

AN INVESTIGATION INTO THE DYNAMICS
OF VEHICLES WITH HYDRAULICALLY
INTERCONNECTED SUSPENSIONS

by

Wade Alister Smith

Submitted in fulfilment of the requirements for the degree of

Doctor of Philosophy

Faculty of Engineering and Information Technology
University of Technology, Sydney

June, 2009

CERTIFICATE OF AUTHORSHIP/ORIGINALITY

I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

Signed

Production Note:
Signature removed prior to publication.

Wade Smith

Acknowledgements

I would like to take the opportunity to thank a number of people for their assistance, encouragement and support throughout my candidature.

First and foremost, I extend my gratitude to my supervisor, Professor Nong Zhang. His guidance was invaluable. My co-supervisor, Jeku Jeyakumaran, is also deserving of many thanks. He was always approachable and helpful.

Chris Chapman's expertise in the lab was a tremendous asset to the project, and I greatly appreciate his help. Building the half-car test rig required a lot of work and a couple of iterations, and the UTS workshop staff – Scott, Richard, Darren, Bill and Harold – are to be commended for their efforts. I would also like to thank Greg Koutchavlis for his work in getting the rig *going*. And Matt Rozyn's help with the experimentation and data processing is greatly appreciated. In programming and simulations, I sought help at times from Miao Wang, Zhan Wang and Wenlong Hu. I thank them for that. A special mention should also go to everyone at Kinetic Pty Ltd, and in particular, Ray Munday, Chris Revill and Stuart Price. The financial support of this work by the Australian Research Council (ARC LP0562440) and the University of Technology, Sydney, is gratefully acknowledged.

On the social side, it is hard to know where to begin. I'll always treasure the times in the early days, sitting next to Anthony in the office. More recently, Janitha, Fook, Debbie, Greg, Pete and Thuyen provided great friendship at UTS, while Baden, Sam and Michelle were assiduous in their provision of remote support via email. And I always looked forward to the weekend adventures with Drew, Paul and Jamie.

I would like to thank my parents, Rosemary and Graham, for their love and support over the years. My mother, in particular, deserves special thanks for all her help in the last few months.

Lastly, I want Ulli to know how much I value her company and her support, especially during the writing of this thesis. I intend to return the favour.

Wade Smith

Sydney, December 2008

Table of Contents

Acknowledgements.....	iii
List of Figures.....	ix
List of Tables	xiv
Abstract.....	xvi
Chapter 1 Introduction.....	1
1.1 Overview of the research	1
1.2 Research objectives and contribution to knowledge.....	2
1.3 Scope of thesis	3
1.3.1 Areas that are addressed.....	3
1.3.2 Areas that are not addressed.....	3
1.4 Outline of thesis	4
Chapter 2 Background and Literature Review.....	6
2.1 Introduction and rationale	6
2.2 Structure and role of vehicle suspension systems	6
2.3 Interconnected suspensions.....	8
2.3.1 Definition	8
2.3.2 Raison d'être	8
2.3.3 Early interconnection schemes	10
2.3.4 Recent research and applications in interconnected suspensions	13
2.4 Kinetic suspension	18
2.4.1 Hydraulically interconnected suspension (HIS)	18
2.4.2 Description of system and components.....	19
2.4.3 How the system works	21
2.4.4 Recent research and state of current knowledge	23
2.4.5 Areas as yet unaddressed.....	24
2.5 Summary of possible approaches.....	24
2.5.1 Hydraulic system modelling.....	24
2.5.2 Vehicle ride modelling	26
2.5.3 Experimentation	27
2.5.4 Suspension optimisation.....	28
2.6 Methods used in this thesis	29
2.7 Summary	30

Chapter 3 System Model Formulation and General Results	31
3.1 Introduction and rationale	31
3.2 Mechanical subsystem	32
3.2.1 Model description.....	32
3.2.2 System equations.....	33
3.3 Mechanical-fluid system boundary conditions	34
3.4 Fluid subsystem	35
3.4.1 Basic impedance modelling.....	36
3.5 Integrated system equations	37
3.6 General impedance matrix for basic wheel-pair interconnections	39
3.6.1 Anti-synchronous half-car arrangement	39
3.6.2 Anti-oppositional half-car arrangement	40
3.7 Independent control of modal parameters and modal decoupling.....	41
3.7.1 Effect of impedance in main interconnecting lines	42
3.7.2 Effect of impedance in an added side branch.....	47
3.8 Discussion.....	50
3.8.1 Limitations of the model	50
3.9 Summary	51
Chapter 4 Free Vibration Analysis.....	53
4.1 Introduction and rationale	53
4.2 Model description	54
4.2.1 Mechanical subsystem.....	54
4.2.2 Fluid subsystem.....	54
4.3 Application of the system equations	56
4.3.1 Characteristic equation.....	56
4.3.2 Fluid system equations	56
4.4 Solution of the characteristic equation	58
4.4.1 Root searching algorithm	58
4.4.2 Interpretation of results	59
4.5 Integrated system free vibration results	61
4.5.1 Equivalent conventional suspension parameters	62
4.5.2 Low frequency natural modes	63
4.5.3 High frequency natural modes	66
4.5.4 Determination of fluid mode shapes.....	66
4.6 Discussion	73
4.6.1 Alternative solution methods.....	73
4.6.2 Alternative boundary model formulation	73
4.7 Summary	76

Chapter 5 Forced Vibration Analysis	77
5.1 Introduction and rationale	77
5.2 Model description	77
5.3 Application of the system equations	78
5.4 Frequency response functions.....	78
5.5 Road surface description.....	80
5.6 Performances indices	82
5.6.1 Definitions.....	82
5.6.2 Equations.....	83
5.7 Vehicle response	84
5.8 Ride simulation results and discussion	85
5.8.1 Baseline HIS and conventional suspension vehicles	86
5.8.2 Sensitivity analysis.....	87
5.9 High frequency simulation results and discussion	91
5.9.1 Baseline HIS and conventional suspension vehicles	92
5.9.2 Sensitivity analysis.....	92
5.10 Summary	96
Chapter 6 Nonlinear Fluid Model Comparison	98
6.1 Introduction and rationale	98
6.2 Mechanical system model.....	98
6.3 Nonlinear fluid system model	99
6.3.1 Fluid component models	100
6.3.2 System model and solution scheme.....	102
6.4 Methodology	103
6.5 Results.....	104
6.5.1 Step response.....	104
6.5.2 Forced vibration	107
6.6 Discussion	107
6.6.1 Step response.....	107
6.6.2 Forced vibration	108
6.7 Summary	108
Chapter 7 Experimental Verification.....	110
7.1 Introduction and rationale	110
7.2 Description of test facility	111
7.2.1 General layout	111
7.2.2 Hydraulic system layout.....	114
7.2.3 Mechanical system parameters.....	115
7.2.4 Instrumentation and data acquisition.....	116

7.2.5	External force application	118
7.3	Testing methodology	120
7.3.1	General	120
7.3.2	Free vibration	120
7.3.3	Forced vibration	121
7.4	Application of mathematical model	122
7.4.1	Initial approach.....	122
7.4.2	Recalculation of effective valve loss coefficient.....	122
7.4.3	Inclusion of rubber bushing model.....	123
7.5	Experimental results.....	124
7.5.1	Free vibration	124
7.5.2	Forced vibration	127
7.5.3	Comparison with theory	131
7.6	Discussion	139
7.6.1	Experimental limitations	139
7.6.2	Unmodelled effects	140
7.6.3	Suggestions for further testing	142
7.7	Summary	142

Chapter 8 System Optimisation and Sensitivity Analysis144

8.1	Introduction and rationale	144
8.2	Model Description	145
8.3	Optimisation Background	146
8.3.1	Definitions and notation	146
8.3.2	Pareto optimality	146
8.3.3	Utopia point and compromise solution.....	147
8.3.4	Typical problem formulation.....	147
8.4	Problem formulation	148
8.4.1	Problem statement	148
8.4.2	Performance indices, design variables and constraints.....	148
8.4.3	Problem formulation	151
8.4.4	Methodology	151
8.5	Optimisation results	153
8.5.1	Pareto optimal set	153
8.5.2	Discussion	158
8.6	Sensitivity analysis.....	160
8.6.1	Introduction	160
8.6.2	Selection of base points and design variables	160
8.6.3	Parameter sensitivities – ride comfort	163
8.6.4	Parameter sensitivities – suspension working space	165

8.6.5	Parameter sensitivities – road holding.....	167
8.6.6	Discussion	169
8.7	Summary	171
Chapter 9	Conclusions and Recommendations.....	172
9.1	Summary	172
9.2	Contributions	174
9.3	Suggestions for future work	176
Appendix A	Half-Car Equations of Motion	179
Appendix B	Fluid System Component Models.....	182
Appendix C	Publications Resulting From This Work.....	188
References	190

List of Figures

Figure 2.1 Hawley's interconnected suspension arrangements from the 1920s.....	11
Figure 2.2 Schematic of the early Citroen 2CV suspension	11
Figure 2.3 Moulton's Hydragas suspension system	12
Figure 2.4 Zapletal's "Balanced Suspension" concept	15
Figure 2.5 Smith and Walker's fully decoupled 4-wheel interconnection scheme.....	16
Figure 2.6 Fontdecaba's HIS system	17
Figure 2.7 Kinetic H2 system	19
Figure 2.8 Cutaway view of Kinetic damper valve.....	20
Figure 2.9 H2 system nominal fluid flow distribution in idealised suspension modes: (a) bounce; (b) roll; (c) pitch; and (d) articulation	22
Figure 3.1 Schematic of a roll-plane half-car with an HIS	32
Figure 3.2 Mechanical-fluid system boundary conditions	34
Figure 3.3 Two-port representation of hydraulic component	36
Figure 3.4 Schematic of a general anti-synchronous half-car HIS arrangement	39
Figure 3.5 Schematic of a general anti-oppositional half-car HIS arrangement.....	40
Figure 3.6 Basic fluid line element	43
Figure 3.7 Schematic of half-car HIS with added side branch including fluid resistance and capacitance: anti-synchronous (left) and anti-oppositional (right) arrangements	48
Figure 4.1 Schematic of a typical half-car HIS for anti-roll applications.....	55
Figure 4.2 Flow chart of free vibration root searching algorithm.....	60
Figure 4.3 Conventional suspension model for 'equivalent' parameters	62
Figure 4.4 Three dimensional plot of $\left \det(\hat{\mathbf{A}}(s) - s\mathbf{I}) \right $ showing the four roots of the characteristic equation corresponding to the half-car's multi-body-dominated modes ..	64

Figure 4.5 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: multi-body bounce mode.....	69
Figure 4.6 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: multi-body roll mode.....	69
Figure 4.7 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: multi-body synchronous wheel hop mode.....	70
Figure 4.8 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: multi-body oppositional wheel hop mode	70
Figure 4.9 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: fluid system first synchronous mode (183 Hz)	71
Figure 4.10 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: fluid system first oppositional mode (182 Hz).....	71
Figure 4.11 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: fluid system second synchronous mode (662 Hz).....	72
Figure 4.12 Fluid system (<i>Line a</i>) mode shape corresponding to natural mode of integrated half-car system: fluid system second oppositional mode (664 Hz)	72
Figure 5.1 Frequency response functions for left ground input.....	79
Figure 5.2 ISO weighting functions for the bounce and roll mode ride evaluation.....	85
Figure 5.3 Half-car vehicle acceleration response PSDs	86
Figure 5.4 Half-car vehicle suspension deflection and tyre force response PSDs.....	86
Figure 5.5 Half-car acceleration PSDs with variation in mean system pressure (in bar); baseline value is 20 bar	88
Figure 5.6 Half-car suspension deflection and tyre force PSDs with variation in mean system pressure (in bar); baseline value is 20 bar.....	88
Figure 5.7 Half-car acceleration PSDs with variation in cylinder valve loss coefficient (in $\text{kg s}^{-1}\text{m}^{-4}$); baseline value is $5.0 \text{ kg s}^{-1}\text{m}^{-4}$	89
Figure 5.8 Half-car suspension deflection and tyre force PSDs with variation in cylinder valve loss coefficient (in $\text{kg s}^{-1}\text{m}^{-4}$); baseline value is $5.0 \text{ kg s}^{-1}\text{m}^{-4}$	89

Figure 5.9 Half-car acceleration PSDs with variation in accumulator valve loss coefficient (in $\text{kg s}^{-1}\text{m}^{-4}$); baseline value is $3.2 \text{ kg s}^{-1}\text{m}^{-4}$	90
Figure 5.10 Half-car suspension deflection and tyre force PSDs with variation in accumulator valve loss coefficient (in $\text{kg s}^{-1}\text{m}^{-4}$); baseline value is $3.2 \text{ kg s}^{-1}\text{m}^{-4}$	90
Figure 5.11 Half-car acceleration PSDs with variation in fluid viscosity (in N s m^{-2}); baseline value is 0.05 N s m^{-2}	91
Figure 5.12 Half-car suspension deflection and tyre force PSDs with variation in fluid viscosity (in N s m^{-2}); baseline value is 0.05 N s m^{-2}	91
Figure 5.13 Half-car vehicle high frequency acceleration response PSDs	92
Figure 5.14 Half-car high frequency acceleration PSDs with variation in overall hydraulic line length (in m); baseline value is 2 m	93
Figure 5.15 HIS cylinder damper valve and accumulator position definition	94
Figure 5.16 Half-car high frequency acceleration PSDs with variation in damper valve position (in m); baseline value is 0 m	95
Figure 5.17 Half-car high frequency acceleration PSDs with variation in accumulator position (in m); baseline value is 1.0 m	95
Figure 5.18 Half-car high frequency acceleration PSDs with variation in effective pipeline bulk modulus (in GPa); baseline value is 1.4 GPa.....	96
Figure 6.1 Transient responses for 2 mm step input: bounce excitation (left) and roll excitation (right).....	104
Figure 6.2 Transient responses for 20 mm step input: bounce excitation (left) and roll excitation (right).....	105
Figure 6.3 Frequency response functions for left wheel input: comparison between impedance and finite element fluid system models	107
Figure 7.1 Half-car test rig (main view).....	111
Figure 7.2 Half-car test rig (top view)	113
Figure 7.3 Schematic of half-car test rig guide rails and roller bearings (side view) ..	113
Figure 7.4 Schematic of half-car test rig low friction roller bearings (top view)	114
Figure 7.5 Schematic of hydraulic layout for anti-oppositional half-car testing	115

Figure 7.6 Schematic of hydraulic layout for anti-synchronous half-car testing	115
Figure 7.7 Schematic of half-car test rig data acquisition layout.....	117
Figure 7.8 Hydraulic actuator and servo-valve for half-car external force application....	119
Figure 7.9 Schematic of half-car test rig input force system	119
Figure 7.10 Inclusion of rubber top mount in the model (left side only shown)	124
Figure 7.11 Sprung mass free decay responses after short duration impulse for the half-car rig with hydraulic system and dampers installed (mass configuration 2).....	126
Figure 7.12 Transmissibilities from forced vibration testing: anti-oppositional arrangement; $d_p = 13.3$ mm; $\bar{p} = 15, 20, 25$ bar	128
Figure 7.13 Transmissibilities from forced vibration testing: anti-oppositional arrangement; $d_p = 16.5$ mm; $\bar{p} = 15, 20, 25$ bar	129
Figure 7.14 Transmissibilities from forced vibration testing: anti-synchronous arrangement; $d_p = 13.3$ mm; $\bar{p} = 15, 20, 25$ bar	130
Figure 7.15 Comparison between experimental and theoretical frequency responses: anti-oppositional arrangement; $d_p = 13.3$ mm; $\bar{p} = 15$ bar	133
Figure 7.16 Comparison between experimental and theoretical frequency responses: anti-oppositional arrangement; $d_p = 13.3$ mm; $\bar{p} = 20$ bar	134
Figure 7.17 Comparison between experimental and theoretical frequency responses: anti-oppositional arrangement; $d_p = 13.3$ mm; $\bar{p} = 25$ bar	135
Figure 7.18 Comparison between experimental and theoretical frequency responses: anti-synchronous arrangement; $d_p = 13.3$ mm; $\bar{p} = 15$ bar	136
Figure 7.19 Comparison between experimental and theoretical frequency responses: anti-synchronous arrangement; $d_p = 13.3$ mm; $\bar{p} = 20$ bar	137
Figure 7.20 Comparison between experimental and theoretical frequency responses: anti-synchronous arrangement; $d_p = 13.3$ mm; $\bar{p} = 25$ bar	138
Figure 7.21 Typical left wheel input displacement amplitude for forced vibration tests.....	139

Figure 7.22 Pressure loss versus flow rate for a typical shock absorber valve	141
Figure 8.1 Pareto optimal set in the objective function space: 2-D projections; $\bar{p}=15$ bar (○), $\bar{p}=20$ bar (+), $\bar{p}=25$ bar (Δ), and conventional (*)	155
Figure 8.2 Pareto optimal set in the objective function space: $\bar{p}=15$ bar, $\bar{p}=20$ bar, $\bar{p}=25$ bar, and conventional suspension.....	156
Figure 8.3 Pareto optimal set for the HIS vehicle in the design variable space: $\bar{p}=15$ bar, $\bar{p}=20$ bar, and $\bar{p}=25$ bar	156
Figure 8.4 Pareto optimal set for the HIS vehicle in the design variable space: 2-D projections; $\bar{p}=15$ bar (○), $\bar{p}=20$ bar (+), and $\bar{p}=25$ bar (Δ)	157
Figure 8.5 Pareto optimal set for the conventional vehicle in the design variable space	158
Figure 8.6 Pareto optimal set for the HIS vehicle in the objective function space, displaying base points selected for sensitivity analysis: $\bar{p}=20$ bar	161
Figure 8.7 Pareto optimal set for the HIS vehicle in the design variable space, displaying base points selected for sensitivity analysis: $\bar{p}=20$ bar	161
Figure 8.8 Ride comfort sensitivity to changes in suspension parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ)	164
Figure 8.9 Ride comfort sensitivity to changes in mechanical system parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ)	165
Figure 8.10 Rattlespace sensitivity to changes in suspension parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ)	166
Figure 8.11 Rattlespace sensitivity to changes in mechanical system parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ)	167
Figure 8.12 Road holding sensitivity to changes in suspension parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ)	168
Figure 8.13 Road holding sensitivity to changes in mechanical system parameters: single parameter perturbation from points $J_{1_{\min}}$ (○), $J_{2_{\min}}$ (□), $J_{3_{\min}}$ (\diamond), and N_{\min} (Δ).....	169

List of Tables

Table 2.1 Wheel pair interconnections.....	10
Table 4.1 Properties of the half-car mechanical subsystem.....	54
Table 4.2 Properties of the half-car fluid subsystem.....	55
Table 4.3 Equivalent parameters of the half-car system for the first four natural modes	63
Table 4.4 Four integrated system natural modes dominated by the half-car multi-body motion	63
Table 4.5 First four integrated system natural modes dominated by the fluid system...	66
Table 6.1 Comparison of results for bounce and roll modes with finite element fluid system model and 2 mm amplitude ground input.....	105
Table 6.2 Comparison of results for bounce and roll modes with finite element fluid system model and 20 mm amplitude ground input.....	105
Table 6.3 Comparison of results for wheel hop modes with finite element fluid system model and 2 mm amplitude ground input	106
Table 6.4 Comparison of results for wheel hop modes with finite element fluid system model and 20 mm amplitude ground input	106
Table 6.5 Parameter difference comparison between impedance and FEM methods for all modes – 2 mm and 20 mm amplitude ground input	106
Table 7.1 Constant parameters of the half-car experimental rig mechanical subsystem..	116
Table 7.2 Variable mass properties of the half-car experimental rig	116
Table 7.3 Technical specifications for the sensors used on the half-car rig	117
Table 7.4 Bounce and roll natural frequencies and damping ratios from free vibration testing: no hydraulic system installed	124
Table 7.5 Bounce and roll natural frequencies and damping ratios from free vibration testing: hydraulic system installed without damper valves; $\bar{p}=10$ bar	125

Table 7.6 Bounce and roll natural frequencies, damping ratios and approximate valve loss coefficients from free vibration testing: hydraulic system installed with damper valves; $\bar{p} = 10$ bar	126
Table 7.7 Valve loss coefficients and sum-of-squares error from forced vibration testing: $d_p = 13.3$ mm; $\bar{p} = 15, 20, 25$ bar	131
Table 8.1 Design variables for HIS vehicle in optimisation process	150
Table 8.2 Design variables for conventional vehicle in optimisation process	151
Table 8.3 Approximate Pareto optimal objective function ranges.....	153
Table 8.4 Points for sensitivity analysis.....	160
Table 8.5 Design variables for sensitivity analysis.....	162
Table 8.6 Results summary for sensitivity analysis: effect on objective functions of increasing the design variables.....	171
Table B.1 Assumptions underpinning adopted fluid line model	185

Abstract

This thesis examines the dynamics of a particular class of vehicle suspension, namely *hydraulically interconnected suspension* (HIS), often claimed to break the compromise between ride and handling performance. Yet such systems have, until quite recently, received little attention in the academic literature. Ideally, interconnected schemes have the capability, unique among passive suspensions, to provide stiffness and damping characteristics dependent on the all-wheel suspension mode of operation.

The modelling approach proposed here is necessarily multidisciplinary, drawing from multi-body vibration theory and fluid dynamics. A simple half-car model is used to illustrate the basic principles and to demonstrate the application of the methodology. The half-car is treated as a lumped-mass multi-body system and the fluid circuits as continuous line elements. Individual fluid components are modelled using the impedance method, and the relationships between the fluid states at the extremities of each circuit are determined by the transfer matrix method. The resulting set of linear, frequency-dependent state-space equations, which govern the coupled dynamics of the half-car system, are derived and then applied in a variety of ways. This includes a free vibration analysis, ride comfort assessment and multi-objective optimisation. A number of key components that influence HIS performance are identified and a sensitivity analysis of their effects is presented.

Validation of the theoretical modelling is performed in two ways. First, simulations of an identical half-car using an alternative, nonlinear finite element fluid model are conducted. Second, experiments with a unique, purpose-built, half-car test rig are performed. The free and forced vibration results obtained with both methods, in general, agree very well with the proposed linear model.

The methodology presented is found to be an effective and useful way of modelling HIS-equipped vehicles, particularly in the frequency domain. The obtained results suggest that interconnected suspension schemes may provide, at least to some extent, an improved compromise between ride and handling. However, further investigation of this claim, including the development of a detailed full-car model, is recommended as a topic for future studies.

Dear Reader, please forgive me if I have wasted your time.

– Leonard Cohen