electric resonant circuit to improve the energy regeneration for low damper velocities.

5.6.3. Dynamic Analysis

To determine the relative performance between the impedance-matching, and passive regenerative interfaces, a comparison between the dynamic response of the two systems was evaluated. For this experiment, the sinusoidal input displacement had an amplitude of 18.9 (mm). Figure 5.33 shows the relative damping performance of the two systems.
Figure 5.33 Passive and Impedance-Matching Interface Isolation Response Comparison.

The dynamic response of the impedance-matching interface was different than for the previous dynamic analysis shown in Figure 5.27. This was because a lower input displacement amplitude was used for the dynamic analysis documented in this section. The results of dynamic experimental comparison indicated that the damping coefficient of the system with the passive circuit was approximately 23 (Ns/m), or 18.4 per cent of the maximum possible damping coefficient of the electromagnetic damper.

For the passive circuit, the maximum amplitude of the relative damper displacement was 151 (mm) at a frequency of 1.38 (Hz). This referred to a maximum damper velocity of 1.30 (m/s). For the machine voltage constant of \( (K_E = 0.155 \text{ Vs/rad}) \), and a gear diameter of 54 (mm), this referred to an open-circuit machine voltage of \( (E_C = 7.46 \text{ V}) \). This voltage was lower than the potential barrier (battery voltage and forward diode voltage) of 7.90 (V). Therefore, under these conditions, there was no current flowing in the circuit and, therefore, no regeneration and no damping. The effective damping coefficient indicated in Figure 5.33, occurred due to the friction in the system. For the experimental dynamic analysis in this section, it was not practical to further increase the input amplitude and, therefore, the relative damper velocity. This was because the system was operated at the displacement limits of the rack-and-pinion, and rotary potentiometer.
An experiment was also undertaken to determine the output power of the impedance-matching interface, compared to the passive design. However, because the passive design was operating below the barrier potential, no output current and, therefore, no output power was measured for the dynamic analysis.

An important aspect of the comparison between the passive and impedance-matching interfaces was due to the potential of the storage battery used, as well as the passive circuit used. In the experimental investigation undertaken by Okada and Harada [19], the battery potential was \( V_{\text{BATT}} = 1.26 \text{ V} \), in comparison to \( V_{\text{BATT}} = 6.20 \text{ V} \) used in this analysis. Also, there was only one diode potential-drop between the damper and battery in the 'double-voltage' regeneration interface, in comparison with two diode potential-drops for the regeneration interface examined in this analysis. This would have had a large influence on the output power and damping performance of the passive regeneration interface, due to the ability for regeneration and at much lower machine velocities.

There were also several differences in the objectives of previous research investigations and the research documented in this thesis. In the research investigations by Okada and Harada [19], and Suda et al. [22], the main objective was to improve damping performance by using semi-active or active control; the process of energy regeneration was a secondary effect that assisted the damping performance. The difference of this research was that the objective was to maximise energy regeneration and maintain adequate vibration control. It was understood, therefore, that the relative differences in damping and regeneration performance, indicated that one particular system would have been more suited to a particular application than another, rather than one system having better overall performance.

As mentioned, at the end of Section 5.3.3, the impedance-matching regenerative interface approximated the passive circuit shown in Figure 5.29 for large input voltages \( E_C \). This was because, for this condition, the control function led to a duty-cycle \( D = 0 \), which resulted in the switch (transistor) being in an open-circuit condition.
Therefore, the utilisation of the impedance-matching interface, as opposed to the passive design, will depend on the trade-off between complexity and efficiency.

5.7. Conclusions

The analysis in this chapter investigated a new form of regenerative interface, the impedance-matching interface, for the purpose of electromagnetic regenerative vibration damping. The interface had several potential advantages over previous designs, especially with respect to the damping and regeneration performance for low damper velocities. The results also suggested the operation of the interface was largely independent of the storage battery potential and, therefore, had an advantage for applications such as regenerative vehicle damping, in which large battery voltages (such as 14 or 42 Volt systems) were expected.

A theoretical investigation was undertaken to determine the overall performance and operation of the system. This analysis revealed that, with relatively simple control, it was possible to control the interface to either maximise energy regeneration, or control the damping coefficient of the damper. An experimental investigation was also undertaken to determine the performance of the actual system and compare the performance against a previous design. It was revealed that the actual response reasonably followed the ideal theoretical model. However, it was also found that resistive circuit elements and non-ideal switching affected the damping and regeneration performance of the system. The results also indicated that the performance of the impedance-matching interface was superior to the previous, passive design for the particular experimental conditions.

It should be noted that the overall analysis presented in this chapter had certain limitations. The theoretical representation of the regenerative interface was based on an ideal model, which did not precisely represent the realistic system. Although the
objective of the subsequent experimental analysis was to determine the limitations of this model, it was only undertaken for a specific set of system parameters. The results also indicated that there were certain discrepancies between the experimental and simulation results (for example, the power measurements in Figure 5.21). The result of this is that, although the results were sufficient for the initial developmental analysis, further research is required to fully evaluate the proposed interface device.

Although, the results had certain limitations, when combined with the analysis in the previous chapter analysing the design of electromagnetic devices, they assisted in determining the overall performance of regenerative vehicle suspension. The results of this chapter are, therefore, directly used in Chapter 7, which provides the cost-analysis of regenerative vehicle suspension.
CHAPTER 6

VEHICLE INTEGRATION OF REGENERATIVE DAMPING

"No data yet," he answered. "It is a capital mistake to theorize before you have all the evidence. It biases the judgement."\(^1\)

Sherlock Holmes to Watson

6.1. Overview

This Chapter presents an analysis of two issues which have important implications for determining the performance of regenerative damping in vehicle systems. The first relates to determining the amount of energy dissipated in vehicle suspension systems and the second relates to the use of rotating dampers in vehicle suspension systems. This investigation was undertaken because these issues had

remained unresolved from previous research in the field, and because of the importance that these issues had on fulfilling the overall research objectives.

The performance of regenerative vehicle damping is largely dependent on the amount of energy dissipated in conventional vehicle suspension. Generally, the more energy that is dissipated, the higher the performance. This is because there will be a larger overall gain to the vehicle system from the regeneration process. It is important to determine the effect of energy dissipation due to the factors of vehicle mass, vehicle velocity and road surface condition. This will assist in determining how these factors affect the regeneration system performance.

The performance of regenerative vehicle damping is also largely dependent on the use of rotating dampers in vehicle suspension. As identified both in the Literature Review (Section 2.3) and the analysis of electromagnetic devices (Section 4.5.5), the major advantage of using rotating dampers occurs due to the mechanical amplification of the damper displacement. However, as shown by Ryba [26], for a one degree-of-freedom suspension model, rotating dampers also affect the vehicle dynamics. The analysis in this chapter (Sections 6.3, 6.4 and 6.5) investigates the use of rotating dampers with the objective of determining the limitations of their use in vehicle suspension.

The material covered in this chapter relates to the integration of regenerative damping in vehicle systems and, therefore, differs from previous thesis chapters. The previous chapters documented the investigation into the operation of regenerative dampers, which included both the analysis of electromagnetic dampers (Chapter 4) and the analysis of the regenerative energy interface (Chapter 5). The purpose of the investigation in this chapter was to build on the previous findings, and to focus on those issues that were important to the integration of regenerative damping in vehicle systems.
6.2. Suspension Energy Dissipation

In the Literature Review (Section 2.4), a survey of previous investigations into energy dissipation in vehicle suspension systems was undertaken. A review of an experimental investigation by Browne and Hamburg [10] found that approximately 40 to 60 (W) of power were dissipated in all four vehicle dampers for the vehicle travelling on good to average road surfaces. Also documented, in the Literature Review chapter, were several theoretical investigations in the field of vehicle suspension loss. In the theoretical analysis, however, the research objective of the authors was to identify the vehicle resistance to motion due to losses in the tire and suspension systems, rather than identify the amount of energy dissipated in vehicle dampers.

With respect to the objectives of this research program, it was considered important to theoretically analyse energy dissipation in vehicle suspension systems and compare the results with the experimental results undertaken by Browne and Hamburg [10]. A comparison of the results would have revealed any limitations of either the theoretical or experimental investigations. Any inconsistencies would indicate that either, the theoretical models were inaccurate or too simplistic, or that there were inaccuracies in the experimental investigations. A comparative analysis would also reveal trends in the relationship between energy dissipation and factors such as vehicle velocity, and road surface condition; trends that were not as easy to determine for limited experimental data. Therefore, in Section 6.2.1, the developed theoretical model is presented to show the analysis of energy loss in a vehicle suspension. Following this, in Section 6.2.2, a comparison between the theoretical model of Section 6.2.1, and previous experimental investigations is presented.
6.2.1. Theoretical Vehicle and Road-Surface Models

The theoretical analysis of vehicle suspension typically involves the modelling of two systems; the vehicle system and the road surface. The vehicle system model chosen for this analysis was the two degree-of-freedom, quarter car model. This model contained the most basic features of a real situation, and was chosen because it provided the most general and useful automotive suspension system design information [41]. It possessed particular advantages over more complex models in terms of:

- Being described by few design parameters,
- Having only a single input, leading to ease of computation of performance and ease of application, and
- Ease of mapping and understanding of the relationships between design and performance [41].

Figure 6.1 shows the two degree-of-freedom, quarter car model and the vehicle parameters. The parameters were based on an experimental vehicle model and are derived in Section 6.2.2. The sprung mass ($mv$), represented quarter of the vehicle mass, and the unsprung mass ($mt$), represented the mass of the tire and axle system. The vehicle suspension was represented by a viscous damper ($cv$), and suspension spring ($kv$), connected in parallel between the sprung and unsprung mass. The interaction between the tire and road surface was represented by spring ($kt$), and damper ($ct$), combination in parallel.
The road surface model used in this analysis provided the input to the dynamic vehicle model. Previous measurements of road surfaces had revealed that the road profile could rationally be described in statistical terms [33, 63]. From measurements from a large number of roads it was found that the vertical amplitude could be represented by a stationary, Gaussian random process and, therefore, could be described by a mean-square spectral density function \( S_{x_0}(w) \) [30, 63]. According to a proposed ISO Standard (1972) [30] the road profile, as a function of temporal frequency, could be represented by the elevation spectrum as shown in Equation 6.1,

\[
S_{x_0}(\omega) = \frac{2\pi V}{\omega^2} A \quad [\text{m}^2/\text{rad}], \tag{6.1}
\]

where: 
\( V = \) vehicle velocity, and 
\( A = \) roughness coefficient (as defined in Table 6.1).

<table>
<thead>
<tr>
<th>Road Surface Classification</th>
<th>Roughness Constant, ( A \times 10^{-4} ) (m. cycle)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very Good</td>
<td>0.001 &lt; ( A &lt; 0.002 )</td>
</tr>
<tr>
<td>Good</td>
<td>0.002 &lt; ( A &lt; 0.0095 )</td>
</tr>
<tr>
<td>Average</td>
<td>0.0095 &lt; ( A &lt; 0.035 )</td>
</tr>
<tr>
<td>Poor</td>
<td>0.035 &lt; ( A &lt; 0.15 )</td>
</tr>
<tr>
<td>Very Poor</td>
<td>0.15 &lt; ( A &lt; 0.55 )</td>
</tr>
</tbody>
</table>

Table 6.1 Road Surface Classification [30].

To determine the power dissipated in the vehicle suspension system, a similar analysis to the investigation by Segel and Lu [30] was undertaken. Segel and Lu [30] evaluated
the motion resistance acting on the vehicle, which was attributed to the vehicle suspension system. For this investigation, the power dissipation in the vehicle damper, rather than motion resistance, was determined. The power dissipation \( P_{\text{Diss}} \), in each vehicle damper was given by,

\[
P_{\text{Diss}} = cv \overline{V_{\text{REL}}}^2 \quad [W], \tag{6.2}
\]

where:

\[
\overline{V_{\text{REL}}}^2 = \text{mean-square, relative velocity across the damper, and}
\]

\[
cv = \text{damping coefficient of the vehicle damper.}
\]

The mean-square damper velocity was determined by integrating the spectral density of the relative damper velocity \( S_{\text{REL}}(\omega) \) over the frequency range of operation [41]. Therefore, the power dissipation was also given by,

\[
P_{\text{Diss}} = cv \frac{1}{\pi} \int_{0}^{\infty} \omega^2 |H_{x_{rel}/x_o}(j\omega)|^2 S_{x_o}(\omega) d\omega \quad [W]. \tag{6.3}
\]

From Equation 6.3, and the theoretical vehicle model (from Figure 6.1), the power dissipation with respect to the spectral density function of the road surface was given by,

\[
P_{\text{Diss}} = cv \frac{1}{\pi} \int_{0}^{\infty} \omega^2 |H_{x_{rel}/x_o}(j\omega)|^2 S_{x_o}(\omega) d\omega \quad [W], \tag{6.4}
\]

where \( H_{x_{rel}/x_o}(j\omega) \), \( s = j\omega \) was the transfer function relating the relative damper displacement to the input roadway displacement, and was given by,

\[
H_{x_{rel}/x_o}(s) = \frac{(ct mt) s^3 + (kt mv) s^2}{(mt mv) s^4 + (cv mt + ct mv + cv mv) s^3 + (ct cv + kv mt + kt mv + kv mv) s^2 + (cv kt + ct kv) s + (kt kv)}. \tag{6.5}
\]

Using Equations 6.1, 6.4, and 6.5, the damper power dissipation was evaluated numerically for the range of road surface conditions and vehicle velocities. The road surface was defined according to Equation 6.1, and the vehicle velocity ranged from \( V = 0 \) to 108 km/h). Figure 6.2, shows the theoretical power dissipation results.
The theoretical results indicated that the power dissipation increased for both an increase in vehicle velocity and increasing level of road roughness. Up to 50 (W) per suspension damper was dissipated for a vehicle travelling at a velocity of 100 (km/h) on a road classified as 'average'. For a vehicle travelling on road surfaces defined as 'very-good', however, a maximum of three Watts were dissipated per damper, and for a road surface classified as 'poor', over 100 (W) were dissipated for the vehicle travelling at \( V = 60 \text{ km/h} \).

As previously stated, it would also provide an important analysis to compare the theoretical results to experimental investigations undertaken by other researchers. This comparison would reveal limitations to either investigation, and would assist in determining the relationship between energy dissipation and factors such as vehicle velocity, and road surface condition. A comparative analysis is given in the following section.
6.2.2. Theoretical and Experimental Energy Dissipation Comparison

During the course of the literature review, only one documented experimental investigation specifically investigating power dissipation in vehicle dampers was uncovered. This investigation was undertaken by Browne and Hamburg [10], and was reviewed in Section 2.4.3. These experimental results were compared with the theoretical energy dissipation investigation previously presented in Section 6.2.1. The investigation undertaken by Browne and Hamburg [10] estimated power dissipation in vehicle suspension by measuring the instantaneous axial force and damper velocity for a vehicle traversing a given road surface. Although Browne and Hamburg analysed two vehicles, only one is referred to in this investigation. This vehicle was a J 2000 Pontiac, and the specifications are given in Table 6.2.

<table>
<thead>
<tr>
<th>Vehicle Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Mass</td>
<td>1125 (kg)</td>
</tr>
<tr>
<td>Passenger and Instrumentation Mass</td>
<td>249 (kg)</td>
</tr>
<tr>
<td>Tires</td>
<td>Firestone 721 P195/70R13 SB</td>
</tr>
<tr>
<td>Suspension</td>
<td>Rally Handling</td>
</tr>
<tr>
<td>Shock Absorbers (GM Part No.)</td>
<td>L. Front 22034719</td>
</tr>
<tr>
<td></td>
<td>R., L. Rear 4993559</td>
</tr>
</tbody>
</table>

**Table 6.2 J 2000 Pontiac Sedan Specifications [10].**

The specifications of the vehicle components of unsprung mass, and suspension spring and damping parameters, were not given by Browne and Hamburg. For this analysis, therefore, the vehicle spring and damper parameters were estimated from the total vehicle mass (given in Table 6.2) and typical vehicle damping ratio and natural frequency parameters of \((\zeta = 0.313)\) and \((\omega_n = 8.37 \text{ rad/s})\) [30], respectively. The tire and suspension parameters were estimated from a typical vehicle system model given by Segel and Lu [30]. Therefore, the J 2000 Pontiac parameters, according to the quarter car, two degree-of-freedom model, are given in Table 6.3.
The two degree-of-freedom, quarter-car model parameters for the J 2000 Pontiac were the same as for the theoretical model used in Section 6.2.1. According to the road surface description by Browne and Hamburg, the experimental power dissipation data was divided into one of five road-surface roughness regions. A detailed analysis of the road surface description and experimental data obtained by Browne and Hamburg [10] is presented in Appendix C. The comparison between the experimental data and the theoretical model is given in Figure 6.3, in which the experimental results are given as individual symbols.

![Figure 6.3 Experimental and Theoretical Power Dissipation Comparison.](image-url)
The energy dissipation results in Figure 6.3 revealed that the experimental measurements of power dissipation in vehicle suspension systems obtained by Browne and Hamburg [10] coincided reasonably accurately with the theoretical estimates. For the experimental power, measured for road surfaces classified as 'poor', the power dissipation ranged from 16.8 to 73.3 (W). These results were, generally, slightly lower than the theoretical estimate. The experimental power measurements for roads classified as 'average', ranged between 9.0 and 17.4 (W) and were also, generally, slightly below the theoretical estimates. The experimental measurements for road surfaces classified as 'good' coincided quite accurately with the theoretical estimates, and ranged between 1.9 and 8.8 (W) per damper. The experimental power measurements for road surfaces classified as 'very good', were slightly larger than the theoretical estimates, and ranged from 1.0 to 8.1 (W). No experimental data was measured for road surfaces classified as 'very poor'.

The overall findings indicated that typical dissipation rates in a vehicle suspension were approximately 5 to 20 (W) per vehicle damper, however, approximately 50 (W) could be dissipated for a vehicle traversing a road classified as 'poor'. The results in Figure 6.3 also indicated the general trends between power dissipation and the factors of vehicle velocity and road surface condition. Overall, it appeared that the power dissipation generally increased for an increase in vehicle velocity, and increased for an increasingly degraded road surface condition.

One limitation of the comparison between the theoretical and experimental data was due to the estimation of the experimental road surface classification. The road surface description, given by Browne and Hamburg [10], could only provide a limited amount of information about the road surface. Therefore, the actual classification of the experimental data in Figure 6.3 should be considered as tentative. Also, the experimental road surfaces had large-scale features, such as isolated pot-holes and expansion joints, which were not included in the theoretical model. These large-scale features would have had a large effect in the excitation of the suspension system [29] and, therefore, would affect the power dissipation results.
The comparison between the experimental and theoretical results was made as consistent as possible. This was achieved by only including experimental data for uniform vehicle conditions and parameters. For instance, data was only included for constant tire inflation pressure, and for the one vehicle model.

The damper dissipation results from Figure 6.3 provided important data for the determination of regenerative damping performance. This was because the performance was largely dependent on the energy dissipation. It was then possible to estimate the amount of energy regenerated due to regenerative damping if the vehicle velocity, road surface condition, and regeneration efficiency were known. This is discussed further in Chapter 7, in which a specific investigation of regenerative suspension performance is documented.

6.3. **Comparison of Linear and Rotating Dampers**

6.3.1. Overview

It was documented in the Literature Review (Section 2.3) and in Chapter 4 (Section 4.5.5), that rotating dampers may provide a significant benefit for applications such as regenerative damping. This benefit stemmed from the mechanical amplification gained from the use of a rack-and-pinion type mechanism. The amplification led to increased damping and energy regeneration for a given damper size, volume and cost. Considering that electromagnetic dampers tend to be heavier than hydraulic devices of similar force levels [25], and that weight minimisation is an important consideration in vehicle design, the use of rotating dampers may have a significant impact on the performance of regenerative damping in vehicle suspension.
The analysis of the dynamic systems in Sections 6.3, 6.4 and 6.5 was undertaken using the 'Bond-Graph' methodology described by Karnopp and Rosenberg [16], and analysed using the mathematical software, MATLAB® [64].

6.3.2. Analysis of Electromagnetic Damper Mass

An alternative to using rotating dampers in vehicle suspension is to use linear dampers. However, because it is difficult to mechanically amplify linear devices, the mass of the linear damper will generally be larger than for a rotating damper. This would significantly reduce the performance of implementing linear electromagnetic dampers in vehicle suspension. Ryba [26] analysed the damping performance of a linear, radial-fin, electromagnetic device, which was suggested originally by Karnopp [25]. This linear device was also described in the Literature Review (Section 2.3). The preliminary calculations by Ryba led to the result that the mass of this linear motor would be quite considerable for the application of a sprung seat [26]. The details of this analysis were not documented, so a brief analysis is given below.

For a conventional vehicle system, a typical damping coefficient may be, \( c_v = 1400 \) (Ns/m) [30]. The analysis of electromagnetic dampers in Chapter 4 (Section 4.3.2), revealed that the maximum damping coefficient \( C_{MAX} \), for a DC electromagnetic device (with no amplification) was given by,

\[
C_{MAX} = \frac{B_0^2 \nu}{\sigma} \quad \text{[Ns/m]},
\]

where:
- \( B_0 \) = magnetic field strength,
- \( \nu \) = volume of conducting material, and
- \( \sigma \) = conductivity of the conducting material.

The analysis of an 'impedance-matching' regenerative interface, in Chapter 5, revealed that there was, approximately, an inverse relationship between the regenerative power efficiency \( E_{REG} \) and normalised damping \( F_{DAMP} \). For example, for a system with a
regeneration efficiency of 50 per cent \( E_{\text{REG}} = 0.5 \), the normalised damping ratio would be \( F_{\text{DAMP}} = 0.5 \). Therefore, in this case, the maximum electromagnetic damping coefficient required would be \( C_{\text{MAX}} = 2800 \text{ Ns/m} \). From Equation 6.6, and for a typical magnetic field strength of \( B_0 = 0.4 \text{ T} \), the conducting volume would be \( V = 300 \text{ cm}^3 \), and conductor mass \( M_{\text{COND}} = 2.67 \text{ kg} \), for the electromagnetic damper (with no amplification).

Because the damping coefficient was proportional to the square of the gear-ratio [18], for a rotating damper with a nominal gear ratio \( \alpha = 3 \), the device volume would reduce to \( V = 33.3 \text{ cm}^3 \), and conductor mass would reduce to \( M_{\text{COND}} = 0.297 \text{ kg} \). Therefore, the device mass and volume could be reduced significantly by using a rack-and-pinion type mechanism. Although this example ignores the added mass and energy losses due to the rack-and-pinion mechanism, it does indicate the potential advantage of using amplification to increase the performance of the regenerative damper system\(^2\).

Section 6.3.3 presents an analysis of the relationship between rotating mass and the damping performance of rotating damping systems. The analysis in Section 6.4 outlines the disadvantage of increasing the rotating mass of a damper for the application of vehicle suspension. Therefore, this analysis outlines the importance of undertaking the investigation of rotating mass, with respect to determining the performance of rotating, regenerative dampers in vehicle suspension.

6.3.3. Rotating Mass and Damping Coefficient Relationship.

For the purposes of determining the performance of regenerative damping, it was important to determine the relationship between the rotating mass and damping coefficient of a rotating damper. As explained in Section 4.5.5, an increase in gear-ratio for a rotating damper led to an increase in damping coefficient for a given mass, volume and cost. The analysis documented in this section investigated the relationship between

\(^2\) The amplification may have a significant influence due to the reduction in the mass of the permanent magnet material of the electromagnetic damper. The significance of this is illustrated in the system performance analysis presented in Chapter 7 (Section 7.3.2).
damping coefficient and rotating mass. The analysis was based on the assumption that, for a particular rotating electromagnetic damper used in a vehicle damping application, it was beneficial to maximise the gear-ratio to maximise the effective device damping coefficient. However, increasing the gear-ratio increased the rotating mass, which could degrade a vehicle's dynamic response.

An important observation, with respect to regenerative damping, was that an increase in the gear-ratio, for a given electromagnetic damper, was beneficial because it led to an increase in regeneration efficiency. This was because, as the gear-ratio increased, so did the maximum electromagnetic damping coefficient for a given electromagnetic damper. To provide the required vehicle damping coefficient, a lower normalised damping ratio \((F_{DAMP})\) was, therefore, required. This led to an increase in regeneration efficiency. (Refer to Chapter 5, Sections 5.3.4 and 5.3.5 for an analysis of the normalised damping and regeneration efficiency).

In Section 4.3.2 the relationship between the damping coefficient of an electromagnetic damper \((c_v)\), and the volume of conducting material \((\mathcal{V})\), magnetic field strength \((B_0)\), and the conducting material conductivity \((\sigma)\), was presented. It was also shown by Suda and Shiiba [18] that the damping coefficient was proportional to the square of the gear-ratio \((\alpha)\). Also, in Section 5.3.4, the relationship between the "effective" damping coefficient, the maximum damping coefficient produced by an electromagnetic device, and the normalised damping ratio \((F_{DAMP})\) was given. From these relationships, the effective damping coefficient of an electromagnetic damper could be given by,

\[
CV = \frac{1}{F_{DAMP}} \frac{B_0^2 \mu}{\sigma \alpha^2} \quad [\text{Ns/m}],
\]

(6.7)

where the gear-ratio, based on the assumption that the force produced by the damper was produced at the average conductor radius, was given by,
\[ \alpha = \frac{r_{\text{COND-AV}}}{r_{\text{GEAR}}}, \quad (6.8) \]

where: 
- \( r_{\text{COND-AV}} = \) average conductor radius of the damper, and 
- \( r_{\text{GEAR}} = \) gear radius of the rack-and-pinion mechanism.

With respect to the mass of the conducting material in the electromagnetic damper \((M_{\text{COND}})\) and the normalised damping ratio, the effective damping coefficient could also be given by,

\[ \alpha_v = \frac{1}{F_{\text{DAMP}}} \frac{B_o^2 M_{\text{COND}}}{\rho_{\text{COND}} \sigma} \alpha^2 \quad [\text{Ns/m}], \quad (6.9) \]

where \((\rho_{\text{COND}})\) was the density of the conducting material. The rotating mass \((m_r)\), was given by,

\[ m_r = \frac{I}{r_{\text{GEAR}}^2} \quad [\text{kg}], \quad (6.10) \]

where: \( I = \) rotating damper inertia.

Assuming that the average conductor radius of the electromagnetic damper \((r_{\text{COND-AV}})\), was half the total conductor radius \((r_{\text{COND}})\), and that the conductors were in the form of a solid cylinder\(^3\), the rotating inertia was given by,

\[ I = 2 M_{\text{COND}} r_{\text{COND-AV}}^2 \quad [\text{kg m}^2]. \quad (6.11) \]

Therefore, the rotating mass was also given by,

\[ m_r = 2 M_{\text{COND}} \alpha^2 \quad [\text{kg}]. \quad (6.12) \]

\(^3\) Although the conductors would not be an exact solid cylinder in a real device, they would, generally, be in a cylindrical arrangement. The significance of the relationship in Equation 6.11, was that the rotating inertia \((I)\), was proportional to the total conductor mass \((M_{\text{COND}})\), and the average conductor radius, squared \((r_{\text{COND-AV}}^2)\). This, generally, would also be the relationship for any cylindrical mass system.
It had already been noted that, for a given electromagnetic rotating damper, the damping coefficient increased for an increase in gear-ratio (Equation 6.9). It was also shown that the rotating mass increased in proportion to the gear-ratio squared (Equation 6.12). From these two relationships it could, therefore, be shown that the damping coefficient was linearly proportional to the rotating mass. This is given by,

\[ c_v = \frac{1}{F_{DAMP}} \frac{B_0^2}{2 \sigma \rho_{COND}} mr \quad [\text{Ns/m}]. \]  

(6.13)

This finding had important implications for determining the performance of electromagnetic regenerative dampers. This was because it was desirable to maximise the damping coefficient for a given electromagnetic device, and normalised damping ratio, however, as the damping coefficient increased, so did the rotating mass. The disadvantages of using a large rotating mass in vehicle suspension are shown in the following section (Section 6.4).

Although an electromagnetic damper was analysed for this investigation, the analysis would apply equally for any form of rotating damper. This was because the resulting damping coefficient would generally be proportional to the mass of the damper, as was the case for this analysis.

### 6.4. Analysis of Rotating Damper Dynamics

#### 6.4.1. Overview

It was shown in Section 2.3 that a disadvantage of rotating dampers stemmed from their effect on the suspension dynamic response. Ryba [26], analysed the use of rotating devices, with the objective of determining an optimal force generator to be used as a semi-active vehicle damper. This analysis revealed that the transmissibility of the
rotating damper system, at high frequencies, did not approach zero as it did for the equivalent linear damper system. Figure 6.4 shows a one degree-of-freedom model with either a linear damper (a) or rotating damper (b), as analysed by Ryba [26].

![Figure 6.4 One Degree-of-Freedom Suspension Model.](image)

A comparison between the dynamic characteristics of the one degree-of-freedom system with either the linear or rotating damper is shown in Figure 6.5 [26]. In this figure, 'H' was the isolation response \( \frac{X_v}{X_o} \), and 'F' was a measure of frequency.

![Figure 6.5 High Frequency Dynamic Response with Rotating Damper [26].](image)

The isolation response for the rotating damper system (solid line), revealed that there was a finite transmission for high frequencies \( (H_\infty) \). However, the response of the linear damper system (dashed line) provided zero transmission at high frequencies. From this result, therefore, Ryba [26] concluded that the use of rotating electric motors may cause problems for the application of damping in vehicle systems.

The finite transmission response for the rotating damper can also be illustrated using the transfer function of the dynamic system. Equations 6.14 and 6.15 give the transfer
function of the sprung mass isolation for a one degree-of-freedom system with either a linear or rotating damper, respectively. These transfer functions were evaluated using a bond-graph analysis [16] of the dynamic system shown in Figure 6.4. For the rotating damper system, with a rotating mass \((mr)\), the order of the numerator increased relative to the linear system, and a 'zero' was added to the transfer function. Therefore, for large frequencies (as \(s = j\omega\) approaches infinity), \(H_{\text{LIN}}(s)\) approached zero, whereas, \(H_{\text{ROT}}(s)\) approached \(\left(\frac{mr}{mr + mv}\right)\). Therefore, adding (or increasing) the rotating mass has a degrading influence on the vehicle dynamics due to transmission of the high-frequency components of the input disturbance.

\[
H_{\text{LIN}}(s) = \frac{cv s + kv}{mvs^2 + cv + kv}. \tag{6.14}
\]

\[
H_{\text{ROT}}(s) = \frac{mrs^2 + cv s + kv}{(mr + mv)s^2 + cv s + kv}. \tag{6.15}
\]

The analysis documented in the following sections further investigated rotating dampers in vehicle suspension systems. The investigation in Section 6.4.2 was similar to the investigation by Ryba [26], however, a two degree-of-freedom vehicle model was analysed. The analysis in Section 6.4.3 investigated the system transfer function with the objective of negating the degrading effects of rotating dampers.

6.4.2. Two Degree-of-Freedom Vehicle Model

Although Ryba's [26] results suggested that rotating electromagnetic dampers were unsuitable for vehicle suspension systems, a more detailed vehicle model was investigated in this analysis. The one degree-of-freedom suspension system analysed by Ryba neglected the vehicle tire dynamics. A more accurate representation of a vehicle system was the two degree-of-freedom, quarter car model. This model was previously
described in Section 6.2.1, and is shown in Figure 6.6, for either a linear (a) or rotating vehicle damper (b).

![Diagram of vehicle damper systems](image)

**Figure 6.6 Two Degree-of-Freedom Suspension Model.**

The effect of adding the extra degree-of-freedom to the vehicle model altered the high-frequency response of the system. This was due to the high spring-rate of the tire system acting as a 'low-pass filter', and reducing the amplitude of the high frequency road disturbances transmitted to the unsprung mass. In the following analysis, a one degree-of-freedom, rotating damper system, as analysed by Ryba [26], was compared against the two degree-of-freedom models shown in Figure 6.6. The dynamic system specifications were typical vehicle parameters used by other researchers [30], and are given in Table 6.4.

<table>
<thead>
<tr>
<th>1-DOF Rotating System</th>
<th>2-DOF Linear System</th>
<th>2-DOF Rotating System</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_v = 267$ (kg)</td>
<td>$m_v = 267$ (kg)</td>
<td>$m_v = 267$ (kg)</td>
</tr>
<tr>
<td>$k_v = 1.87 \times 10^4$ (N/m)</td>
<td>$k_v = 1.87 \times 10^4$ (N/m)</td>
<td>$k_v = 1.87 \times 10^4$ (N/m)</td>
</tr>
<tr>
<td>$c_v = 1400$ (Ns/m)</td>
<td>$c_v = 1400$ (Ns/m)</td>
<td>$c_v = 1400$ (Ns/m)</td>
</tr>
<tr>
<td>$m_t = 36.6$ (kg)</td>
<td>$m_t = 36.6$ (kg)</td>
<td></td>
</tr>
<tr>
<td>$k_t = 1.84 \times 10^5$ (N/m)</td>
<td>$k_t = 1.84 \times 10^5$ (N/m)</td>
<td>$k_t = 1.84 \times 10^5$ (N/m)</td>
</tr>
<tr>
<td>$c_t = 294$ (Ns/m)</td>
<td>$c_t = 294$ (Ns/m)</td>
<td></td>
</tr>
<tr>
<td>$m_r = 10$ (kg)</td>
<td></td>
<td>$m_r = 10$ (kg)</td>
</tr>
</tbody>
</table>

**Table 6.4 Theoretical Vehicle Specifications for the Dynamic Analysis.**
The rotating mass of \((mr = 10 \text{ kg})\) is estimated to be a typical rotating mass for vehicle electromagnetic damping applications [65]. Figure 6.7 provides a comparison between the frequency response of the three systems.

![Figure 6.7 Comparison of Vehicle Isolation Response with Rotating Damper.](image)

The isolation response of the three systems in Figure 6.7 reveals that, although the response was degraded between approximately 5 and 12 (Hz), a two degree-of-freedom model attenuated the high frequency component of the input disturbance. This attenuation could also be shown by the system transfer function for the linear and rotating dampers, which are given in Equations 6.16, and 6.17, respectively. These transfer functions were evaluated using a bond-graph analysis.

\[
H_{\text{LIN}}(s) = \frac{(ct \cdot cv) s^2 + (cv \cdot kt + ct \cdot kv) s + (kt \cdot kv)}{(mv \cdot mt) s^4 + (cv \cdot mt + ct \cdot mv + cv \cdot mv) s^3 + (kv \cdot mt + kt \cdot mv + kv \cdot mv + ct \cdot cv) s^2 + (cv \cdot kt + ct \cdot kv) s + (kt \cdot kv)}
\]

(6.16)

For the two degree-of-freedom, linear damper system (Equation 6.16), the order of the numerator was two, and the order of the denominator was four. Therefore, for high frequencies (as \(s = jo\) approached infinity), the output response of the linear suspension model approached zero.
For the two degree-of-freedom, rotating damper system (Equation 6.17), the order of the numerator was three, and the order of the denominator was four. Therefore, for high frequencies, \((s = j\omega)\) approached infinity, the output response also approached zero. This differed from the one degree-of-freedom model, analysed by Ryba [26], in which the response for the rotating damper system did not approach zero for high frequencies. This result had important implications for the use of rotating dampers in vehicle suspension applications, and indicated that the use of rotating dampers may not have been as restricted as stated by Ryba [26]. In the context of this research dissertation this result is important, because the use of rotating dampers significantly affected the performance of regenerative damping in vehicle systems.

It was possible to improve the vehicle isolation response further by analysing the pole and zero locations of the transfer function. By adding additional elements to the system model, it was possible to negate the effects of the rotating damper, altogether. This investigation is documented in the following section.

6.4.3. Pole - Zero Analysis of the Rotating Damper System

Due to the benefits of mechanical amplification with rotating dampers, the results in Section 6.4.2 had important implications for the use of rotating dampers in regenerative damping applications. The isolation response of the rotating damper system was degraded compared to a linear damper system, however. It was possible to counteract the negative dynamic effects of the rotating mass altogether, by modifying the dynamic system. This was achieved by modifying the pole-zero locations of the rotating damper system to match those of an ideal, linear damper system, with the remaining pole-zero locations negated using pole-zero cancellation. A modified suspension model was
proposed to provide pole-zero cancellation, and is shown in Figure 6.8. This model had a spring \((kr)\), and damper \((cr)\), combination in series with the rotating damper.

\[
H_{\text{ROT.M}}(s) = \frac{(mr \, cr) \, s^3 + \{(mr \, (kv + kr) + cv \, cr)\} \, s^2 + \{(cv \, (kv + kr) + cr \, kv)\} \, s + \{(kv \, kr)\}}{(mv \, mr) \, s^3 + \{(mv \, (cv + cr) + cr \, mr)\} \, s^2 + \{(mr \, (kv + kr) + mv \, kr + cv \, cr)\} \, s + \{(cv \, (kv + kr) + cr \, kv)\} \, s + \{(kv \, kr)\}}.
\]

\[(6.18)\]

It was possible to modify the added damping and spring parameters \((cr)\) and \((kr)\), such that pole-zero cancellation occurred. For the system to match a linear damper system, two of the pole-zero groupings cancelled, and the other poles and zeros matched the locations of the linear damper, one degree-of-freedom system. As an example, the system in Figure 6.8 was analysed for the typical vehicle parameters given in Table 6.5.
Due to the complexity of analytically evaluating the pole and zero locations of the modified system, the added damping and spring parameters ($kr$ and $cr$) were numerically evaluated for this example. The parameters ($cr = 1400$ Ns/m) and ($kr = 1.96 \times 10^5$ N/m) provided pole-zero cancellation, which led to the pole-zero locations, as shown in Table 6.6. The pole and zero locations of the linear system were matched by the modified rotating system poles and zeros, and pole-zero cancellation occurred for the remaining modified system poles and zeros.

The isolation response of the modified and linear systems in Figure 6.9 revealed that the dynamic response of the modified rotating system was the same as for the linear system.
A disadvantage with the modified rotating damping system was that, when used with regenerative dampers, the added spring and damper would reduce the amplitude of the input disturbance transmitted to the rotating damper and, therefore, would reduce the regenerative energy efficiency. However, in order to avoid high frequency transmission to the vehicle body this characteristic was apparently unavoidable. The relative displacement \( \frac{x_{rel}}{x_0} \) across both the modified and unmodified rotating damper system was, once again, evaluated using a bond-graph approach. This response is shown in Figure 6.10.
An important aspect of the modified design was that pole-zero cancellation was possible for a range of typical vehicle parameters. By numerically evaluating the damper and spring parameters for a range of rotating damper masses, it was found that for pole-zero cancellation to occur,

\[ cr = cv \quad [\text{Ns/m}], \quad \text{and} \quad kr = \frac{cv^2}{mr} \quad [\text{N/m}]. \tag{6.19} \]

Equation 6.19 revealed that, for pole-zero cancellation to occur, the damping coefficient of the added damper \((cr)\), needed to be as large as the suspension damper coefficient. Also, for this example, the added spring coefficient \((kr)\), was an order of magnitude larger than suspension spring constant. Due to the relatively high added spring and damper parameters, the feasibility of implementing the added damper and spring combination, as a solution to the rotating damper response, would have been limited. However, it would have also been possible to use damper and spring elements in series with the rotating damper, but with lower damping and spring parameters than specified in Equation 6.19. The effect of this would have been that the dynamic isolation response would have been improved relative to the unmodified system, but without pole-zero cancellation.
An analytical investigation of the present proposal, as well as an analysis of a more comprehensive vehicle model, would provide further insights, and reveal further limitations of the present system. Therefore, this analysis is suggested for continued research.

6.5. Limits of the Damper Rotating Mass

The analysis in Section 6.3.3 indicated that, for a particular rotating electromagnetic damper used in a vehicle damping application, it was beneficial to maximise the gear-ratio to maximise the effective damping coefficient (Equation 6.7). However, by increasing the gear-ratio, the rotating mass also increased (Equation 6.12) and, as shown in Section 6.4, this led to the degradation of the vehicle dynamic response. By determining the allowable rotating mass, it would be possible to maximise the available damping coefficient with a particular electromagnetic damper, which would then lead to the maximisation of regeneration efficiency and would assist in determining the overall performance of regenerative vibration dampers.

In Section 6.4.3 it was revealed that it would be possible negate the effects of the rotating mass by modifying the dynamic system, and adding spring and damper components in series with the rotating damper. However, the findings also indicated that, to negate the effect of the rotating mass, the added damper and spring would have had relatively high force constants. As such, this proposal may not have been a feasible solution to the use of a rotating mass in vehicle suspension. The analysis documented in this section, therefore, proposed a method to determine the limits of the rotating mass in vehicle suspension, for a rotating damper directly connected between the sprung and unsprung mass.

The method to determine the rotating mass limits for vehicle suspension in the following section, used human response to vibration. By measuring the passenger
vibration response as a function of rotating damper mass, and establishing passenger vibration limits, it was possible to establish the limit of the mass of rotating dampers in vehicle suspension.

Although using a human response to vibration was one method to determine the limits of rotating mass, there may be other methods. For example, the limit of rotating mass may have been determined by undertaking a stress analysis of the suspension components; especially the rack-and-pinion mechanism used to convert linear to rotary motion. Generally, for an increase in rotating mass, the forces on the rack-and-pinion mechanism connecting the damper between the sprung and unsprung mass would also increase. The rotating mass would, therefore, be limited by the breaking strain and fatigue properties of the rack-and-pinion material. However, the limitation of this method was due to the uncertainty of the particular mechanical construction at an early stage of system development, such as for when this research was undertaken. Although a mechanical stress analysis was considered beyond the scope of the present research program, it would be necessary when implementing rotating dampers in an actual vehicle suspension system and, as such, is suggested for further research.

6.5.1. Rotating Mass Effects on Passenger Isolation

The use of human sensitivity to vibration had previously been used to evaluate the value of suspension system design [33, 66-71], and was used in this investigation as a method to determine the maximum allowable rotating mass in vehicle suspension. Human response to vibration was measured using subjective ride assessment; shake table tests, and ride measurements in vehicles [66]. An International Standard (ISO 2631) [72] had given guidance to the acceptable human exposure to whole body vibration, and was used in the research documented herein to analyse suspension system design with the use of rotating dampers.

Figure 6.11 shows the (ISO 2631) fatigue-decreased proficiency boundary for a human subjected to vertical vibration. This response was used to determine the limits of human
vibration encountered in this analysis. The response was described by the root-mean-square (RMS) vertical acceleration as a function of frequency for various exposure times. In this analysis, the rotating damper mass was limited by maintaining the passenger acceleration below a particular boundary. Refer to Appendix D, Table D.1 for a tabulated version of the (ISO 2631) reduced-comfort boundary.

Figure 6.11 Longitudinal Acceleration Limits (Fatigue-Decreased Proficiency Boundary) [72].

The analysis was conducted using theoretical vehicle and road surface models. The vehicle model was based on a quarter car, two degree-of-freedom model, as previously described in Section 6.2.1, and is shown in Figure 6.12. This model was used because it had the most general and useful suspension system design information, being described by few parameters and it only had one input [41].
The passenger discomfort was evaluated at various frequencies, by multiplying the passenger acceleration by the frequency-dependent weighting factor (or the 'Human Sensitivity for Vibrations') [33]. The ISO weighting factor \((Q^2)\), is shown in Figure 6.13 as a function of frequency ('Simic' was another weighting factor not included in this analysis).

As previously described in Section 6.2.1 the road surface model could be rationally described in statistical terms [33, 63]. According to the proposed ISO Standard (1972)
the road profile, as a function of temporal frequency, could be represented by the elevation spectrum, as previously shown in Equation 6.1, and repeated in Equation 6.20.

\[
S_{xo}(\omega) = \frac{2\pi V}{\omega^2} A \quad [\text{m}^2/\text{rad}], \quad (6.20)
\]

where: 
\[ V = \text{vehicle velocity}, \quad \text{and} \]
\[ A = \text{road surface roughness coefficient (as defined in Table 6.1).} \]

A measure of discomfort was obtained by measuring the spectral density of the passenger acceleration, weighted for human sensitivity as given by,

\[
S_{ao}(\omega) = Q^2 |H_{xv\timeso}(\omega)|^2 S_{xo}(\omega) \quad [\text{m}^2/\text{s}], \quad (6.21)
\]

where \( H_{xv\timeso}(\omega) \) was the transfer function relating the sprung mass displacement \( (x_v) \), to the vertical roadway displacement \( (x_o) \), given previously in Equation 6.17, and \( (Q^2) \), was the weighting factor for human sensitivity. The RMS value of acceleration \( (a_z) \), for a particular frequency band, with centre frequency \( (\omega_c) \), was evaluated by integrating the spectral density function over the frequency band of 0.89 to 1.12 \( \omega_c \) [33], as described in Equation 6.22.

\[
a_z = \left[ \frac{1}{2\pi} \int_{0.89\omega_c}^{1.12\omega_c} S_{ao}(\omega) d\omega \right]^{1/2} [\text{m/s}^2]. \quad (6.22)
\]

By comparing the RMS acceleration, measured using Equations 6.21 and 6.22, against the ISO fatigue-decreased proficiency boundary, it was possible to evaluate whether the vibration response of a particular vehicle system was acceptable. A typical theoretical vehicle model was analysed, and the dynamic system specifications for a typical vehicle model, used by other researchers [30], are given in Table 6.7.
For this analysis, the acceleration response was evaluated for a road surface classified between 'average' and 'poor'. This surface was considered below average but, nevertheless, representative of a typical road surface encountered in a common driving schedule. Therefore, from Table 6.1, the road roughness coefficient was \( A = 0.035 \times 10^{-4} \) m.cycle. The acceleration response, as a function of rotating mass, was compared against the reduced comfort boundary (ISO 2361) for one, eight and twenty-four hours duration. Figure 6.14 shows the effect of rotating damper mass on vehicle acceleration for a vehicle travelling at a velocity of \( V = 28 \text{ m/s} \) for a rotating mass of \( m_r = 0, 10, 20, \) and 30 kg). Refer to Appendix D, Table D.2 for the acceleration response measurements.

The results in Figure 6.14 revealed that an increase in rotating mass increased the longitudinal acceleration of the vehicle passengers. Also, the rotating mass mainly influenced the vehicle system around 8 (Hz), which was the natural frequency of the tire system. For a rotating mass of \( m_r = 30 \) kg), the acceleration response exceeded the ISO 8 Hour, reduced comfort boundary, and for no rotating mass \( m_r = 0 \), the acceleration response exceeded the 24 Hour, reduced comfort boundary.

---

Table 6.7 Vehicle Specifications for Acceleration Response Analysis.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle (Sprung) Mass</td>
<td>( m_v )</td>
<td>267</td>
<td>kg</td>
</tr>
<tr>
<td>Vehicle Spring Constant</td>
<td>( k_v )</td>
<td>( 1.87 \times 10^4 )</td>
<td>N/m</td>
</tr>
<tr>
<td>Vehicle Damping Coefficient(^4)</td>
<td>( c_v )</td>
<td>1400</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Tire (Unsprung) Mass</td>
<td>( m_t )</td>
<td>36.6</td>
<td>kg</td>
</tr>
<tr>
<td>Tire Spring Constant</td>
<td>( k_t )</td>
<td>( 1.84 \times 10^5 )</td>
<td>N/m</td>
</tr>
<tr>
<td>Tire Damping Coefficient</td>
<td>( c_t )</td>
<td>294</td>
<td>Ns/m</td>
</tr>
</tbody>
</table>

\(^4\) In this analysis, it was assumed that the vehicle damping coefficient was held constant. For the application of regenerative vibration damping, this was in accordance with the regenerative interface control function developed to control the damping ratio (Section 5.3.4).
An analysis was also undertaken to determine the influence of road surface roughness on the acceleration of the vehicle passengers. The results of this analysis were undertaken for a rotating mass \((m_r = 10.0 \text{ kg})\), and for a vehicle travelling at a velocity \((V = 28 \text{ m/s})\). The results of this analysis are shown in Figure 6.15. Refer to Appendix D, Table D.3 for the acceleration response measurements. For a given rotating mass, an increasingly degraded road surface condition led to an increase in longitudinal acceleration for all frequencies. For a rotating mass \((m_r = 10.0 \text{ kg})\), the acceleration response remained under the 8 Hour, reduced comfort boundary, for roads classified as 'average' or better. For road surfaces classified as 'poor' or 'very poor', however, the response exceeded the 8 Hour, reduced comfort boundary.

The results in Figure 6.14 and Figure 6.15 indicated that it was possible to obtain the rotating mass limits, provided that suitable constraints were determined for the particular vehicle system. For instance, from Figure 6.14, if an 8 Hour, reduced comfort boundary was specified for a vehicle traversing a road classified 'average' or better, then a rotating mass \((m_r = 20.0 \text{ kg})\), was acceptable, but a rotating mass \((m_r = 30.0 \text{ kg})\), was not acceptable. Also, from Figure 6.15, a rotating mass \((m_r = 10.0 \text{ kg})\), was acceptable.
only for road surfaces classified as average or better, for an 8 Hour, reduced comfort boundary.

![Figure 6.15 RMS Vehicle Acceleration and Reduced Comfort Boundary - Variable Road-Surface Condition.](image)

The typical vehicle requirements for suspension design may be defined for a limit of an 8 Hour exposure (as a typical long drive-cycle period), on road surfaces classified as 'average' or better. In this case the rotating mass would be limited below approximately \(mr = 20.0\) kg).

Further analysis is needed to experimentally verify the theoretical results presented here, and determine whether the rotating mass was limited by other factors such as mechanical stress. Also, these results may be improved by including a more detailed vehicle model, which includes the effects of all four wheels, or the inclusion of seat dynamics, and is, therefore, suggested for further research.
6.6. Conclusions

The analysis documented in this Chapter raised two issues that had important implications for determining the performance of regenerative damping in vehicle systems. Firstly, an investigation of the amount of energy dissipated in vehicle suspension systems was undertaken and, secondly, an investigation was undertaken into the use of rotating dampers in vehicle suspension systems.

In Section 6.2.1, a theoretical analysis of vehicle suspension dissipation was undertaken by modelling the vehicle dynamic system and the road surface. The theoretical results indicated that up to 50 (W) per suspension damper were dissipated for a vehicle travelling at a velocity of 100 (km/h) on a road classified as 'average'. For a vehicle travelling on road surfaces defined as 'very-good', however, a maximum of three Watts were dissipated per damper, and for a road surface classified as 'poor', over 100 (W) were dissipated travelling at 60 (km/h). In Section 6.2.2, it was shown that the experimental estimates of power dissipation in vehicle suspension systems obtained by Browne and Hamburg [10] coincided reasonably accurately with the theoretical estimates.

The analysis in Sections 6.3 and 6.4 investigated the use of rotating dampers in vehicle suspension. This benefit of using rotating dampers stemmed from the mechanical amplification gained from the use of a rack-and-pinion type mechanism. This led to increased damping and energy regeneration for a given damper size, volume and cost. The disadvantage of rotating dampers stemmed from their effect on the suspension dynamic response. In Section 6.4.2, it was shown that, although the isolation response was degraded between approximately 5 and 12 (Hz), a two degree-of-freedom model attenuated the high frequency component of the input disturbance. This result differed from the previous analysis by Ryba [26] in which, for a one degree-of-freedom model, the rotating damper system had a finite high-frequency isolation response.
A method to negate the effects of the rotating mass was presented in Section 6.4.3. The rotating damper system was modified by adding a spring and damper combination in series with the rotating damper. It was shown that it was possible to negate the effects of the rotating mass with appropriate damping and spring parameters. It was also found, however, that for pole-zero cancellation to occur, the damping and spring parameters were relatively high and, therefore, the feasibility of implementing the added damper and spring combination, as a solution to the rotating damper response, may be limited.

In Section 6.5, the limits of rotating mass in vehicle suspension were estimated for a rotating damper directly connected between the sprung and unsprung mass. This was achieved by theoretically analysing the human sensitivity to vibration within the vehicle system. A typical example was given for an 8 Hour, reduced comfort boundary, for a vehicle traversing a road classified 'average' or better. In this case, a rotating mass \(mr = 20.0 \text{ kg}\) was acceptable, but a rotating mass \(mr = 30.0 \text{ kg}\) was not acceptable.

The results of the rotating mass investigation, presented in this Chapter, assisted in determining the maximum allowable damping coefficient for a given electromagnetic damper. They also assisted in determining the energy regeneration efficiency as well as the mass, volume and cost of the regenerative damper. These factors were important for determining the performance of regenerative suspension. A detailed analysis of these issues, and more issues related to regenerative suspension performance, is presented in Chapter 7, Performance Analysis of Regenerative Damping.
CHAPTER 7

PERFORMANCE ANALYSIS OF REGENERATIVE DAMPING

"To define it rudely but not inaptly, engineering is the art of doing that well with one dollar which any bungler can do with two after a fashion."  

Arthur M. Wellington

7.1. Overview

It was the intention of this research program to provide a comprehensive study of energy recovery, as it pertained to vehicle damping, including both engineering and economic performance factors. The research documented in this dissertation has, thus far, concentrated on determining the most promising regenerative damping system, attempting to maximise the damping and energy regeneration performance and determining the operational limitations for such a system.

It was the purpose of the investigation documented in this chapter to apply the previous research findings to investigate the overall operational performance of regenerative damping for vehicle systems. By analysing particular regenerative system parameters as a function of system performance, it was possible to gain an understanding of the overall design and control of the electromagnetic regenerative system. For instance, with this analysis, it was possible to evaluate the effect of the damper rotating mass, rotating inertia, damping coefficient ratio and vehicle drive-cycle on the overall performance of the regenerative system. The effect of these parameters was only known for an investigation with a broad scope, that included both the regenerative damper, regenerative interface, vehicle system and vehicle drive-cycle.

The method used to analyse the operational performance of the system was based on a cost-analysis. The cost analysis weighed the overall energy gains and losses in the system, and to obtain an objective measure of the value, or feasibility, of such a system. The following section introduces a methodology, to provide an overall objective analysis of regenerative damping as it pertains to vehicle suspension.

7.2. Cost-Analysis - Methodology Description.

7.2.1. Overview

The analysis in this section documents a methodology developed to determine the value of regenerative damping in vehicle systems. The methodology was based on a cost-analysis of the overall regenerative damper and vehicle system. It was based on the hypothesis that regenerative damping had the potential to recover energy and provide a benefit to the vehicle system. However, it would also be detrimental due to factors such as additional weight and material cost. Overall, the factors that contributed to a
generalised cost-analysis investigation, included both economic and practical issues, such as [35]:

- Useful operational life of the system,
- Developmental costs,
- Annual maintenance costs,
- Salvage value or disposal cost,
- Procurement cost of the item,
- Transportation (delivery) and installation costs,
- Annual operating cost of the item. Which included:
  - Energy cost,
  - Cost of supplies,
  - Labor cost, and
  - Cost of materials.

Specifically, with respect to regenerative damping in vehicle systems, these issues may also have included:

- The technical complexity of the system,
- The reliability and safety issues associated with the operation of the system, and
- The perceived market/environmental need for (and marketing potential of) the system.

Each of these factors could be complex in its own right and, moreover, some of the factors were interdependent and crossed traditional disciplinary boundaries (including Engineering and Economics, for example). The objective of this investigation was to provide a methodology detailed enough to give an accurate estimate of the system value, yet have ease of computation and be able to indicate the relationship between system parameters and the overall cost. In order to simplify the analysis, it was not possible to include all the issues listed in the cost-analysis methodology.
The cost-analysis model was partly based on previous investigations in the field of Life Cycle Analysis (LCA) [34-38]. In this analysis, a cost-analysis of the particular system was developed, in which the losses were weighed against the benefits. The system was deemed feasible (of positive value) if the overall benefits of the system outweighed the costs.

To obtain a cost function it was necessary to define the units of 'cost'. The units could be chosen from a number of examples. One example was monetary units, such as Dollars ($). However, there were several disadvantages of using monetary units. Monetary units were both temporally, and spatially variable. Depending on the time and place, the monetary unit may have varied due to local currency variations, and relative to other locations. To determine the value of a recovery system, such as regenerative damping, a more useful cost unit was energy, as the recovery of energy was a major factor of regenerative vibration damping.

According to a cost-analysis methodology, the cost equation of the particular system was given by,

\[ Cost_T = B_T - C_T \] \hspace{1cm} (7.1)

where:
- \( Cost_T \) = overall system energy value,
- \( B_T \) = total energy gained from the system (the benefits), and
- \( C_T \) = total energy lost (the losses).

In this analysis, the units were given as energy per year of operation. This was to provide a measure that was independent of the average operating life-time of the particular regenerative system. According to the cost-analysis methodology, the system was deemed feasible if the total cost (\( Cost_T \)) was positive, and infeasible if (\( Cost_T \)) was negative.

The following sections (Sections 7.2.2 and 7.2.3) document the energy benefit and loss relationships with respect to a regenerative vehicle damping performance analysis.
7.2.2. Energy Benefit Estimation

In this analysis, the total benefits ($B_T$), were assumed to be attributed to the energy added to the vehicle system due to the energy regeneration process. From Equation 7.1, the energy benefit ($B_T$), was the energy added to the overall system per year. To estimate the regenerated energy, it was necessary to determine:

(i) The amount of regenerated energy for a particular vehicle drive condition (i.e., the amount of energy regenerated for a particular road surface, and for a particular vehicle velocity).

(ii) The average vehicle drive cycle for a year of operation (i.e., the total vehicle kilometres travelled, and the proportion of kilometres travelled on a particular road surface and at a particular velocity).

In this analysis, the energy regenerated for a particular vehicle drive condition was estimated from the energy normally dissipated in the vehicle damper and the energy regeneration efficiency of the regenerative damper. An analysis of the energy dissipation in vehicle suspension dampers was presented in Chapter 6, Section 6.2. In this investigation, a comparison between theoretical and experimental power dissipation in vehicle suspension was undertaken. From the power dissipation estimate it was possible to estimate the energy dissipated (over a given time-period) for a particular drive condition. To estimate the energy dissipated per year, the damper dissipation was incorporated into a typical vehicle drive cycle over a year of operation.

To determine the typical vehicle drive cycle over a year of operation, a number of external sources provided statistical drive cycle information. For instance, the Australian Bureau of Statistics provided information for estimates of the average vehicle kilometres travelled per year, and the average age of the total vehicle fleet (for Australian vehicles) [73]. This information was collected from a number of sources within and outside Australia. This included the Motor Vehicle Census, which was a count of all vehicles that were legally registered in Australia at a specific date, and the
Survey of Motor Vehicle Use, which included data on the total and average annual distance travelled for different vehicle types [73].

To determine the energy dissipation, it was necessary to estimate the proportion of the typical vehicle drive-cycle that was undertaken on a particular road surface, at a particular velocity. To reasonably simplify the analysis, it was assumed that 55 per cent of the total kilometres travelled per year were driven on urban road surfaces, and the remainder (45 per cent) on non-urban road surfaces. The same assumption was used by the U.S. Environmental Protection Authority (EPA) for calculating the composite petrol efficiency for each vehicle model [34]. From this, it was possible to estimate the energy dissipation (for a particular damper) per year of operation for either urban ($B_{URB}$), or non-urban ($B_{RUR}$), road use.

The energy recovered due to the regenerative damping process was then evaluated by multiplying the energy dissipation estimates ($B_{URB}$ and $B_{RUR}$), with the expected energy regeneration efficiency ($E_{REG}$). The energy benefit is given in Equation 7.2.

$$B_T = E_{REG} (B_{URB} + B_{RUR}) \text{ [kJ/year].} \quad (7.2)$$

The regeneration efficiency was analysed in Chapter 5, for an impedance-matching regeneration circuit. This investigation revealed that the regeneration efficiency was dependent on the normalised damping ratio ($F_{DAMP}$), (proportion of the actual damping to the maximum electromagnetic damping) of the particular regeneration system. An impedance-matching regeneration scheme would have a specified normalised damping coefficient ($F_{DAMP}$) and, therefore, the regeneration efficiency ($E_{REG}$) would also be specified.

Previous research investigations had revealed that particular regenerative vibration schemes had the ability to not only provide energy regeneration, but also improve the vehicle dynamic response [19, 22]. This improvement may have been perceived as a benefit to the vehicle system. However, it was not possible to quantify the benefit of improved vehicle response with respect to energy usage. Also, the objective of this
research investigation was to specifically investigate the energy regeneration aspects of regenerative vehicle damping, alone. Therefore, dynamic response was not used as a determinant of system performance for this analysis.

7.2.3. Energy Cost Estimation

According to the cost-analysis model, it was necessary to estimate the losses attributed to the use of regenerative damping in vehicle suspension. Based on the vehicle cost-analysis by Schuckert et al. [34], together with other literature in the field of cost-analysis engineering [36-38], it was assumed that the energy losses in the vehicle system due to regenerative damping were due to:

- The energy required (from the vehicle system) to compensate for the effect of the added regenerative damper mass and,
- The energy required to produce the materials required for the regenerative damping system.

Therefore, according to the cost analysis model, the energy cost of the regenerative damping system \( C_T \), was given by,

\[
C_T = C_M + C_p \quad [kJ/\text{year}],
\]

where:

- \( C_M \) = energy loss due to added vehicle mass (per year), and
- \( C_p \) = energy required to produce the damper materials (per year).

This analysis was based on the 'Net Average Rate of Return' criterion described by Sassone and Schaffer [36], in which the sum of the overall system cost was divided by the number of years of operation.

The energy associated with the added vehicle mass \( C_M \), was estimated from the energy lost due to accelerating and decelerating the added mass. Energy was required to accelerate the added mass to the driving velocity (kinetic energy) in every stop-start
vehicle drive-cycle. It was then assumed that all of the kinetic energy was subsequently lost when the vehicle decelerated to stand-still, and dissipated in the vehicle brakes. The amount of energy lost changed according to different drive-cycles. Similar to the previous analysis of energy benefits due to regenerative damping (Section 7.2.2), it was assumed that the vehicle travelled on either urban or non-urban roads. For each of these road surfaces, it was possible to estimate the number of stop-start cycles for each kilometre travelled by the vehicle. If the vehicle velocity was known, it was then possible to estimate the (kinetic) energy required to accelerate the added mass \( M \). This energy is given by Equation 7.4.

\[
C_M = \frac{1}{2} M \left( N_{URB} V_{URB}^2 + N_{RUR} V_{RUR}^2 \right) \text{ [kJ/year]},
\]

(7.4)

where:
- \( V_{URB} \) = average maximum vehicle velocity for urban road use,
- \( V_{RUR} \) = average maximum vehicle velocity for non-urban road use,
- \( N_{URB} \) = average stop-start cycles for urban roads (per year), and
- \( N_{RUR} \) = average stop-start cycles for non-urban roads (per year).

The U.S. Environmental Protection Authority (EPA), Highway Fuel Economy Test specified 1.25 (stops/km) for urban driving and 0.097 (stops/km) for highway driving [74].

From Equation 7.3, the material production cost \( (C_P) \), was the energy cost associated with the production of the materials used in the regenerative damper. The energy cost associated with material production, usage, and disposal was also analysed in the field of Life Cycle Analysis (LCA). In an LCA analysis, material and energy flows for all system processes were investigated and used to identify the energy impact of the particular system under investigation. The scope of LCA began with the extraction of raw materials from the earth, and ended with the disposal of wastes back to it [34]. The production energy for the cost-analysis was the sum of the material production energy and the assembly energy [34]. At this stage of the regenerative damping investigation, it was impractical to estimate the amount of energy associated with the assembly of the particular regenerative system. Therefore, in this cost-analysis methodology, the main interest was to determine the energy associated with the material production.
With respect to the material production energy, the primary production energy was the sum of all energies required to obtain raw material from the earth and to process it into a useable form. Secondary production referred to the process of recycling, and avoided much of the primary production energy. A list of some estimated primary and secondary material production energies is given in Table 7.1 [34].

<table>
<thead>
<tr>
<th>Material</th>
<th>Primary (kJ/kg)</th>
<th>Secondary (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>40,000</td>
<td>18,100</td>
</tr>
<tr>
<td>Iron</td>
<td>34,000</td>
<td>24,000</td>
</tr>
<tr>
<td>Copper</td>
<td>100,000</td>
<td>45,000</td>
</tr>
<tr>
<td>Lead</td>
<td>41,100</td>
<td>8,000</td>
</tr>
<tr>
<td>Aluminium</td>
<td>189,000</td>
<td>26,000</td>
</tr>
<tr>
<td>Glass</td>
<td>30,000</td>
<td>13,000</td>
</tr>
<tr>
<td>Rubber</td>
<td>67,600</td>
<td>43,600</td>
</tr>
<tr>
<td>Nylon</td>
<td>119,000</td>
<td>32,100</td>
</tr>
</tbody>
</table>

Table 7.1 Material Production Energy [34].

It was necessary to determine the mass of the additional material needed for the regenerative system and use this information to estimate the energy required to produce the overall system material. The additional material was estimated for the electromagnetic damper and included the copper, permanent-magnets and the iron pole-pieces. To obtain the cost measure in terms of energy per year of operation, the total material energy cost was divided by the (estimated) average system lifetime.

As the regenerative damper replaced a conventional vehicle damper, the material production costs of the conventional damper were subtracted from the overall suspension material cost.

The performance issues raised in Section 7.2, included factors such as developmental costs, maintenance costs, technical complexity and reliability. However, it was difficult to quantify many of these issues with respect to the energy regeneration process and the cost-analysis model. In this analysis it was assumed that, with suitable development, the reliability of the regenerative damping system would be at least that of a conventional vehicle suspension system. It was also assumed that the technical complexity did not
affect the performance, and that the maintenance costs were negligible. However, further theoretical and experimental research is needed to determine the overall reliability and relative complexity of the system.

7.3. Cost-Analysis Examples

7.3.1. Overview

This section documents an investigation into the operational performance of regenerative damping based on the methodology described in the previous section. To simplify the analysis, several assumptions were made. These assumptions were:

- A passenger vehicle with the typical parameters of total vehicle mass \((m_v = 1,068 \text{ kg})\), and suspension damping coefficient \((c_v = 1,400 \text{ Ns/m})\) was investigated.
- The vehicle traversed either of two road surfaces:
  i) Roads classified as 'urban', which would typically be encountered in city driving at velocities at or below \((V = 60 \text{ km/h})\). These roads had an 'average' road-surface roughness.
  ii) Roads classified as 'non-urban', which would typically be encountered outside of city regions at velocities at or below \((V = 100 \text{ km/h})\). These roads also had 'average' road-surface roughness.
- The damper power dissipation (per damper) for road surfaces classified as 'urban' was \((P_{\text{Diss}} = 15 \text{ W})\), and was \((P_{\text{Diss}} = 40 \text{ W})\) for roads classified as 'non-urban'. This is shown in Figure 7.1, together with the experimental and theoretical power dissipation estimates (from the previous analysis in Section 6.2.2).
The regenerative damper was a DC electromagnetic machine. According to the analysis in Section 4.3.2, and Section 6.3.3, the damping coefficient of the electromagnetic damper was related to the volume of conducting material ($V$), the magnetic field strength ($B_0$), the normalised damping ratio ($F_{DAMP}$), and the gear-ratio ($\alpha$), according to,

$$C_{MAX} = \frac{1}{F_{DAMP}} \frac{B_0^2 V}{\sigma \alpha^2} \text{ [Ns/m]}. \quad (7.5)$$

The electromagnetic damper had a rotational topology, and mechanical amplification was provided by a rack-and-pinion type mechanism. The gear-ratio ($\alpha$), was the ratio between the gear and average-conductor radius (as previously defined in Equation 6.8).

The volume of the iron pole-pieces in the core of the electromagnetic damper was the same as the volume of the conducting material ($V$). This was estimated from the typical structure of a slotted, permanent magnet DC machine, as shown in Figure 7.1 Experimental and Theoretical Power Dissipation Comparison.
7.2. The cross-section revealed that the area of the copper was similar to the area of the iron and, therefore, the volume of the iron would be similar to the copper for a given rotor length.

![Diagram of a slotted permanent-magnet DC machine structure.](image)

**Figure 7.2** Slotted Permanent-Magnet DC Machine Structure [75].

- A rare-earth permanent-magnet material (NdFeB) was used to provide the magnetic field. Rare-earth material was chosen to minimise the device volume and mass. The typical magnetic-field strength supplied by the magnet was \( B_0 = 0.4 \, \text{T} \) [25].

- The volume of the permanent-magnet material was the same as the copper volume \( V \). This was estimated from the relationship between the electromagnetic machine parameters of cross-sectional air-gap area \( S_g \), and permanent-magnet area \( S_m \), and the air-gap length \( l_g \), and permanent magnet length \( l_m \). To maximise the magnetic energy supplied to the air-gap and minimise the permanent magnet volume, these parameters were related by [76],

\[
\frac{S_g \, l_m}{S_m \, l_g} = \frac{B_m}{\mu_0 \, H_m},
\]

(7.6)

where \( (B_m) \) and \( (H_m) \) were the magnetic flux-density and field-strength for the maximum energy product, respectively. The ratio \( \frac{B_m}{\mu_0 \, H_m} \) was around 0.64 for the rare-earth (NdFeB) permanent magnet material (Grade N30M). The area \( (S_g) \) would typically be smaller than \( (S_m) \) due to the flux-density within the iron pole-pieces being higher than...
the flux-density supplied by the permanent magnets. In this case, if the area ratio was \( S_g / S_m = 0.8 \), the maximum energy product occurred when the volume of the permanent magnets equalled the volume of the air-gap (which approximately equalled the volume of the conducting material, \( V \)).

- An impedance-matching regenerative energy interface was used to provide the damping and energy regeneration control of the electromagnetic damper. According to the analysis in Chapter 5, Section 5.3.5, for large battery voltages, the energy regeneration efficiency \( E_{\text{REG}} \) and normalised damping ratio \( F_{\text{DAMP}} \) were related by,

\[
E_{\text{REG}} = \left( 1 - F_{\text{DAMP}} \right). 
\] (7.7)

- The energy cost due to the power-electronic components for the regenerative energy interface, was considered negligible compared to the other energy costs. This was mainly due to the relatively small mass of the power-electronic components relative to the overall regenerative damper mass.

- The energy required to produce the regenerative damper support and rack-and-pinion type mechanism material was equivalent to the production energy of the original damper material. This was due to the relatively similar mass, and material composition (typically steel and aluminium) of the two systems. Therefore, the production energy of the support material was neglected in this analysis, and the mass added to the vehicle system was due to the copper \( M_{\text{COND}} \), iron pole-pieces, and permanent-magnets of the electromagnetic damper.

- Secondary material production costs were used for the copper and iron material production estimates.

Table 7.2 gives the vehicle system, damping and drive-cycle specifications for the performance analysis in this section.
### Table 7.2 Damping, Vehicle and Drive-Cycle Specifications.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kilometres travelled per Year [73]</td>
<td>-</td>
<td>14,300</td>
<td>km</td>
</tr>
<tr>
<td>Average Vehicle Age [73]</td>
<td>-</td>
<td>10.6</td>
<td>years</td>
</tr>
<tr>
<td>Proportion Urban / Non Urban Driving [34]</td>
<td>-</td>
<td>0.55</td>
<td>-</td>
</tr>
<tr>
<td>Suspension Damping Coefficient [30]</td>
<td>cv</td>
<td>1400</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Electromagnetic Damping Conducting Mass</td>
<td>MCOND</td>
<td>0.30</td>
<td>kg</td>
</tr>
<tr>
<td>Normalised Damping Coefficient</td>
<td>FDAMP</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>Vehicle Velocity - Urban</td>
<td>VURB</td>
<td>60</td>
<td>km/h</td>
</tr>
<tr>
<td>Vehicle Velocity - Non Urban</td>
<td>VRUR</td>
<td>100</td>
<td>km/h</td>
</tr>
<tr>
<td>Urban Stop-Starts Per Kilometre[74]</td>
<td>-</td>
<td>1.25</td>
<td>stops/km</td>
</tr>
<tr>
<td>Non-Urban Stop-Starts Per Kilometre [74]</td>
<td>-</td>
<td>0.097</td>
<td>stops/km</td>
</tr>
<tr>
<td>Rare-Earth Material (NdFeB) Production Energy</td>
<td>-</td>
<td>200,000</td>
<td>kJ/kg</td>
</tr>
</tbody>
</table>

In the course of this research investigation, it was not possible to obtain information for the material production energy of the rare-earth alloy (Nd$_2$Fe$_{14}$B), used as the permanent magnet material in the regenerative damper. The relative mass of the neodymium (Nd), iron (Fe), and boron (B), are 26.3, 72.7 and 1.0 per cent, respectively. Although the (primary) production energy of iron is 34,000 (kJ/kg) it was believed that, due to its relative scarcity and difficulty of extraction, the production energy of neodymium would be considerably more than this. The rare-earth material (NdFeB) production energy was chosen as 200,000 (kJ/kg) for this example, however, an analysis of the relationship between performance and the rare-earth production energy is presented in Section 7.3.2. Further research is required for an accurate estimate of this production energy.

From the vehicle and drive-cycle parameters and the assumptions previously given, it was possible to estimate the overall cost of the regenerative damping system for a typical vehicle application. A detailed tabulation of the cost-analysis evaluation is given in Appendix E, Table E.1. A summary of the energy cost-analysis is given in Table 7.3.
Table 7.3 Cost-Analysis Summary for a Typical Regenerative Vehicle Damping Application.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
<th>Percentage of Total Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suspension Regeneration Urban</td>
<td>3,539 kJ</td>
<td>kJ</td>
<td>22.2%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>4,633 kJ</td>
<td>kJ</td>
<td>29.1%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ($B_T$)</td>
<td>8,172 kJ</td>
<td>kJ</td>
<td>51.4%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>1,274 kJ</td>
<td>kJ</td>
<td>8.0%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>597 kJ</td>
<td>kJ</td>
<td>3.8%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>4,570 kJ</td>
<td>kJ</td>
<td>28.7%</td>
</tr>
<tr>
<td>Total Material Production Energy, ($C_p$)</td>
<td>6,440 kJ</td>
<td>kJ</td>
<td>40.5%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>1,100 kJ</td>
<td>kJ</td>
<td>6.9%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>194 kJ</td>
<td>kJ</td>
<td>1.2%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ($C_M$)</td>
<td>1,294 kJ</td>
<td>kJ</td>
<td>8.1%</td>
</tr>
<tr>
<td>Total Costs, ($C_T = C_p + C_M$)</td>
<td>7,734 kJ</td>
<td>kJ</td>
<td>48.6%</td>
</tr>
<tr>
<td>Total Energy (= $B_T + C_T$)</td>
<td>15,907 kJ</td>
<td>kJ</td>
<td>100.0%</td>
</tr>
<tr>
<td>Total Energy Cost, ($Cost_T = B_T - C_T$)</td>
<td>438 kJ</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

From the cost-analysis results in Table 7.3 the findings indicated that, for this example, regenerative vehicle suspension was feasible, with an overall benefit (per year) of ($Cost_T = 438$ kJ/year). The findings also indicated that the main energy cost of the regenerative suspension was from the material production energy (40.5 per cent). There was a smaller energy cost from the added vehicle mass (8.1 per cent).

At the outset of the regenerative damping research program, it was recognised that there could be no absolute answer to the performance question. This was due to the large number of dimensions to the problem. These dimensions included the particular type of regeneration scheme used, the type of vehicle, vehicle drive-cycle, and so on. Therefore, the following sections document an investigation undertaken to determine the relationship between the system performance and the variable system parameters. The performance of regenerative vehicle damping was related to the parameters of the electromagnetic conducting mass (Section 7.3.3), normalised damping (Section 7.3.4), and vehicle drive-cycle (Section 7.3.5). In Section 7.3.2, an analysis of the effect of the permanent-magnet production energy on regenerative damping performance is also presented.
7.3.2. Effect of Permanent-Magnet Material Production Energy

It has already been noted that in this research program, it was not possible to determine an exact estimate of the production energy of the rare-earth magnetic material (NdFeB). Therefore, this section documents an analysis into the relationship between the regenerative system performance and the permanent-magnet production energy. In this section, all damping, vehicle and drive-cycle parameters besides the permanent-magnet production energy were the same as given in Table 7.2, and the performance results are also tabulated in Appendix E, Table E.2

Figure 7.3 shows the relationship between the regenerative performance and the permanent-magnet material production cost. In this example, the mass of the permanent-magnet material per damper was 0.24 (kg). The results indicated that the performance of regenerative damping decreased for an increase in the magnet production energy, and that the system was feasible for a permanent-magnet production energy below approximately 220,000 (kJ/kg). In comparison to the material production cost of iron (34,000 kJ/kg), the energy for the permanent-magnet production cost was quite high. However, the actual permanent-magnet production energy may be of this magnitude, due to the relative scarcity and difficulty of extraction [77].
7.3.3. Effect of Conductor Mass

This section documents an investigation into the relationship between regenerative system performance and the electromagnetic conductor mass ($M_{\text{cond}}$). This analysis was undertaken because, according to the cost-analysis methodology, a parameter that was variable, was the conductor mass. In this section, all damping, vehicle and drive-cycle parameters besides the conductor mass were the same as given in Table 7.2. The results for this section are also tabulated in Appendix E, Table E.3.

The cost-analysis results in Figure 7.4 and Figure 7.5 indicated that the performance of regenerative vehicle damping increased for a reduction in the mass of the conducting material. Figure 7.4 reveals that the energy cost, ($C_T = C_M + C_p$) reduced to zero for a conductor mass approaching zero. This was because the permanent-magnet and iron mass were proportional to the conductor mass and, therefore, as the conductor mass approached zero, both the material production energy and added-mass energy reduced to zero.

![Figure 7.4 Percentage of Total Energy - Conductor Mass.](image-url)
The results in Figure 7.5 indicated that the regenerative system performance increased for a decrease in conductor-mass, and was feasible for the conductor-mass lower than approximately $M_{\text{COND}} = 0.3$ kg.

![Figure 7.5 Regenerative Performance - Conductor Mass.](image)

For low values of conductor mass, however, the gear-ratio ($\alpha$), became large, and approached infinity when the conducting mass ($M_{\text{COND}}$) reduced to zero (the gear-ratio as a function of conductor-mass is tabulated in Table E.3). A large gear-ratio may cause problems with respect to the device construction. This was due to the requirement for a small gear-radius or large conductor-radius for high gear-ratios. For this system, however, a conductor mass ($M_{\text{COND}} = 0.2$ kg), led to a gear-ratio ($\alpha = 3.67$), and a conductor mass ($M_{\text{COND}} = 0.3$ kg), led to a gear-ratio ($\alpha = 3.00$).

The rotating-mass ($m_r$) of the damper was independent of the conductor mass, however, and, for this example was ($m_r = 10.79$ kg). The rotating-mass remained constant because, as the gear-ratio increased, the actual device mass reduced. This was important with respect to maintaining adequate vehicle dynamics (see Section 6.4.1). To maximise the performance of the regenerative damping system it was, therefore, necessary to minimise the conductor mass. However, this would be limited by the allowable gear-ratio.
7.3.4. **Effect of Normalised Damping Ratio.**

The previous performance analysis assumed that the normalised damping-ratio was specified as \( F_{\text{DAMP}} = 0.5 \). To maximise the regenerative damping performance, however, this was not necessarily the optimum damping-ratio. Therefore, this section documents an investigation into the relationship between regenerative system performance and the normalised damping-ratio. In this section, all damping, vehicle and drive-cycle parameters, besides the normalised damping ratio, were the same as given in Table 7.2. The performance results are also tabulated in Appendix E, Table E.4.

Figure 7.6 and Figure 7.7 give the results of this analysis. The results in Figure 7.6 indicate that, as the normalised damping-ratio approached unity, the proportion of regenerated energy to the total energy approached zero and, therefore, the system performance reduced. This was because, for this condition, the energy regeneration efficiency \( E_{\text{REG}} \) reduced to zero.

![Figure 7.6 Percentage of Total Energy - Normalised Damping Ratio.](image)

The results in Figure 7.7 indicate that the regenerative system was feasible for a normalised damping ratio below approximately \( F_{\text{DAMP}} = 0.5 \).
For this analysis, the conducting mass \((M_{\text{COND}})\), remained constant. Therefore, for a low normalised damping-ratio, the gear-ratio had to increase to provide the required vehicle damping coefficient \((c_v)\). However, the increase in gear-ratio (for a given conducting mass) led to an increase in the rotating-mass of the damper (which is also shown in Table E.4). The rotating-mass approached infinity for zero normalised damping. To maximise the performance of the regenerative damping system it is, therefore, necessary to minimise the normalised damping ratio. However, this will be limited by the allowable rotating mass of the vehicle damper.

### 7.3.5. Effect of Vehicle Drive Cycle

Using the cost function analysis, it was possible to estimate the effect of the vehicle drive-cycle on the regenerative damping performance. Therefore, this section documents an analysis into the relationship between the regenerative system performance and the vehicle drive-cycle. In this section all damping, vehicle and drive-cycle parameters
besides the vehicle drive-cycle, were the same as given in Table 7.2, and the performance results are also tabulated in Appendix E, Table E.5 and Table E.6.

Figure 7.8 and Figure 7.9 give the results of an analysis into the effect of vehicle kilometres travelled per year on the regenerative system performance, and Figure 7.10 gives the results of an analysis into the effect of the proportion of urban and non-urban driving. Figure 7.8 indicates that the proportion of the regenerated energy and added-mass energy approached zero as the vehicle kilometres reduced. This was because, when the vehicle did not travel, no energy was regenerated and, also, there was no energy lost due to accelerating and decelerating the added mass.

![Figure 7.8 Percentage of Total Energy - Vehicle Kilometres Traveled Per Year.](image)

The results in Figure 7.9 indicate that the regenerative system performance increased for an increase in vehicle kilometres travelled per year. This was due to an increase in energy gained from regenerative damping for a constant material production cost. The system was feasible when more than approximately 13,000 (km) were travelled per year².

² This figure is comparable to the average vehicle kilometres travelled per year of 14,300 (km), estimated by the Australian Bureau of Statistic (ABS) [73].
Another vehicle drive-cycle parameter that influenced the system performance was the proportion of urban to non-urban road use. In the previous analysis, it was assumed that urban road use accounted for 55 per cent of the total kilometres travelled [34]. Shown in Figure 7.10 is the effect of road surface use on the total cost of regenerative vibration damping. The average number of kilometres travelled per year was 14,300 (km).

Figure 7.9 Regenerative Performance - Vehicle Kilometres Travelled Per Year.

Figure 7.10 Regenerative Performance - Proportion of Urban and Non-Urban Driving.
The results in Figure 7.10 indicate that the system performance increased for an increase in the proportion of kilometres travelled on non-urban roads. This was mainly due to the increased energy gained from the suspension due to the higher vehicle velocity and reduced quality of the road surface for the non-urban road-surfaces. The results in this section indicated that the regenerative system performance was highest for an increase in vehicle kilometres travelled per year, as well as for an increase in the proportion of non-urban driving.

7.4. Conclusions

The analysis documented in this chapter proposed a method to obtain an objective measure of the performance of regenerative damping in vehicle systems. The methodology was based on the premise that a regenerative damper has the potential to recover energy and provide a benefit to the vehicle system. However, the effect of the overall regenerative system would also be detrimental due to factors such as additional weight and material cost.

The accuracy of the performance analysis depended on the assumptions given in Section 7.3. The assumptions were used to simplify the overall analysis and would, therefore, lead to an understanding of the relationship between system parameters and the overall performance. The estimates of system parameters, such as the permanent-magnet and iron volume of the electromagnetic damper, should be further analysed to determine the limitations of the assumptions used in this analysis. Further analysis also needs to be undertaken to increase the understanding of the actual energy regenerated in regenerative vehicle dampers. For accuracy, this should be analysed in an actual regenerative vehicle system and on actual road-surfaces.

The performance of regenerative damping also depended on the allowable gear-ratio of the rack-and-pinion mechanism, as well as the allowable rotating-mass of the vehicle
damper. Although a method of determining the allowable rotating mass was proposed in Chapter 6, further research is necessary to more accurately determine the limits of these parameters. This will result in a more accurate estimate of the regenerative system performance. The analysis was able to provide an estimate of the system performance as a function of factors such as damper mass, normalised damping-ratio and the vehicle drive-cycle. It was determined that regenerative system performance increased for:

- A reduction in the permanent-magnet production energy of the electromagnetic damper,
- A reduction in the mass of the conducting material of the electromagnetic damper,
- A reduction in the normalised damping ratio ($F_{DAMP}$),
- An increase in vehicle kilometres travelled per year, and
- An increase in the proportion of non-urban to urban driving.

Using the cost-analysis methodology, it was also possible to determine how to maximise the performance of the regenerative damper system with respect to device construction, and the limitations of integrating the regenerative system into a vehicle. For instance, to maximise the regenerative damping performance, it was necessary to minimise the conductor mass. The minimisation of conductor mass was limited by the allowable gear-ratio ($\alpha$). Also, to maximise the regenerative damping performance, it was necessary to minimise the normalised damping ratio ($F_{DAMP}$). The minimisation of the normalised damping ratio was limited by the allowable rotating mass ($mr$), of the vehicle damper.

From the cost-analysis presented in this chapter, it was possible to determine the limitations of regenerative vehicle damping as well as estimate the system performance. The overall results revealed that regenerative vehicle damping was feasible provided that certain design parameters such as damper mass, and damping control were constrained.
"The end is never as satisfying as the journey. To have achieved everything but to have done so without integrity and excitement is to have achieved nothing."

Source Unknown

8.1. Overview

This chapter presents the final discussion and conclusions for this research program. In this chapter, the main research findings are related to the project objectives and the research 'problem statement'. The limitations of the investigation are also identified and, from this, recommendations are given for future research in the field of regenerative vibration damping.
8.2. Discussion of the Overall Research Program

This research program was undertaken according to the following problem statement:

"To investigate various means of recovering energy from damped, vibrating systems; to select the most promising alternative and to assess the performance of such a system for use in vehicle suspension systems."

The investigation of regenerative vibration damping stemmed from the need for continuous improvement of vehicle efficiency, and the potential benefits from the development of regenerative vehicle suspension. At the outset of the research program, it was noted that there had already been several previous research investigations into regenerative vibration damping and its application for vehicle suspension systems. However, when applied to vehicle systems, previous regenerative damping research had focused on energy regeneration as a means of improving the efficiency of semi-active damping systems, rather than for regenerative dampers in their own right. Following an extensive review of literature in the field, the research program evolved into what was believed to be the first attempt at specifically investigating regenerative suspension with the objective of establishing its potential performance for vehicle systems.

The research documented in this thesis began with a broad overview of the field and, subsequently, the research concentrated on more specific areas that were uncovered as major determinants of the regenerative system performance. The research culminated with the regenerative performance analysis in Chapter 7. This analysis was based on the overall research findings as well as externally sourced data. Therefore, with respect to the overall project objectives and the thesis 'problem statement', an attempt was made to determine the most promising alternative regenerative damping scheme and to assess the viability of such a system for use in vehicle suspension systems.
Irrespective of how detailed or inclusive the performance analysis documented in this thesis was, however, there could be no definite answer to the performance question. This was mainly due to the large number of dimensions to the problem, including the type of regeneration scheme used, the type of vehicle, vehicle drive cycle, and so on.

It was also recognised that this investigation attempted to determine the "most promising", rather than the "optimum", regenerative damping system. It was not possible to determine the optimum design due to areas of research in which further investigation was needed. An example of this was for the analysis of Magneto-Hydro-Dynamic (MHD) energy conversion devices in Chapter 3. It was possible that MHD devices offered advantages over conventional electromagnetic dampers. However, to verify this, further research was needed, which was considered beyond the scope of the present research program. Another example was for the investigation of material production estimates for the cost-analysis in Chapter 7. To increase the accuracy of the performance estimates, a more accurate measure of the production energy of rare-earth permanent magnet material was required. Due to the, somewhat separate issues (compared with the areas of research covered in this thesis) of mineral extraction, refinement, and processing involved to estimate the material production energy of this permanent magnet material, it was also suggested for future research.

There were also external influences which could influence the performance of a regenerative energy system, and it was not possible to include all these potential influences in the performance analysis presented in this thesis. Some of these factors included:

- The perceived environmental need for the regenerative system,
- The marketing potential of the regenerative system,
- The value or cost of energy as a global commodity.

The consequence of the environmental impact of vehicles, particularly through NOX and 'greenhouse' gas emissions, could influence the performance of regenerative damping in vehicle systems. Environmental factors were expected to continue to drive government...
regulations, such as the California Zero-Emission Vehicle (ZEV) mandate of 1996. This would, in turn, influence both the investment in environmental vehicle technology and, overall, the kind of vehicles that would travel on the world's roads.

The value of energy is another factor that would influence the performance of regenerative vibration damping. The 'energy crisis' of the 1970s, which resulted from the perceived shortage of fossil-fuel, led to a dramatic increase in the price of fuel. An increase in the fuel price, which referred to an increase in the energy value, would directly influence the value of a regenerative energy system.

The issues of (large-scale) device production, and the influence of technological advances on regenerative system performance, were not investigated in this research. This was because of the difficulty in determining the influence of these parameters at an early stage of system development, such as for when this research was undertaken. For instance, it was expected that, if a regenerative damping system was produced for a commercial market, a higher production quantity would lead to a lower production cost per device. This would largely be a result of a lower overhead costs [37]. Also, technological advances (which would, to some extent, be driven by a higher demand) would improve the overall system performance. As an example, the performance of permanent-magnet technology was constantly improving [78], with then current rare-earth magnets having an energy-product ranging from two to twelve times that of traditional ferrite magnets [79]. The improvement in the magnet energy-product would lead directly to a reduction in mass, volume and cost of electromagnetic damping device. This, in turn, would lead to improved device performance and improved system feasibility. Improvement would also be expected with other system components such as for the power-electronic devices used in the regenerative interface.
8.3. Research Findings and Contributions

A significant outcome of this research was the investigation into the overall performance and feasibility of regenerative damping for vehicle suspension systems. A methodology was proposed to obtain an objective performance measure, and was based on a cost-analysis model for the overall regenerative vehicle system. From the performance analysis it was possible to estimate the system performance as a function of factors such as damper mass, normalised damping-ratio and the vehicle drive-cycle. It was determined that regenerative system performance increased for:

- A reduction in the permanent-magnet production energy,
- A reduction in the mass of the conducting material of the electromagnetic damper,
- A reduction in the normalised damping ratio,
- An increase in vehicle kilometres travelled per year, and
- An increase in the proportion of non-urban to urban driving.

Using the cost-analysis methodology, it was also possible to determine how to maximise the performance of the regenerative damper system with respect to device construction, and the limitations of integrating the regenerative system into a vehicle. For instance, to maximise the regenerative damping performance, it was found that it was necessary to minimise the conductor mass. However, the minimisation of conductor mass was limited by the allowable gear-ratio. Additionally, it was found that, to maximise the regenerative damping performance, it was necessary to minimise the normalised damping ratio. However, the minimisation of the normalised damping ratio was limited by the allowable rotating-mass of the vehicle damper.

From the cost-analysis methodology, it was possible to determine the limitations of regenerative vehicle damping as well as estimate the overall system feasibility. The results revealed that regenerative vehicle damping was feasible provided that certain design parameters such as damper mass, and damping control were constrained.
The results of this analysis were included in a paper published in the Proceedings of the 1999 Society of Automotive Engineers Australasia, Young Engineers Conference, entitled: "Cost Function Analysis of Regenerative Vehicle Systems".

There were several other important findings made throughout the research program in relation to the optimal design of regenerative damping systems. These findings are described below.

(i) Literature Survey (Chapter 2)

A survey was conducted to determine the current understanding in the field of regenerative vibration damping, electromagnetic devices and vehicle suspension research. The findings revealed that, although electromagnetic devices had been used as regenerative dampers, there was no evidence of attempts to optimise their design and find the most suitable form of device for use as a regenerative damper.

The results of the previous investigations into electromagnetic regenerative damping revealed several limitations with respect to the regenerative interface systems. The previous regeneration interfaces were mainly passive systems, and used either diodes or relays for energy transference to the storage device. It was found that a disadvantage of the passive regenerative systems was that there was no damping and no regeneration when the electromagnetic damper potential was lower than the storage device potential.

It was found that rotating dampers could be used as regenerative dampers, and had an advantage of mechanical amplification. It was also revealed that the effect of the rotating mass may affect the dynamics of the vehicle system.
The review investigated previous research in the area of cost-analysis, and revealed several important factors for a cost-analysis for vehicle systems. This included the need for an investigation into material production, assembly, maintenance, disposal, transportation and operating energies of the system. An example of the life-cycle analysis of a vehicle system was given, and revealed that the operating efficiency and material use were major determinants of life-cycle energy.

(ii) Principles of Regenerative Damping (Chapter 3)

In Chapter 3, an analysis was presented into the overall principles of regenerative damping and the requirements for such a system for implementation in an application such as vehicle suspension. It was revealed that two main requirements of the regenerative damping system were to maximise energy regeneration and maintain adequate damping. Also described in this analysis were the potential advantages of converting the vibrational energy into electrical energy. The advantage of electrical energy mainly stemmed from its use for subsequent applications, as well as its ability to be conveniently stored for long periods of time through the use of devices such as secondary batteries.

A number of devices were also discussed, with respect to their potential for converting the mechanical, vibrational energy into electrical energy, especially for a regenerative damping application. It was found that, due to their energy conversion efficiency, high compliance and viscous damping properties, electromagnetic devices had the most promise as regenerative damping elements.
(iii) Electromagnetic Damping Devices (Chapter 4)

The investigation documented in Chapter 4 analysed the major types of electromagnetic devices, with the objective of determining which one had the most potential as regenerative dampers. The analysis revealed that, due to their damping response, two particular devices had the ability to perform as regenerative dampers. These were the DC generator, based on the production of 'motional EMF' in the windings, and the AC synchronous generator, based on the production of 'transformer EMF' in the windings. The damping performance of the two devices was theoretically analysed. The results revealed that DC electromagnetic devices had the largest potential for use as regenerative dampers due to an inherently linear force-velocity characteristic and compatibility with an impedance-matching interface.

The results of this analysis led to a paper published in the Journal of Sound and Vibration, entitled: "Theoretical Comparison of Motional and Transformer EMF Device Damping Efficiency".

A generalised topology structure was also developed to evaluate the important design characteristics of the DC electromagnetic devices. With respect to the magnetic circuit design it was revealed that the minimisation of the pole-piece mass was an important design consideration of these devices. With respect to the electrical circuit design, a DC generator with a series circuit offered the advantage of increased internal resistance and high output voltage in comparison to parallel circuit designs.

The generalised topology also revealed that either rotating or linear electromagnetic machines may be used as regenerative dampers. The advantage of rotating dampers was that an increase in damping performance could be achieved for a given device mass, due to mechanical amplification with the rack-and-pinion mechanism. The disadvantage of rotating dampers occurred for
applications such as vehicle suspension in which the vehicle dynamics were degraded.

The results of this analysis were included in a paper published in the International Journal of Vehicle Design, entitled: "Electromagnetic Regenerative Damping in Vehicle Suspension Systems".

(iv) Regenerative Energy Interface (Chapter 5)

In Chapter 5, a proposal for an 'impedance-matching' regenerative interface was presented. This was developed in response to the limitations of the passive interface designs analysed by previous research investigations in the field of regenerative damping. Both theoretical and experimental analysis revealed that the interface had several advantages over the previous designs, with respect to the damping and regeneration performance.

The analysis revealed that, with relatively simple control, it was possible to control the regenerative interface to maximise energy regeneration, or to control the damping coefficient of the system. An experimental investigation was undertaken to determine the performance of the system for a more realistic situation. It was revealed that the experimental damping and energy regeneration reasonably followed the ideal theoretical model, and that the discrepancies were mainly caused by non-ideal parasitic effects, and non-ideal switching of the power-electronic devices. It was also shown that, under the experimental conditions, the damping and regeneration performance of the impedance-matching regenerative system was superior to the previous, passive design.
(v) Vehicle Integration of Regenerative Damping (Chapter 6)

In Chapter 6 an investigation was undertaken with respect to two issues which had important implications for determining the performance of regenerative damping in vehicle systems. The first issue was due to the amount of energy dissipated in vehicle suspension systems and the second was due to the use of rotating dampers in vehicle suspension systems.

A theoretical analysis of vehicle suspension energy dissipation was undertaken by modelling the vehicle dynamic system and the road surface. The theoretical results were compared against experimental energy dissipation obtained in other research investigations. The comparison revealed that the dissipation estimates between theory and experiment coincided reasonably accurately. The results of this analysis were included in the proceedings of the 32\textsuperscript{nd} International Symposium on Automotive Technology and Automation (ISATA), entitled: "Regenerative Vehicle Suspension".

The analysis also investigated the use of rotating dampers in vehicle suspension. This benefit of using rotating dampers stemmed from the mechanical amplification gained from the use of a rack-and-pinion type mechanism. This led to increased damping and energy regeneration for a given damper size, volume and cost. The disadvantage of rotating dampers stemmed from their effect on the suspension dynamic response. It was shown that, although the isolation response was degraded between approximately 5 and 12 (Hz), a two degree-of-freedom model attenuated the high frequency component of the input disturbance. This result differed from the previous analysis by Ryba [26] in which, for a one degree-of-freedom model, the rotating damper system had a finite high-frequency isolation response.

A method to negate the effects of the rotating mass was also presented. It was shown that it was possible to negate the effects of the rotating mass by adding
appropriate damping and spring elements to the dynamic system. It was also found, however, that to totally negate the effect of the rotating-mass, the damping and spring parameters were relatively high and, therefore, the feasibility of implementing the added damper and spring combination, as a solution to the rotating damper response, may be limited.

The results of this analysis were also included in the paper published in the International Journal of Vehicle Design, entitled: "Electromagnetic Regenerative Damping in Vehicle Suspension Systems".

The limits of the allowable rotating-mass in vehicle suspension were also estimated. This was achieved by theoretically analysing the human sensitivity to vibration within the vehicle system. A typical example was given for an 8 Hour, reduced comfort boundary, for a vehicle traversing a road classified 'average' or better. In this case, a rotating mass \((mr = 20.0 \text{ kg})\) was acceptable, but a rotating mass \((mr = 30.0 \text{ kg})\) was not acceptable. The results assisted in determining the maximum allowable damping coefficient for a given electromagnetic damper. Therefore, it also assisted in determining the energy regeneration efficiency and the mass, volume and cost of the regenerative damper.

### 8.4. Proposal for Future Research

A major objective of the regenerative damping research program was to investigate the overall performance of regenerative damping for vehicle suspension systems. The objective was to present an analysis that was detailed enough to give an accurate estimate of the system performance, yet have ease of computation and be able to indicate the relationship between system parameters and the overall performance. For the investigation undertaken in this research, it was necessary to make several assumptions concerning the electromagnetic damping process and the vehicle drive-cycle. The assumptions included the estimates of the amount of permanent-magnet and
iron pole-piece material needed for the electromagnetic damper, and assumptions for the vehicle drive-cycle data. As the accuracy of the performance analysis depended on those assumptions, further research should be undertaken to determine the limitations of the assumptions.

There were three main parameters that remained uncertain following the analysis into regenerative damping performance. These issues were identified in Chapter 7, and were the permanent-magnet production energy, the allowable rotating-mass, and the allowable gear-ratio of the rack-and-pinion type mechanism. Further research should, therefore, be undertaken to obtain a more accurate estimate of these parameters.

During the course of the literature review, it was found that there had been several investigations analysing regenerative damping systems [14, 15] using a bond-graph methodology. A primary objective was to maximise the energy efficiency of the regenerative damping schemes, which was also a primary objective of the research documented in this thesis. However, the research documented in this thesis focused specifically on the 'electrical' regenerative interface. It would provide a worthwhile extension to this research by representing, and investigating, the regenerative interface, developed in this research, using a 'bond-graph' methodology. This investigation would allow a comparison of the system with the equivalent mechanical and hydraulic systems. It may also reveal limitations to the 'electrical' system analysis, presented in this thesis.

To increase the understanding of the electromagnetic regenerative damping process, and to further resolve the limitations of such a system, it would also be beneficial to implement the regenerative system within an actual vehicle. This would help determine, more accurately, factors such as the electromagnetic damper construction. An actual regenerative damping system would also give a more accurate estimate of the amount of energy regenerated for a typical driving situation.

One final issue, worthy of discussion, was the relationship between the regenerative damping research documented in this thesis, and the field of semi-active damping. Semi-active damping had been shown to provide improvement over passive damping
[80] without the power, complexity and bandwidth requirements of active suspension [81]. It was revealed in the Literature Review chapter, that there had been several research investigations into the use of semi-active, regenerative vibration damping. The previous investigations did not use an active regenerative interface such as the 'impedance-matching' interface developed for the investigation in this thesis. It was shown in Chapter 5 that it was possible to control the damping coefficient of the regenerative damper by controlling the normalised damping ratio. A worthwhile extension to this research would be to use the 'impedance-matching' regenerative interface to improve the vehicle dynamics as well as regenerate the vibration energy. Such a system could be beneficial for applications such as electric vehicles, in which consumers want the environmental advantages of electric vehicle technology, as well as to experience improved ride-comfort.

8.5. Final Comments

In summary, the research documented in this thesis provided an insight into the operation and performance of regenerative vehicle systems. The research led to a number of contributions to the field, and revealed the importance of developing an objective measure of the regenerative system performance.
REFERENCES


REFERENCES


REFERENCES


REFERENCES


APPENDIX

A

PUBLICATIONS ARISING DURING THE PhD RESEARCH PROGRAM

A.1 Journal Articles and Conference Proceedings


A.2 Contributions to Text Books on PhD Related Work


THIS section documents the experimental analysis performed to determine the specifications of the experimental apparatus. The results were for the experimental investigations documented in Chapter 5, Sections 5.4, 5.5 and 5.6.

### B.1 Regenerative Interface Operation

**Figure B.1** Current Path Diagrams for each Circuit State.
### State 1
Closed-Circuit, $E_C > 0$
- $i_L > 0$ (rising in magnitude)
- $v_{DS1} = v_{DS2} = v_{DS3} = v_{DS4} = 0$
- $i_{DS1} = i_{DS4} = -i_L/2$
- $i_{DS2} = i_{DS3} = i_L/2$
- $v_D = -V_{OUT}$
- $i_D = 0$
- $i_{BATT} = -i_C$ (falling)

### State 3
Closed-Circuit, $E_C < 0$
- $i_L < 0$ (rising in magnitude)
- $v_{DS1} = v_{DS2} = v_{DS3} = v_{DS4} = 0$
- $i_{DS1} = i_{DS4} = -i_L/2$
- $i_{DS2} = i_{DS3} = i_L/2$
- $v_D = -V_{OUT}$
- $i_D = 0$
- $i_{BATT} = -i_C$ (falling)

### State 2
Open-Circuit, $E_C > 0$
- $i_L > 0$ (reducing in magnitude)
- $v_{DS2} = v_{DS3} = V_{OUT} - v_D$
- $v_{DS1} = v_{DS4} = 0$
- $i_{DS1} = i_{DS3} = 0$
- $i_{DS1} = i_{DS4} = i_L$
- $v_D = 0.85$ V
- $i_D = i_L$
- $i_{BATT} = i_D - i_C$ (rising)

### State 4
Open-Circuit, $E_C < 0$
- $i_L < 0$ (reducing in magnitude)
- $v_{DS1} = v_{DS4} = V_{OUT} - v_D$
- $v_{DS2} = v_{DS3} = 0$
- $i_{DS1} = i_{DS4} = 0$
- $i_{DS2} = i_{DS3} = i_L$
- $v_D = 0.85$ V
- $i_D = -i_L$
- $i_{BATT} = i_D - i_C$ (rising)

**Figure B.2** State-Diagram for the Impedance-Matching Regeneration Circuit.
B.2 Experimental Apparatus

Figure B.3 Experimental Circuit Diagram for the Steady-State Analysis.

Figure B.4 Experimental Circuit Diagram for the Dynamic Analysis.
### Appendix B - Regenerative Interface Analysis Data

#### Table B.1 Experimental Interface Specifications

<table>
<thead>
<tr>
<th>Component</th>
<th>Brand</th>
<th>Model No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Computer (Hardware)</td>
<td>IBM -Compatible</td>
<td>80486SX</td>
<td>33 (MHz) clock speed 4 Mbyte RAM</td>
</tr>
<tr>
<td>Computer (Software)</td>
<td></td>
<td></td>
<td>MS-DOS Ver. 6.22 Borland C++ Ver. 3.1</td>
</tr>
<tr>
<td>Computer I/O Card</td>
<td>Industrial Computer</td>
<td>ML16-P</td>
<td>8x 8-bit Digital Inputs 8x 8-bit Digital Outputs 8x 8-bit Analog Inputs 3x 16-bit programmable counters (8253-5)</td>
</tr>
</tbody>
</table>
| Transistor Driver  | International Rectifier | IR2110   | $t_{\text{ON/OFF}}$ (typ.) 120 & 94 ns  
|                  |                        |           | $V_{\text{OFFSET}}$ 500V max.                                               |
| Rotary Potentiometer | Spectrol - 534-1-501 |           | Resistance: 500 (Ω)  
|                   |                        |           | Resistance Tolerance: ±5 (%)  
|                   |                        |           | Linearity Tolerance: ±0.25 (%)  
|                   |                        |           | Rotation: 3600°  
|                   |                        |           | Start. Torque (Max.): 3.6×10^-4 (kg.m)                                      |
| Current Transducer | LEM HA 10-NP/SP1       |           | Accuracy: ±1 (%)  
|                   |                        |           | Linearity: ±1 (%)  
|                   |                        |           | Frequency Bandwidth (-3dB): 50 (kHz)                                        |

#### Table B.2 Experimental Electrical Component Specifications

<table>
<thead>
<tr>
<th>Component</th>
<th>Brand</th>
<th>Model No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driving Machine</td>
<td>Electro-craft Servo Products</td>
<td>S-642-1B</td>
<td>DC machine 60 Volt Max.</td>
</tr>
<tr>
<td>Tacho-Generator</td>
<td>Mabuchi RS-540 SH</td>
<td></td>
<td>12 Volt DC machine</td>
</tr>
<tr>
<td>Electromagnetic Damper</td>
<td>Electro-craft Servo Products</td>
<td>S-19-3B</td>
<td>DC machine 60 Volt Max.</td>
</tr>
<tr>
<td>Power Supply</td>
<td>Jaytech MP-3080</td>
<td></td>
<td>0-30 Volt, 2.5 Amp</td>
</tr>
<tr>
<td>Power Supply</td>
<td></td>
<td>MP-3080</td>
<td>0-15 Volt, 10.0 Amp</td>
</tr>
<tr>
<td>Storage Battery</td>
<td>CNB B38-6A</td>
<td></td>
<td>6 Volt, 13 A-h, Lead-Acid</td>
</tr>
<tr>
<td>Electronics:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Transistor (x4)</td>
<td>Harris Semiconductor</td>
<td>F15N06L</td>
<td>N-type Power FET 60 Watt, 15 Amp</td>
</tr>
<tr>
<td>Diode</td>
<td>Harris Semiconductor</td>
<td>RURP1560</td>
<td>600Volt, 15Amp</td>
</tr>
<tr>
<td>Capacitor</td>
<td>BHC/Aerovox</td>
<td>AL-10A102BB100</td>
<td>1000μF, 100 VDC</td>
</tr>
</tbody>
</table>
## Brand | Model Number | Range | Resolution (Max.) | Accuracy | Bandwidth
--- | --- | --- | --- | --- | ---
ISO-TECH | IDM101 | DC Volts (0 - 4V) | 1.0 (mV) | ± 0.7 (%) | -
 | | DC Volts (4 - 40V) | 10.0 (mV) | ± 0.5 (%) | -
 | | DC Current (0 - 400 mA) | 0.1 (mA) | ± 0.8 (%) | -
 | | DC Current (400mA - 10 A) | 10 (mA) | ± 0.5 (%) | -
Tektronix Digital Oscilloscope | TDS3052 | DC Voltage | 9-bit | ±[0.02x|reading| +0.15 div + 0.6 mV]$^1$ | 500 MHz
Tektronix Current Probe | P6302 | 0-20 A | - | within 3% of indicated current / division | 50 MHz

Table B.3 Voltage and Current Meter Specifications.

---

$^1$ Excludes offset accuracy.
B.3 Preliminary Experimental Analysis

The internal resistance of the storage battery used in the experimental investigation was experimentally evaluated. The storage battery used in the analysis was a 'CNB, B38-6A', 6 (V), 13 (A-h), lead-acid battery. To measure the internal battery resistance, the current-voltage relationship of the battery was analysed. To evaluate the storage battery resistance, the battery was charged with a constant current ($I_{\text{INITIAL}}$), and the steady-state terminal voltage ($V_{\text{INITIAL}}$) was measured. The current was then instantaneously reduced to zero, and the resulting battery terminal voltage was measured ($V_{\text{FINAL}}$). This is illustrated in Figure B.5.

![Figure B.5 Battery Voltage-Current Response.](image)

The results of this analysis are shown in Table B.4. It can be seen that the resistance varied from 0.25 to 0.275 (Ω).

<table>
<thead>
<tr>
<th>$I_{\text{INITIAL}}$ (A)</th>
<th>$V_{\text{INITIAL}}$ (V)</th>
<th>$V_{\text{FINAL}}$ (V)</th>
<th>Resistance (Ω)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.80</td>
<td>6.45</td>
<td>6.25</td>
<td>0.250</td>
</tr>
<tr>
<td>1.00</td>
<td>6.52</td>
<td>6.26</td>
<td>0.260</td>
</tr>
<tr>
<td>1.20</td>
<td>6.60</td>
<td>6.27</td>
<td>0.275</td>
</tr>
</tbody>
</table>

Table B.4 Battery Voltage and Current Response.

Therefore, for the purposes of the experimental analysis documented in the Chapter 5, the internal battery resistance was assumed to be 0.25 (Ω). The internal resistance (and also the internal voltage) of the battery varied as a function of the state-of-charge.
Therefore, for the experimental analysis, the state-of-charge was maintained at a constant level for each measurement.

To estimate the internal winding resistance of the electromagnetic damper, the voltage-current relationship of the windings was analysed. For this analysis, the current was measured as a function of an applied voltage to the machine terminals. To negate the influence of the internal winding EMF, the relative machine velocity was kept at zero by physically securing the rotor. To minimise the resistive effects of the machine brushes, the minimum resistance (maximum current) was recorded for the rotor at different angles. The analysis was undertaken for the applied voltages, 0.5, 1.0 and 1.5 (V), and the results are presented in Table B.5. The results from this analysis indicated that the internal winding resistance \( R_{\text{INT}} \) was approximately 1.06 (\( \Omega \)).

<table>
<thead>
<tr>
<th>Applied Voltage (V)</th>
<th>Measured Current (A)</th>
<th>Winding Resistance (( \Omega ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.50</td>
<td>0.47</td>
<td>1.06</td>
</tr>
<tr>
<td>1.00</td>
<td>0.95</td>
<td>1.05</td>
</tr>
<tr>
<td>1.50</td>
<td>1.41</td>
<td>1.06</td>
</tr>
</tbody>
</table>

Table B.5 Internal Machine Winding Resistance.

To estimate the internal winding inductance, a transient analysis was performed on the machine. The machine was analysed using the apparatus shown in Figure B.6.

It was possible to estimate the internal inductance of the electromagnetic damper windings by analysing the time response of the current, for an input voltage step-
response. The current was analysed using a current probe and oscilloscope. Once again, to negate the effects of the internal EMF of the electromagnetic device, the relative machine velocity was kept at zero by physically securing the rotor. To minimise the effects of the brush resistance, measurements were undertaken for different rotor angles. The switch was closed at time \( t = 0 \), and the time constant \( \tau \), was evaluated for the exponential current response, as shown in Figure B.7. The time constant was evaluated as 2.17 (ms). For the previously evaluated internal device resistance of \( R_{\text{INT}} = 1.06 \, \Omega \), the internal circuit inductance was, therefore, estimated as \( L = 2.30 \, \text{mH} \).

![Figure B.7 Transient Response - Circuit Current.](image-url)
**B.4 Steady-State Experimental Data**

The data in this section are for the experimental steady-state investigations documented in Chapter 5, Sections 5.4. The data in Table B.6 is the theoretical and experimental open-circuit machine voltage at the boundary of continuous and discontinuous conduction. Refer to Section 5.4.3, Figure 5.13 for a graphical representation.

<table>
<thead>
<tr>
<th>Duty-Cycle, $D$</th>
<th>0</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.8</th>
<th>0.9</th>
<th>1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Theoretical</td>
<td>7.05</td>
<td>6.36</td>
<td>5.67</td>
<td>4.98</td>
<td>4.29</td>
<td>3.59</td>
<td>2.88</td>
<td>2.17</td>
<td>1.45</td>
<td>0.73</td>
<td>0.73</td>
</tr>
<tr>
<td>Experimental</td>
<td>-</td>
<td>5.91</td>
<td>5.20</td>
<td>4.50</td>
<td>3.89</td>
<td>3.29</td>
<td>2.69</td>
<td>2.08</td>
<td>1.34</td>
<td>0.68</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table B.6 Machine Voltage at the Boundary of Continuous-Discontinuous Conduction.**

The data in Table B.7 is the experimental transistor drain current ($I_{\text{DRAIN}}$), source-drain voltage ($V_{\text{DS}}$) and effective source-drain resistance ($R_{\text{DS}}$). Refer to Section 5.4.4, Figure 5.15 for a graphical representation.

<table>
<thead>
<tr>
<th>$E_{\text{C}}=2.0$ (V)</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.8</th>
<th>0.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_{\text{DRAIN}}$ (A)</td>
<td>0.19</td>
<td>0.33</td>
<td>0.46</td>
<td>0.63</td>
<td>0.81</td>
<td>1.04</td>
<td>1.17</td>
<td>1.42</td>
<td>1.43</td>
</tr>
<tr>
<td>$V_{\text{DS}}$ (mV)</td>
<td>45</td>
<td>57</td>
<td>70</td>
<td>85</td>
<td>97</td>
<td>116</td>
<td>130</td>
<td>149</td>
<td>149</td>
</tr>
<tr>
<td>$R_{\text{DS}}$ (Ω)</td>
<td>0.24</td>
<td>0.17</td>
<td>0.15</td>
<td>0.13</td>
<td>0.12</td>
<td>0.11</td>
<td>0.11</td>
<td>0.10</td>
<td>0.10</td>
</tr>
<tr>
<td>$E_{\text{C}}=4.0$ (V)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$I_{\text{DRAIN}}$ (A)</td>
<td>0.42</td>
<td>0.66</td>
<td>1.01</td>
<td>1.35</td>
<td>1.75</td>
<td>2.02</td>
<td>2.44</td>
<td>2.59</td>
<td>3.20</td>
</tr>
<tr>
<td>$V_{\text{DS}}$ (mV)</td>
<td>96</td>
<td>115</td>
<td>143</td>
<td>172</td>
<td>210</td>
<td>228</td>
<td>267</td>
<td>281</td>
<td>339</td>
</tr>
<tr>
<td>$R_{\text{DS}}$ (Ω)</td>
<td>0.23</td>
<td>0.17</td>
<td>0.14</td>
<td>0.13</td>
<td>0.12</td>
<td>0.11</td>
<td>0.11</td>
<td>0.10</td>
<td>0.10</td>
</tr>
<tr>
<td>$E_{\text{C}}=6.0$ (V)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$I_{\text{DRAIN}}$ (A)</td>
<td>0.67</td>
<td>1.05</td>
<td>1.73</td>
<td>2.33</td>
<td>2.64</td>
<td>3.21</td>
<td>3.50</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$V_{\text{DS}}$ (mV)</td>
<td>168</td>
<td>181</td>
<td>244</td>
<td>294</td>
<td>318</td>
<td>370</td>
<td>395</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$R_{\text{DS}}$ (Ω)</td>
<td>0.25</td>
<td>0.17</td>
<td>0.14</td>
<td>0.13</td>
<td>0.12</td>
<td>0.11</td>
<td>0.11</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table B.7 Transistor Source-Drain Resistance - Impedance-Matching Interface.**
The data in Table B.8 is the measured transistor transition loss. The power loss for the 'on' to 'off' state (switching off) is defined as $P_{OFF}$. The power loss for the 'off' to 'on' state (switching on) is defined as $P_{ON}$. Refer to Section 5.4.5, Figure 5.18 for a graphical representation.

<table>
<thead>
<tr>
<th>$E_c$ (V)</th>
<th>Normalised Damping Coefficient, $F_{DAMP}$</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.8</th>
<th>0.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{OFF}$ (mW)</td>
<td>3.0</td>
<td>6.0</td>
<td>7.0</td>
<td>8.0</td>
<td>10.0</td>
<td>13.0</td>
<td>16.0</td>
<td>18.0</td>
<td>20.0</td>
<td></td>
</tr>
<tr>
<td>$P_{ON}$ (mW)</td>
<td>2.0</td>
<td>2.0</td>
<td>3.0</td>
<td>5.0</td>
<td>6.0</td>
<td>8.0</td>
<td>9.0</td>
<td>11.0</td>
<td>12.0</td>
<td></td>
</tr>
<tr>
<td>$P_{TOTAL}$ (mW)</td>
<td>5.0</td>
<td>8.0</td>
<td>10.0</td>
<td>13.0</td>
<td>16.0</td>
<td>21.0</td>
<td>25.0</td>
<td>29.0</td>
<td>32.0</td>
<td></td>
</tr>
<tr>
<td>$E_c$=4.0 (V)</td>
<td>$P_{OFF}$ (mW)</td>
<td>3.0</td>
<td>9.0</td>
<td>14.0</td>
<td>21.0</td>
<td>28.0</td>
<td>38.0</td>
<td>45.0</td>
<td>54.0</td>
<td>65.0</td>
</tr>
<tr>
<td>$P_{ON}$ (mW)</td>
<td>2.0</td>
<td>6.0</td>
<td>8.0</td>
<td>13.0</td>
<td>16.0</td>
<td>21.0</td>
<td>25.0</td>
<td>29.0</td>
<td>39.0</td>
<td></td>
</tr>
<tr>
<td>$P_{TOTAL}$ (mW)</td>
<td>5.0</td>
<td>15.0</td>
<td>22.0</td>
<td>34.0</td>
<td>44.0</td>
<td>59.0</td>
<td>70.0</td>
<td>83.0</td>
<td>104.0</td>
<td></td>
</tr>
<tr>
<td>$E_c$=6.0 (V)</td>
<td>$P_{OFF}$ (mW)</td>
<td>3.0</td>
<td>12.0</td>
<td>20.0</td>
<td>30.0</td>
<td>38.0</td>
<td>54.0</td>
<td>66.0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$P_{ON}$ (mW)</td>
<td>3.0</td>
<td>6.0</td>
<td>12.0</td>
<td>15.0</td>
<td>23.0</td>
<td>30.0</td>
<td>38.0</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>$P_{TOTAL}$ (mW)</td>
<td>6.0</td>
<td>18.0</td>
<td>32.0</td>
<td>45.0</td>
<td>61.0</td>
<td>84.0</td>
<td>104.0</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

*Table B.8 Transistor Switching Transition Loss - Impedance-Matching Interface.*

The data in Table B.9 is the experimental measured battery current ($I_{BATT}$), battery voltage ($V_{BATT}$) and output power ($P_{OUT}$). Refer to Section 5.4.7, Figure 5.21 for a graphical representation.
APPENDIX B - REGENERATIVE INTERFACE ANALYSIS DATA

## Appendix B

### Normalised Damping Coefficient, $F_{\text{DAMP}}$

<table>
<thead>
<tr>
<th>$E_c$ (V)</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
<th>0.4</th>
<th>0.5</th>
<th>0.6</th>
<th>0.7</th>
<th>0.8</th>
<th>0.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_{\text{BATT}}$ (A)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>0.044</td>
<td>0.079</td>
<td>0.099</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
</tr>
<tr>
<td>0.2</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
</tr>
<tr>
<td>0.3</td>
<td>0.099</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
</tr>
<tr>
<td>0.4</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
</tr>
<tr>
<td>0.5</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
</tr>
<tr>
<td>0.6</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
</tr>
<tr>
<td>0.7</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
</tr>
<tr>
<td>0.8</td>
<td>0.062</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
</tr>
<tr>
<td>0.9</td>
<td>0.025</td>
<td>0.044</td>
<td>0.079</td>
<td>0.115</td>
<td>0.119</td>
<td>0.113</td>
<td>0.091</td>
<td>0.062</td>
<td>0.025</td>
</tr>
</tbody>
</table>

### Table B.9 Output Power - Impedance-Matching Interface.

The data in Table B.10 is the data for the theoretical, simulation (including transistor 'on' resistance, $R_{\text{DS}}$) and experimental output power. The experimental output power in this table includes the transistor transition loss. Refer to Section 5.4.7, Figure 5.21 for a graphical representation.

<table>
<thead>
<tr>
<th>$E_c$ (V)</th>
<th>Theory</th>
<th>Simulation</th>
<th>Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0</td>
<td>0.297</td>
<td>0.282</td>
<td>0.278</td>
</tr>
<tr>
<td>4.0</td>
<td>1.134</td>
<td>1.092</td>
<td>1.110</td>
</tr>
<tr>
<td>6.0</td>
<td>2.398</td>
<td>2.397</td>
<td>2.456</td>
</tr>
</tbody>
</table>

### Table B.10 Output Power - Theoretical, Simulation and Experimental Comparison.
The data in Table B.11 is the measured drive current data ($I_D$), for the impedance-matching regenerative interface system.

<table>
<thead>
<tr>
<th>Normalised Damping Coefficient, $F_{DAMP}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
</tr>
<tr>
<td>$E_C=2.0 (V)$</td>
</tr>
<tr>
<td>$E_C=4.0 (V)$</td>
</tr>
<tr>
<td>$E_C=6.0 (V)$</td>
</tr>
</tbody>
</table>

**Table B.11 Drive Current - Impedance-Matching Interface.**

The data in Table B.12 is the measured damping coefficient ratio ($C_{EFF}$). This data was estimated from the damping relationship given in Equation 5.30, the current-voltage ratio, $R_E = 1.39 \text{ (V/A)}$, and the drive current data given in Table B.11. Refer to Section 5.4.8, Figure 5.22 for a graphical representation.

<table>
<thead>
<tr>
<th>Normalised Damping Coefficient, $F_{DAMP}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
</tr>
<tr>
<td>$E_C=2.0 (V)$</td>
</tr>
<tr>
<td>$E_C=4.0 (V)$</td>
</tr>
<tr>
<td>$E_C=6.0 (V)$</td>
</tr>
</tbody>
</table>

**Table B.12 Normalised Damping Coefficient - Impedance-Matching Interface.**

The data in Table B.13 is the measured power efficiency data. This data was estimated from the power relationships given in Equations 5.21, 5.35 and 5.36, and the drive current data given in Table B.11. Refer to Section 5.4.9, Figure 5.24 for a graphical representation.

<table>
<thead>
<tr>
<th>Normalised Damping Coefficient, $F_{DAMP}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
</tr>
<tr>
<td>$E_C=2.0 (V)$</td>
</tr>
<tr>
<td>$E_C=4.0 (V)$</td>
</tr>
<tr>
<td>$E_C=6.0 (V)$</td>
</tr>
</tbody>
</table>

**Table B.13 Power Efficiency - Impedance-Matching Interface.**
B.5 Dynamic Experimental Data

The data in this section are for the experimental dynamic investigations documented in Chapter 5, Sections 5.5. The data in Table B.14 was the measured output power ($P_{OUT}$), for the dynamic analysis of the regenerative interface. Refer to Section 5.5.3, Figure 5.26 for a graphical representation.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Output Power, $P_{OUT}$ (W)</th>
<th>$F_{DAMP}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.2</td>
<td>0.4</td>
</tr>
<tr>
<td>1.031</td>
<td>0.067</td>
<td>0.070</td>
</tr>
<tr>
<td>1.078</td>
<td>0.076</td>
<td>0.084</td>
</tr>
<tr>
<td>1.172</td>
<td>0.113</td>
<td>0.099</td>
</tr>
<tr>
<td>1.266</td>
<td>0.468</td>
<td>0.311</td>
</tr>
<tr>
<td>1.312</td>
<td>0.684</td>
<td>0.429</td>
</tr>
<tr>
<td>1.359</td>
<td>1.000</td>
<td>0.540</td>
</tr>
<tr>
<td>1.406</td>
<td>0.794</td>
<td>0.583</td>
</tr>
<tr>
<td>1.453</td>
<td>0.618</td>
<td>0.574</td>
</tr>
<tr>
<td>1.593</td>
<td>0.477</td>
<td>0.484</td>
</tr>
<tr>
<td>1.734</td>
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<td>0.307</td>
<td>0.345</td>
</tr>
<tr>
<td>2.156</td>
<td>0.291</td>
<td>0.322</td>
</tr>
<tr>
<td>2.344</td>
<td>0.320</td>
<td>0.400</td>
</tr>
</tbody>
</table>

Table B.14 Output Power - Impedance-Matching Interface.
B.6 Regenerative Interface Comparison Data

The data in this section are for the experimental steady-state investigations documented in Chapter 5, Sections 5.6. The data in Table B.15 was the measured output power \( P_{\text{OUT}} \), for the passive regenerative system. Refer to Section 5.6.2, Figure 5.31 for a graphical representation and comparison with the impedance-matching regenerative system output power measurements.

\[
\begin{array}{|c|c|c|c|c|c|c|c|c|c|}
\hline
E_{C} (V) & 0 & \ldots & 6.0 & 7.0 & 7.5 & 8.0 & 8.5 & 9.0 & 9.5 & 10.0 \\
\hline
V_{\text{BATT}} (V) & 6.20 & 6.20 & 6.20 & 6.21 & 6.22 & 6.27 & 6.35 & 6.45 & 6.56 \\
I_{\text{BATT}} (A) & 0 & 0 & 0 & 0.04 & 0.16 & 0.50 & 0.82 & 1.14 & 1.47 \\
P_{\text{OUT}} (W) & 0 & 0 & 0 & 0.25 & 0.99 & 3.13 & 5.21 & 7.35 & 9.64 \\
\hline
\end{array}
\]

Table B.15 Output Power - Passive Rectifier Interface.

The data in Table B.16 is the measured output power \( P_{\text{OUT}} \), for the impedance-matching regenerative interface. Refer to Section 5.6.2, Figure 5.31 for a graphical representation and comparison with the passive interface output power measurements.

\[
\begin{array}{|c|c|c|c|c|c|}
\hline
F_{\text{DAMP}} & \multicolumn{5}{c}{\text{Output Power, } P_{\text{OUT}} (W)} \\
& \text{Open-Circuit Machine Voltage, } E_{C} (V) & 0 & 2 & 4 & 6 \\
\hline
0.3 & 0 & 0.615 & 2.535 & 5.581 & 9.782 \\
0.5 & 0 & 0.739 & 3.051 & 6.934 & - \\
0.7 & 0 & 0.565 & 2.387 & 5.503 & - \\
\hline
\end{array}
\]

Table B.16 Output Power - Impedance-Matching Interface.

The data in Table B.17 is the measured drive current \( I_{0} \), and damping coefficient ratio \( C_{\text{EFF}} \), for the passive regeneration circuit. Refer to Section 5.6.2, Figure 5.32 for a graphical representation and comparison with the impedance-matching damping coefficient ratio measurements.
Table B.17 Normalised Damping Coefficient - Passive Rectifier Interface.

The data in Table B.18 is the measured drive current \((I_d)\), for the impedance-matching regenerative system.

Table B.18 Drive Current - Impedance-Matching Interface.

The data in Table B.19 is the measured normalised damping coefficient \((C_{EFF})\), of the impedance-matching regenerative interface. Refer to Section 5.6.2, Figure 5.32 for a graphical representation and comparison with the passive interface damping coefficient ratio measurements.

Table B.19 Normalised Damping Coefficient - Impedance-Matching Interface.
B.7 PSpice Simulation Program

* Regeneration Circuit Simulation

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>VEC</td>
<td>1 0 4volt</td>
<td>Open-Circuit Machine Voltage</td>
</tr>
<tr>
<td>RINT</td>
<td>1 2 1.06ohm</td>
<td>Internal Machine Resistance</td>
</tr>
<tr>
<td>L1</td>
<td>2 3 2.3mH</td>
<td>Internal Machine Inductance</td>
</tr>
<tr>
<td>VSW</td>
<td>5 0 PULSE(-1V 2V 0.1us 0.1us 0.1us 0.0716ms 0.1ms)</td>
<td>Switch Signal</td>
</tr>
<tr>
<td>SWITCH1</td>
<td>7 0 5 0 SWITCH</td>
<td></td>
</tr>
<tr>
<td>RON</td>
<td>7 3 0.150ohm</td>
<td></td>
</tr>
<tr>
<td>D1</td>
<td>3 4 DMOD</td>
<td>Diode</td>
</tr>
<tr>
<td>CL</td>
<td>4 0 1000uF</td>
<td>Output Capacitor</td>
</tr>
<tr>
<td>RBATT</td>
<td>4 6 0.25ohm</td>
<td>Storage Battery Resistance 0.25</td>
</tr>
<tr>
<td>VBATT</td>
<td>6 0 6.2volts</td>
<td>Storage Battery Potential 6.2</td>
</tr>
</tbody>
</table>

.model DMOD D
.model SWITCH VSWITCH RON=1mohm
.model DSWITCH VSWITCH RON=0.001ohm ROFF=9E9ohm VON=0.1volts VOFF=0volts

.OPTIONS RELTOL=0.001
.PROBE
.TRAN/OP 1ms 20ms ;
.END

Table B.20 PSpice Simulation Program - Impedance-Matching Interface.
APPENDIX

C

EXPERIMENTAL ROAD SURFACE DATA FOR DAMPER DISSIPATION COMPARISON

This appendix documents the experimental road-surface analysis given by Browne and Hamburg [10]. The data is used for the comparison between experimental and theoretical vehicle suspension, energy dissipation analysis in Chapter 6, Section 6.2.2. Refer to Figure 6.3 for a graphical representation of the data given in this appendix.

The data in Table C.1 gives the experimental road-surface description given by Browne and Hamburg [10] (Table III), and the road-surface classification used in this research program, as estimated from the description given by Browne and Hamburg.
APPENDIX C - EXPERIMENTAL ROAD SURFACE DATA FOR DAMPER DISSIPATION COMPARISON

<table>
<thead>
<tr>
<th>Route</th>
<th>Road</th>
<th>Description</th>
<th>Classification*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tech Center N-S Test Track</td>
<td>North Heading Straightaway</td>
<td>Smooth level asphalt with a few mild expansion joints</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>South Heading Straightaway</td>
<td>Smooth level concrete in excellent condition</td>
<td>Very Good</td>
</tr>
<tr>
<td>Interstate, Suburban Main Artery Route</td>
<td>I-696</td>
<td>New interstate in excellent condition.</td>
<td>Very Good</td>
</tr>
<tr>
<td></td>
<td>I-75</td>
<td>Deteriorating interstate potholes, poorly repaired expansion joints</td>
<td>Poor</td>
</tr>
<tr>
<td></td>
<td>16 Mile Road</td>
<td>2-Lane suburban artery with major waviness and isolated rough sections</td>
<td>Poor</td>
</tr>
<tr>
<td></td>
<td>Mound Road</td>
<td>4-Lane parkway with isolated rough sections</td>
<td>Average</td>
</tr>
<tr>
<td></td>
<td>Gravel Road</td>
<td>Graded dirt-gravel roads with isolated potholes and short washboard sections</td>
<td>Poor</td>
</tr>
<tr>
<td></td>
<td>Rural Main Artery Route</td>
<td>Dequindre Road South of 18 Mile Road</td>
<td>Poor</td>
</tr>
</tbody>
</table>

Table C.1 Road Surface Classification from Browne and Hamburg [10], Table III.

The data in Table C.2 gives the experimental energy dissipation by Browne and Hamburg [10] (Figure 6). The road-surface classification used in this research program was estimated from the road-surface description given by Browne and Hamburg. All data for Table C.2 was taken for the J 2000 Pontiac at the Tech Center test track.

<table>
<thead>
<tr>
<th>Road Surface</th>
<th>Classification*</th>
<th>32</th>
<th>48</th>
<th>64</th>
<th>80</th>
<th>96</th>
</tr>
</thead>
<tbody>
<tr>
<td>Concrete</td>
<td>Very Good</td>
<td>1.0</td>
<td>2.0</td>
<td>2.9</td>
<td>3.5</td>
<td>4.6</td>
</tr>
<tr>
<td>Asphalt</td>
<td>Good</td>
<td>1.9</td>
<td>3.0</td>
<td>4.8</td>
<td>6.7</td>
<td>8.8</td>
</tr>
</tbody>
</table>

Table C.2 Road Surface Data from Browne and Hamburg [10], Figure 6.

The data in Table C.3 gives the experimental energy dissipation by Browne and Hamburg [10] (Figure 7). The road-surface classification used in this research program was estimated from the road-surface description given by Browne and Hamburg. All data in Table C.3 was taken for the J 2000 Pontiac at the main rural artery (Dequindre Road).

* Classification as defined in this investigation from the description given by Browne and Hamburg [10].
The data in Table C.4 gives the experimental energy dissipation by Browne and Hamburg [10] (Table VII). The road-surface classification used in this research program was estimated from the road-surface description given by Browne and Hamburg. The data shown in Table C.4 was obtained only for the J 2000 Pontiac with a constant tire pressure of 221 (kPa).

The data in Table C.5 gives the experimental energy dissipation by Browne and Hamburg [10] (Table IX). The road-surface classification used in this research program was estimated from the road-surface description given by Browne and Hamburg. The data shown in Table C.5 was obtained only for the J 2000 Pontiac.

*Classification as defined in this investigation from the description given by Browne and Hamburg [10].
<table>
<thead>
<tr>
<th>Test Roadway</th>
<th>Velocity (km/h)</th>
<th>Loss Rate (W)</th>
<th>Road Surface Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>I-696 Mound to I-75</td>
<td>89</td>
<td>6.7</td>
<td>Very Good</td>
</tr>
<tr>
<td>I-75 10 to 16 Mile</td>
<td>89</td>
<td>10.8</td>
<td>Average</td>
</tr>
<tr>
<td>16 Mile I-75 to Mound</td>
<td>72</td>
<td>17.4</td>
<td>Average</td>
</tr>
<tr>
<td>Mound 16 to 14</td>
<td>80</td>
<td>12.8</td>
<td>Average</td>
</tr>
<tr>
<td>Gravel S to N</td>
<td>48</td>
<td>16.8</td>
<td>Poor</td>
</tr>
<tr>
<td>Gravel N to S</td>
<td>48</td>
<td>56.1</td>
<td>Poor</td>
</tr>
</tbody>
</table>

*Classification as defined in this investigation from the description given by Browne and Hamburg [10].

Table C.5 Road Surface Data from Browne and Hamburg [10], Table IX.
ACCELERATION RESPONSE DATA FOR THE ROTATING DAMPER ANALYSIS

This appendix documents data used for the theoretical analysis in Chapter 6, Section 6.5. This analysis documented an investigation of passenger response to vibration, as a method to determine the limits of rotating mass in vehicle suspension systems.

The data in Table D.1 gives the "reduced comfort boundary" for vibrational acceleration in the longitudinal direction as specified according to ISO 2631 [72]. Multiply the acceleration values in Table D.1 by 3.15 (10 dB higher) to obtain "fatigue-decreased proficiency boundary", and multiply by 6.30 (16 dB higher) to obtain "exposure limits". Refer to Chapter 6, Figure 6.11 for a graphical representation.

<table>
<thead>
<tr>
<th>Frequency, $f_c$ (Hz)</th>
<th>RMS Acceleration, $a_z$ (m/s²)</th>
<th>Exposure Time (Hours)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.0</td>
<td>8.0</td>
</tr>
<tr>
<td>1.0</td>
<td>$7.49 \times 10^{-1}$</td>
<td>$2.00 \times 10^{-1}$</td>
</tr>
<tr>
<td>4.0</td>
<td>$3.75 \times 10^{-1}$</td>
<td>$1.00 \times 10^{-1}$</td>
</tr>
<tr>
<td>8.0</td>
<td>$3.75 \times 10^{-1}$</td>
<td>$1.00 \times 10^{-1}$</td>
</tr>
<tr>
<td>80.0</td>
<td>$3.75 \times 10^{-6}$</td>
<td>$1.00 \times 10^{-6}$</td>
</tr>
</tbody>
</table>

Table D.1 Numerical Values of ISO 2631 "Reduced Comfort Boundary" [72].
The data in Table D.2 is the theoretical passenger, RMS acceleration, for a two degree-of-freedom vehicle model, with vehicle velocity \( (V = 28 \text{ m/s}) \), and road-surface roughness constant \( (A = 0.035\times10^{-4} \text{ m.cycle}) \). Refer to Section 6.5.1, Figure 6.14 for a graphical representation.

<table>
<thead>
<tr>
<th>Frequency, ( f_c ) (Hz)</th>
<th>RMS Acceleration, ( a_z ) (m/s(^2))</th>
<th>Rotating Mass, ( m_r ) (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>2.99\times10^{-2}</td>
<td>2.91\times10^{-2}</td>
</tr>
<tr>
<td>1.2</td>
<td>3.57\times10^{-2}</td>
<td>3.47\times10^{-2}</td>
</tr>
<tr>
<td>1.5</td>
<td>3.24\times10^{-2}</td>
<td>3.13\times10^{-2}</td>
</tr>
<tr>
<td>2.0</td>
<td>3.39\times10^{-2}</td>
<td>3.26\times10^{-2}</td>
</tr>
<tr>
<td>3.0</td>
<td>3.99\times10^{-2}</td>
<td>3.88\times10^{-2}</td>
</tr>
<tr>
<td>4.0</td>
<td>4.43\times10^{-2}</td>
<td>4.44\times10^{-2}</td>
</tr>
<tr>
<td>5.0</td>
<td>4.99\times10^{-2}</td>
<td>5.20\times10^{-2}</td>
</tr>
<tr>
<td>6.0</td>
<td>5.68\times10^{-2}</td>
<td>6.21\times10^{-2}</td>
</tr>
<tr>
<td>7.0</td>
<td>6.26\times10^{-2}</td>
<td>7.15\times10^{-2}</td>
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<tr>
<td>8.0</td>
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<td>7.46\times10^{-2}</td>
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<td>9.0</td>
<td>6.29\times10^{-2}</td>
<td>7.08\times10^{-2}</td>
</tr>
<tr>
<td>10.0</td>
<td>5.26\times10^{-2}</td>
<td>5.12\times10^{-2}</td>
</tr>
<tr>
<td>12.0</td>
<td>3.08\times10^{-2}</td>
<td>2.67\times10^{-2}</td>
</tr>
<tr>
<td>15.0</td>
<td>1.27\times10^{-2}</td>
<td>1.20\times10^{-2}</td>
</tr>
</tbody>
</table>

Table D.2 RMS Acceleration Response - Variable Rotating Mass.

The data in Table D.3 is the theoretical passenger, RMS acceleration, for a two degree-of-freedom vehicle model, with vehicle velocity \( (V = 28 \text{ m/s}) \), and vehicle damper, rotating mass of \( (m_r = 10.0 \text{ kg}) \). Refer to Section 6.5.1, Figure 6.15 for a graphical representation.
## Table D.3 RMS Acceleration Response - Variable Road-Surface Condition

<table>
<thead>
<tr>
<th>Frequency, $f_c$ (Hz)</th>
<th>Roughness Constant, $A \times 10^{-4}$ (m.cyc1e)</th>
<th>RMS Acceleration, $a_r (m/s^2)$</th>
<th>0.001</th>
<th>0.002</th>
<th>0.0095</th>
<th>0.035</th>
<th>0.15</th>
<th>0.55</th>
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<tbody>
<tr>
<td>1.2</td>
<td></td>
<td></td>
<td>4.93x10^{-3}</td>
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<td>1.15x10^{-1}</td>
</tr>
<tr>
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<td></td>
<td>5.86x10^{-3}</td>
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<td></td>
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<td>3.13x10^{-2}</td>
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<td>1.24x10^{-1}</td>
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<td></td>
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<td></td>
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<td></td>
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<td>6.21x10^{-2}</td>
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<td>2.46x10^{-1}</td>
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<td></td>
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<td>7.15x10^{-2}</td>
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<td></td>
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<td>7.46x10^{-2}</td>
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<td>3.69x10^{-2}</td>
<td>7.08x10^{-2}</td>
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<td>8.65x10^{-3}</td>
<td>1.22x10^{-2}</td>
<td>2.67x10^{-2}</td>
<td>5.12x10^{-2}</td>
<td>1.06x10^{-1}</td>
<td>2.03x10^{-1}</td>
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<td>1.06x10^{-1}</td>
</tr>
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<td></td>
<td>2.03x10^{-3}</td>
<td>2.87x10^{-3}</td>
<td>6.26x10^{-3}</td>
<td>1.20x10^{-2}</td>
<td>2.49x10^{-2}</td>
<td>4.77x10^{-2}</td>
</tr>
</tbody>
</table>
This appendix documents the detailed findings of the theoretical cost-analysis investigation undertaken to estimate the performance of regenerative damping in vehicle suspension systems. The results are for the theoretical investigation documented in Chapter 7.

Table E.1 presents the cost-analysis results for a typical regenerative vehicle damping application. The data in Table E.1 was classified as either 'specified', 'constant' or 'evaluated'. The data referred to as 'specified' was determined in the original analysis and included information such as the required damping coefficient \( (c_v) \). The data referred to as 'constant' included all data that was unchanging such as the conductivity of copper \( (\sigma) \). The data referred to as 'evaluated' was determined from the 'specified' and 'constant' data through the relationships documented throughout this thesis.
The following parameters were specified for this analysis:

- Effective Damping Coefficient ($c_v = 1,400 \text{ Ns/m}$),
- Mass Conducting Material ($M_{\text{COND}} = 0.30 \text{ kg}$),
- Normalised Damping Coefficient ($F_{\text{DAMP}} = 0.5$),
- Rare-Earth Material Production Energy = 200,000 (kJ/kg).

### Parameter Data Type Symbol Value Unit

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Data Type</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effective Damping Coefficient</td>
<td>Specified</td>
<td>$c_v$</td>
<td>1,400</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Mass Conducting Material</td>
<td>Specified</td>
<td>$M_{\text{COND}}$</td>
<td>0.30</td>
<td>kg</td>
</tr>
<tr>
<td>Copper Density</td>
<td>Constant</td>
<td>$\rho_{\text{COND}}$</td>
<td>$8.96 \times 10^3$</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td>Copper Volume</td>
<td>Evaluated</td>
<td>$\kappa$</td>
<td>$3.35 \times 10^{-5}$</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Permanent Magnetic Field Strength</td>
<td>Specified</td>
<td>$B_0$</td>
<td>0.40</td>
<td>T</td>
</tr>
<tr>
<td>Copper Conductivity</td>
<td>Constant</td>
<td>$\sigma$</td>
<td>$1.72 \times 10^{-8}$</td>
<td>$\Omega \text{ m}$</td>
</tr>
<tr>
<td>Motor Damping Coefficient</td>
<td>Evaluated</td>
<td>$F_{\text{DAMP}}$</td>
<td>0.50</td>
<td></td>
</tr>
<tr>
<td>Gear Ratio</td>
<td>Evaluated</td>
<td>$\alpha$</td>
<td>3.00</td>
<td></td>
</tr>
<tr>
<td>Rotating Mass</td>
<td>Evaluated</td>
<td>$mr$</td>
<td>10.79</td>
<td>kg</td>
</tr>
</tbody>
</table>

**a) Regenerative Damper Specifications**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Data Type</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Age Vehicle Fleet (1995)</td>
<td>Specified</td>
<td></td>
<td>10.60</td>
<td>years</td>
</tr>
<tr>
<td>Average Kilometres Travelled Per Year (1995)</td>
<td>Specified</td>
<td></td>
<td>14,300</td>
<td>km</td>
</tr>
<tr>
<td>Conversion Efficiency</td>
<td>Evaluated</td>
<td>$E_{\text{REG}}$</td>
<td>0.50</td>
<td></td>
</tr>
<tr>
<td>Wh/kJ</td>
<td>Constant</td>
<td></td>
<td>3.60</td>
<td></td>
</tr>
<tr>
<td>Damper Power Dissipation - Urban</td>
<td>Specified</td>
<td></td>
<td>15.00</td>
<td>W</td>
</tr>
<tr>
<td>Proportion Urban Driving</td>
<td>Specified</td>
<td></td>
<td>0.55</td>
<td></td>
</tr>
<tr>
<td>Kilometres Urban Driving Per Year</td>
<td>Evaluated</td>
<td></td>
<td>7,865</td>
<td>km</td>
</tr>
<tr>
<td>Average Speed Urban</td>
<td>Specified</td>
<td></td>
<td>60.00</td>
<td>km/h</td>
</tr>
<tr>
<td>Average Driving Time Urban Per Year</td>
<td>Evaluated</td>
<td></td>
<td>131.08</td>
<td>h</td>
</tr>
<tr>
<td>Regenerated Energy Per Year Urban Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>983</td>
<td>Wh</td>
</tr>
<tr>
<td>Total Regenerated Energy Urban</td>
<td>Evaluated</td>
<td></td>
<td>3,539</td>
<td>kJ</td>
</tr>
<tr>
<td>Damper Power Dissipation - Non-Urban</td>
<td>Specified</td>
<td></td>
<td>40.00</td>
<td>W</td>
</tr>
<tr>
<td>Proportion Non-Urban Driving</td>
<td>Evaluated</td>
<td></td>
<td>0.45</td>
<td></td>
</tr>
<tr>
<td>Kilometres Non-Urban Driving Per Year</td>
<td>Evaluated</td>
<td></td>
<td>6,435</td>
<td>km</td>
</tr>
<tr>
<td>Average Speed Non-Urban</td>
<td>Specified</td>
<td></td>
<td>100.00</td>
<td>km/h</td>
</tr>
<tr>
<td>Average Driving Time Non-Urban Per Year</td>
<td>Evaluated</td>
<td></td>
<td>64.35</td>
<td>h</td>
</tr>
<tr>
<td>Regenerated Energy Per Year Non-Urban Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>1,287</td>
<td>Wh</td>
</tr>
<tr>
<td>Total Regenerated Energy Non-Urban</td>
<td>Evaluated</td>
<td></td>
<td>4,633</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Regenerated Energy Per Year</td>
<td>Evaluated</td>
<td>$B_T$</td>
<td>8,172</td>
<td>kJ</td>
</tr>
</tbody>
</table>
### APPENDIX E - REGENERATIVE PERFORMANCE ANALYSIS DATA

#### Parameter Data Type Symbol Value Unit

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Data Type</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper Production Energy Per Kilogram</td>
<td>Constant</td>
<td></td>
<td>45,000</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>Copper Mass Per Damper, Specified</td>
<td>Specified</td>
<td>$M_{\text{COND}}$</td>
<td>0.30</td>
<td>kg</td>
</tr>
<tr>
<td>Total Copper Production Energy</td>
<td>Evaluated</td>
<td></td>
<td>13,500</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Copper Production Energy per Year</td>
<td>Evaluated</td>
<td></td>
<td>1,274</td>
<td>kJ</td>
</tr>
<tr>
<td>Iron Volume Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>$3.35 \times 10^{-5}$</td>
<td>m³</td>
</tr>
<tr>
<td>Iron Density</td>
<td>Constant</td>
<td></td>
<td>$7.87 \times 10^3$</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Iron Mass Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>0.26</td>
<td>kg</td>
</tr>
<tr>
<td>Iron Production Energy Per Kilogram</td>
<td>Constant</td>
<td></td>
<td>24,000</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>Total Iron Production Energy</td>
<td>Evaluated</td>
<td></td>
<td>6,324</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Iron Production Energy per Year</td>
<td>Evaluated</td>
<td></td>
<td>597</td>
<td>kJ</td>
</tr>
<tr>
<td>Permanent Magnet Volume Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>$3.35 \times 10^{-5}$</td>
<td>m³</td>
</tr>
<tr>
<td>Permanent Magnet Density</td>
<td>Constant</td>
<td></td>
<td>$7.23 \times 10^3$</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Permanent Magnet Mass Per Damper</td>
<td>Evaluated</td>
<td></td>
<td>0.24</td>
<td>kg</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy Per Kilogram</td>
<td>Specified</td>
<td></td>
<td>200,000</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>Total Permanent Magnet Production Energy</td>
<td>Evaluated</td>
<td></td>
<td>48,442</td>
<td>kJ</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy Per Year</td>
<td>Evaluated</td>
<td></td>
<td>4,570</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Material Production Energy Per Year</td>
<td>Evaluated</td>
<td>$C_p$</td>
<td>6,440</td>
<td>kJ</td>
</tr>
</tbody>
</table>

#### c) Energy Negative - Production Energy

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Data Type</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Added Mass Per Damper</td>
<td>Evaluated</td>
<td>$M$</td>
<td>0.81</td>
<td>kg</td>
</tr>
<tr>
<td>Kilometres Urban Driving</td>
<td>Evaluated</td>
<td></td>
<td>7,865</td>
<td>km</td>
</tr>
<tr>
<td>Kilometres between Stop-Start Urban</td>
<td>Constant</td>
<td></td>
<td>0.80</td>
<td>km</td>
</tr>
<tr>
<td>No. of Stop-Starts Urban</td>
<td>Evaluated</td>
<td>$N_{\text{URB}}$</td>
<td>9,831</td>
<td>stops</td>
</tr>
<tr>
<td>Maximum Speed Urban</td>
<td>Specified</td>
<td>$V_{\text{URB}}$</td>
<td>60.0</td>
<td>km/h</td>
</tr>
<tr>
<td>Energy Urban Stop-Start - Added Mass</td>
<td>Evaluated</td>
<td></td>
<td>0.11</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Energy Stop-Start Urban</td>
<td>Evaluated</td>
<td></td>
<td>1,100</td>
<td>kJ</td>
</tr>
<tr>
<td>Kilometres Non-Urban Driving</td>
<td>Evaluated</td>
<td></td>
<td>6,435</td>
<td>km</td>
</tr>
<tr>
<td>Kilometres between Stop-Start Non-Urban</td>
<td>Constant</td>
<td></td>
<td>10.31</td>
<td>km</td>
</tr>
<tr>
<td>No. of Stop-Starts Non-Urban</td>
<td>Evaluated</td>
<td>$N_{\text{RUR}}$</td>
<td>624</td>
<td>stops</td>
</tr>
<tr>
<td>Maximum Speed Non-Urban</td>
<td>Specified</td>
<td>$V_{\text{RUR}}$</td>
<td>100.0</td>
<td>km/h</td>
</tr>
<tr>
<td>Energy Non-Urban Stop-Start - Added Mass</td>
<td>Evaluated</td>
<td></td>
<td>0.31</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Energy Stop-Start Non-Urban</td>
<td>Evaluated</td>
<td></td>
<td>194</td>
<td>kJ</td>
</tr>
<tr>
<td>Total Energy Stop-Start - Added Mass</td>
<td>Evaluated</td>
<td>$C_M$</td>
<td>1,294</td>
<td>kJ</td>
</tr>
</tbody>
</table>

#### d) Energy Negative - Added Mass
### APPENDIX E - REGENERATIVE PERFORMANCE ANALYSIS DATA

#### Parameter Summary

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Percentage of Total Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suspension Regeneration Urban</td>
<td></td>
<td>3,539</td>
<td>kJ</td>
<td>22.2%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td></td>
<td>4,633</td>
<td>kJ</td>
<td>29.1%</td>
</tr>
<tr>
<td>Total Regenerated Energy, $B_T$</td>
<td></td>
<td>8,172</td>
<td>kJ</td>
<td>51.4%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td></td>
<td>1,274</td>
<td>kJ</td>
<td>8.0%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td></td>
<td>597</td>
<td>kJ</td>
<td>3.8%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td></td>
<td>4,570</td>
<td>kJ</td>
<td>28.7%</td>
</tr>
<tr>
<td>Total Material Production Energy, $C_p$</td>
<td></td>
<td>6,440</td>
<td>kJ</td>
<td>40.5%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td></td>
<td>1,100</td>
<td>kJ</td>
<td>6.9%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td></td>
<td>194</td>
<td>kJ</td>
<td>1.2%</td>
</tr>
<tr>
<td>Total Added Mass Energy, $C_M$</td>
<td></td>
<td>1,294</td>
<td>kJ</td>
<td>8.1%</td>
</tr>
<tr>
<td>Total Energy Cost, $C_T (= C_p + C_M)$</td>
<td></td>
<td>7,734</td>
<td>kJ</td>
<td>48.6%</td>
</tr>
<tr>
<td>Total Energy</td>
<td>$B_T + C_T$</td>
<td>15,907</td>
<td>kJ</td>
<td>100.0%</td>
</tr>
<tr>
<td>Total Energy Cost, $Cost_T (=B_T - C_T)$</td>
<td></td>
<td>438</td>
<td>kJ</td>
<td></td>
</tr>
</tbody>
</table>

### e) Energy Cost Summary

Table E.1 Cost-Analysis For A Typical Vehicle Application.

The data in Table E.2 gives the relationship between the regenerative damping performance and the material production cost of the permanent magnet material, (NdFeB). The data is referred to in Section 7.3.2. The following parameters were specified for this analysis: All parameters besides the magnet production energy are the same as presented in Table E.1.
### Parameter

<table>
<thead>
<tr>
<th>Percentage of Total Energy ($B_t + C_t$)</th>
<th>Material Production Energy ((^{000} \text{kJ/kg}))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Suspension Regeneration Urban</td>
<td>31.2%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>40.9%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ($B_t$)</td>
<td>72.1%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>11.2%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>5.3%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>0.0%</td>
</tr>
<tr>
<td>Total Material Production Energy, ($C_p$)</td>
<td>16.5%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>9.7%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>1.7%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ($C_M$)</td>
<td>11.4%</td>
</tr>
<tr>
<td>Total Energy Cost, $Cost_t$ (kJ)</td>
<td>5,008</td>
</tr>
</tbody>
</table>

**Table E.2** Regenerative Damping Performance - Permanent Magnet Production Energy.

The data in Table E.3 gives the relationship between the regenerative damping performance and the conducting mass ($M_{COND}$). The data is referred to in Section 7.3.3. All parameters besides the mass of conducting material are the same as presented in Table E.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Percentage of Total Energy ($B_t + C_t$)</th>
<th>Mass of Conducting Material, $M_{COND}$ (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Suspension Regeneration Urban</td>
<td>43.3%</td>
<td>32.9%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>56.7%</td>
<td>43.1%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ($B_t$)</td>
<td>100.0%</td>
<td>76.0%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>0.0%</td>
<td>3.9%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>0.0%</td>
<td>1.8%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>0.0%</td>
<td>14.2%</td>
</tr>
<tr>
<td>Total Material Production Energy, ($C_p$)</td>
<td>0.0%</td>
<td>20.0%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>0.0%</td>
<td>3.4%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>0.0%</td>
<td>0.6%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ($C_M$)</td>
<td>0.0%</td>
<td>4.0%</td>
</tr>
<tr>
<td>Gear Ratio ($\alpha$)</td>
<td>$\infty$</td>
<td>5.19</td>
</tr>
<tr>
<td>Total Energy Cost, $Cost_t$ (kJ)</td>
<td>8,172</td>
<td>5,594</td>
</tr>
</tbody>
</table>

**Table E.3** Regenerative Damping Performance - Conducting Mass.
The data in Table E.4 gives the relationship between the regenerative damping performance and the normalised damping ratio ($F_{\text{DAMP}}$). The data is referred to in Section 7.3.4. All parameters besides the normalised damping ratio are the same as presented in Table E.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Percentage of Total Energy ($B_T+C_T$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Normalised Damping Ratio, $F_{\text{DAMP}}$</td>
</tr>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Suspension Regeneration Urban</td>
<td>29.4%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>38.5%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ($B_T$)</td>
<td>67.9%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>5.3%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>2.5%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>19.0%</td>
</tr>
<tr>
<td>Total Material Production Energy, ($C_p$)</td>
<td>26.7%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>4.6%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>0.8%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ($C_M$)</td>
<td>5.4%</td>
</tr>
<tr>
<td>Rotating Mass, $mr$ (kg)</td>
<td>$\infty$</td>
</tr>
<tr>
<td>Total Energy Cost, $Cost_T$ (kJ)</td>
<td>8,611</td>
</tr>
</tbody>
</table>

*Table E.4 Regenerative Damping Performance - Normalised Damping Ratio.*

The data in Table E.5 gives the relationship between the regenerative damping performance and the vehicle kilometres travelled per year. The data is referred to in Section 7.3.5. All parameters besides the vehicle kilometres travelled per year are the same as presented in Table E.1.
## Table E.5 Regenerative Damping Performance - Vehicle Kilometres Travelled Per Year

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Percentage of Total Energy ((B_T+C_T))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Vehicle Kilometres Travelled Per Year ('000)</td>
</tr>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Suspension Regeneration Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ((B_T))</td>
<td>0.0%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>19.8%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>9.3%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>71.0%</td>
</tr>
<tr>
<td>Total Material Production Energy, ((C_p))</td>
<td>100.0%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ((C_M))</td>
<td>0.0%</td>
</tr>
<tr>
<td>Total Energy Cost, (Cost_T) (kJ)</td>
<td>-6,440</td>
</tr>
</tbody>
</table>

The data in Table E.6 gives the relationship between the regenerative damping performance and the proportion of vehicle kilometres travelled on urban and non-urban roads. The data is referred to in Section 7.3.5. All parameters besides the proportion of kilometres travelled on urban and non-urban roads are the same as presented in Table E.1.

## Table E.6 Regenerative Damping Performance - Proportion of Urban and Non-Urban Driving

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Percentage of Total Energy ((B_T+C_T))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Proportion of Urban Driving</td>
</tr>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Suspension Regeneration Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Suspension Regeneration Non-Urban</td>
<td>60.0%</td>
</tr>
<tr>
<td>Total Regenerated Energy, ((B_T))</td>
<td>60.0%</td>
</tr>
<tr>
<td>Copper Production Energy</td>
<td>7.4%</td>
</tr>
<tr>
<td>Iron Production Energy</td>
<td>3.5%</td>
</tr>
<tr>
<td>Permanent Magnet Production Energy</td>
<td>26.6%</td>
</tr>
<tr>
<td>Total Material Production Energy, ((C_p))</td>
<td>37.5%</td>
</tr>
<tr>
<td>Added Mass Urban</td>
<td>0.0%</td>
</tr>
<tr>
<td>Added Mass Non-Urban</td>
<td>2.5%</td>
</tr>
<tr>
<td>Total Added Mass Energy, ((C_M))</td>
<td>2.5%</td>
</tr>
<tr>
<td>Total Energy Cost, (Cost_T) (kJ)</td>
<td>3,425</td>
</tr>
</tbody>
</table>