Development of an EV drivetrain for a small car

by

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Abstract

Electrical vehicles (EVs) have a significant role in reducing transportation emissions and the dependence on fossil fuels. The government policies and taxation arrangements to support EVs have created new opportunities. This research has focused on the mechanical design of an in-wheel drivetrain for a small car.

An in-wheel drivetrain has been selected to maximise the energy efficiency with reduced transmission loses. As a consequence, packaging of the motor has been confined to the rim envelope. Based on the selection of a specific small car, a corresponding low rolling resistance tyre has been chosen to increase the EV range. Accordingly, an optimised low mass rim has been developed. The conceptual and detailed design of the motor has been based on the rim envelope.

In the motor design stage, a focused literature review and specific commercial drivetrain study have been carried out, which revealed that majority of current EVs use permanent magnet motors (PMMs). The main drawbacks of PMM are dependency and availability risks as well as higher costs of the rare earth elements. The switch reluctance motor (SRM) has been identified as a feasible alternative, because it uses more readily available materials such as steel and copper for the magnetic path. However, SRMs have reduced power density for a specific mass, which may result in large unsprung mass and undesirable driving characteristics for the EV.

In the conceptual design stage, depending on the magnetic path orientation vertical and horizontal concepts were developed. Among the two alternative concepts developed, the vertical magnetic orientation using virtual reality (VR) has been chosen in the conceptual design. This is based on an assessment of: i) the maximised space utilisation, ii) the optimised magnetic path for stabilising 1mm air gap and iii) ease of the fitment and the assembly. In the detailed design, power density of the SRM has been further maximised by: (i) designing with two stabilised air gaps within the magnetic path and (ii) space utilisation by integrating the motor with the wheel rather than as an add-on arrangement. In the mechanical sense, the stiffness of motor parts has become crucial for stabilising the air gap between stator and rotor. Therefore, this research has
accomplished a thermo-structural optimisation using finite element methods (FEM) in ANSYS for motor shaft, hub and motor covers. Additionally, virtual reality (VR) and augmented reality (AR) based simulation studies have been carried out to maximise the space utilisation for the magnetic path.

EV performance analyses have been conducted for optimisation of: i) rim, ii) brake, iii) suspension, iv) vehicle weight distribution and v) tyre slip. As the motor occupies the existing brake space and with the available limited packaging space the light weight brake design was essential. To increase the thermal stability and in order to fit within the limited space an increased surface area and reduced thickness disc brake has been designed. To minimise the thermally excited instability, FEM based optimisation has been carried out.

A further ride and comfort study of an EV suspension using a quarter car and a Bode plot analysis has confirmed handling of the large unsprung mass. Life cycle of the EV suspension using the rain flow counting method and Palmgren-Miner rule has established the fatigue performance. The AR analysis has resulted in identification of operator safety and tyre serviceability using the NOISH equation. Finally, the performance of the EV specification has been demonstrated based on the range, fitment and the weight distribution models. In conclusion, this research has developed an in-wheel SRM drivetrain for a small car, which can be disseminated to the other vehicles with minimal modifications.

**Key words:** EV, drivetrain, SRM, in-wheel, FEM, VR, AR, brake, rim, tyre and suspension
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Dedicated to:

My lovely wife and two kids
Declaration

I declare that this thesis represents my own work and has not been previously submitted to meet requirements for an award at this or any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due acknowledgement has been made.

Ambarish Kulkarni

Date: 12-03-2014
# Contents

*Abstract*  
*Acknowledgements*  
*Declaration*  
*Contents*  
*List of Figures*  
*List of Tables*  
*List of symbols*  
*List of Abbreviations*

<table>
<thead>
<tr>
<th>Chapter 1</th>
<th>Introduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1 Chapter overview</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Background of research</td>
<td>1</td>
</tr>
<tr>
<td>1.2.1 Environmental issues</td>
<td>2</td>
</tr>
<tr>
<td>1.2.2 Government policies</td>
<td>4</td>
</tr>
<tr>
<td>1.2.3 Commercial viability</td>
<td>8</td>
</tr>
<tr>
<td>1.2.4 New opportunities</td>
<td>10</td>
</tr>
<tr>
<td>1.3 Research problem</td>
<td>13</td>
</tr>
<tr>
<td>1.4 Aim and objectives of research</td>
<td>14</td>
</tr>
<tr>
<td>1.5 Scope of the research</td>
<td>15</td>
</tr>
<tr>
<td>1.6 Overview methodological approach and outline</td>
<td>15</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Chapter 2</th>
<th>Selection of an EV drivetrain</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1 Chapter overview</td>
<td>19</td>
</tr>
<tr>
<td>2.2 Drivetrain selection</td>
<td>20</td>
</tr>
<tr>
<td>2.2.1 Conventional drivetrain</td>
<td>20</td>
</tr>
<tr>
<td>2.2.2 In-wheel drivetrain</td>
<td>21</td>
</tr>
<tr>
<td>2.2.3 By-wheel drivetrain</td>
<td>22</td>
</tr>
<tr>
<td>2.2.4 Drivetrain configuration selection</td>
<td>22</td>
</tr>
<tr>
<td>2.3 EV drivetrains in commercial cars</td>
<td>23</td>
</tr>
<tr>
<td>2.4 Motor selection</td>
<td>30</td>
</tr>
</tbody>
</table>
Chapter 4  Motor design for an EV drivetrain

4.1 Chapter overview 83
4.2 Motor conceptual design 84
  4.2.1 Preliminary free hand sketches 84
  4.2.2 Concept evaluation 88
  4.2.3 Magnetic path details 92
4.3 Motor detailed design 94
  4.3.1 Motor packaging 95
    4.3.1.1 Stage one motor design 96
    4.3.1.2 Stage two motor design 97
    4.3.1.3 Final motor design 98
  4.3.2 Bearing selection 101
  4.3.3 Shaft design 108
    4.3.3.1 Finite element modelling-shaft 109
    4.3.3.2 Load and boundary conditions-shaft 110
    4.3.3.3 Finite element results-shaft 110
    4.3.3.4 Finite element modelling-motor 111
    4.3.3.5 Load and boundary conditions-motor 112
    4.3.3.6 Finite element results-motor 113
    4.3.3.7 Nut selection for motor assembly 114
  4.3.3.6 Shaft final design 115
4.4 Summary 116

Chapter 5  Mechanical optimisation of motor

5.1 Chapter overview 121
5.2 Weight reduction of motor covers 122
  5.2.1 Material selection 122
  5.2.2 Mass optimisation 124
    5.2.2.1 Finite element modelling 124
    5.2.2.2 Load and boundary conditions 125
    5.2.2.3 Finite element results 126
5.3 Space utilisation and assembly sequencing 126
Chapter 5  

5.3.1 Virtual reality based space utilisation  
5.3.1.1 Virtual reality modelling  
5.3.1.2 Virtual reality results  
5.3.2 Augmented reality based assembly sequencing  
5.3.2.1 Virtual reality modelling  
5.3.2.2 Augmented reality modelling  
5.3.2.3 Augmented reality results  
5.4 Thermal stability analysis and cooling design  
5.4.1 Thermal stability analysis  
5.4.2 Forced cooling design  
5.5 Motor key characteristics  
5.6 Summary

Chapter 6  Primary mechanical braking system  
6.1 Chapter overview  
6.2 Brake design considerations  
6.2.1 Performance standards for brake design  
6.2.1.1 Ordinary performance test with cold brakes  
6.2.1.2 Fade and recovery test for hot braking  
6.2.2 Brake design parameters  
6.2.3 Mechanical brake selection  
6.2.3.1 Drum brake  
6.2.3.2 Disc brake  
6.2.4 Disc brake concepts  
6.3 Brake ADR compliance  
6.3.1 Brake force experimentation  
6.3.2 Brake ADR compliance study  
6.4 Brake disc optimisation  
6.4.1 Brake thermal experimentation  
6.4.2 Finite element methods  
6.4.2.1 Finite element modelling  
6.4.2.2 Material and boundary conditions  
6.4.2.3 Loading
Chapter 7  Evaluations of suspension system

7.1 Chapter overview

7.2 Suspension modelling
    7.2.1 Free body diagram
    7.2.2 Modelling

7.3 Suspension ride comfort analysis
    7.3.1 Sprung mass
    7.3.2 Unsprung mass
    7.3.3 Driver-seat mass
    7.3.4 Simscape analysis
    7.3.5 Bode plot analysis

7.4 Fatigue analysis
    7.4.1 Static study
        7.4.1.1 Finite element modelling
        7.4.1.2 Material
        7.4.1.3 Load and boundary conditions
        7.4.1.4 Finite element results
    7.4.2 Fatigue study
        7.4.2.1 Variable amplitude loads
        7.4.2.2 Rain flow counting method
        7.4.2.3 Finite element results

7.5 Summary

Chapter 8  Vehicle performance modelling

8.1 Chapter overview

8.2 Vehicle fitment
    8.2.1 Front bay fitment
    8.2.2 Rear bay fitment
Chapter 9  Conclusions and recommendations

9.1 Chapter overview 237
9.2 Research approach and key conclusions 237
9.3 Significant outcomes and recommendations 239
9.4 Novelty of research 240
  9.4.1 EV design 240
  9.4.2 In-wheel SRM 242
  9.4.3 Structured design methodology 243
  9.4.4 VR based simulations for evaluations 243
9.5 Endorsements and publications 244
  9.5.1 Positive feedbacks from conference and seminars 244
  9.5.2 Positive feedbacks from demonstration 244
  9.5.3 Acceptance from industry stake holders 245
  9.5.4 Publications 245
    9.5.4.1 Published articles 245
    9.5.4.2 Articles in press 246
    9.5.4.3 Submitted articles 246
Chapter 10 Future scope

10.1 Chapter overview 248
10.2 Scope of further research 248
  10.2.1 Switch reluctance motor 248
  10.2.2 Vehicle performance 248
  10.2.3 Adoption of SRM in other vehicle sectors 249
  10.2.4 Use of styling for improving vehicle range 249
  10.2.5 Enhancement of VR application for auto industry 249
  10.2.6 Improvements to design methodology 249

References 247

Appendices 267

Appendix 1 EV conventional and by-wheel drivetrains
Appendix 2 Scanning with Artec® 3D scanner
Appendix 3 Bazire curves and non-uniform B-spline mathematical methods
Appendix 4 Finite Element results for rim optimisation
Appendix 5 Drawing documentation
Appendix 6 Bearing SKF data sheet
Appendix 7 Shaft calculations
Appendix 8 Auto stereoscopic script file
Appendix 9 Thermal expansion of Aluminium bushes
Appendix 10 Motor characteristics
Appendix 11 Drum brakes
Appendix 12 Disc brakes and calipers
Appendix 13 Brake Testa Millenium
Appendix 14 MATLAB® program ICE
Appendix 15 MATLAB® program EV
Appendix 16 Simscape® model and results
List of figures

Figure 1.1 Government incentives for backing green vehicles
Figure 1.2 Daily retail petrol prices, 2007 to 2011—Australian cents per litre
Figure 1.3 Electric vehicle sales decline during early 1900
Figure 2.1 Conventional drivetrain
Figure 2.2 In-wheel drivetrain
Figure 2.3 By-wheel drivetrain
Figure 2.4 First EV concept
Figure 2.5 First in-wheel electric motor used late 1800s
Figure 2.6 tm4 motors
Figure 2.7 Michelin in-wheel technology
Figure 2.8 Siemens eCorner
Figure 2.9 Heinzmann electric hub motor
Figure 2.10 XTi Hub motors
Figure 2.11 Copenhagen wheel
Figure 2.12 e-Traction wheel concept
Figure 2.13 Protean electric hub motor
Figure 3.1 Dimension clearance analysis for mud guards of the Holden Barina Spark-a) rear wheel, b) front wheel
Figure 3.2 Hand held 3D Scanner
Figure 3.3 Two or three recordings of the car surface using 3D Scanner
Figure 3.4 Vehicle digitisation using hand held 3D Scanner (4 stages), a) point cloud, b) mesh creation, c) integration and d) final mesh model for refinement
Figure 3.5 Final digital model of the Holden Barina Spark
Figure 3.6 Digital model showing suspension and tyre details
Figure 3.7 Mud guard clearance, digitised Holden Barina Spark fitted with R17 wheel
Figure 3.8 J tyre configuration
| Figure 3.9 | Rim-Tyre nomenclature | 61 |
| Figure 3.10 | Rim 1, a positive offset single piece rim with solid end cap | 62 |
| Figure 3.11 | Rim 2, a zero offset single piece rim with solid end cap | 63 |
| Figure 3.12 | Rim 3, a positive offset two piece rim with hollow end cap | 63 |
| Figure 3.13 | Rim 4, a positive offset single piece rim with five support end cap | 64 |
| Figure 3.14 | Rim 5, a positive offset single piece rim with three support end cap | 64 |
| Figure 3.15 | Rim optimisation flow diagram using FE methods | 68 |
| Figure 3.16 | Rim 3 FE model | 69 |
| Figure 3.17 | Rim loads and boundary conditions | 72 |
| Figure 3.18 | Rim 3 Von-mises stress areas (enlarged area showing high stress concentration areas, stress in MPa) | 73 |
| Figure 3.19 | Rim 3 deformation analysis (mm) | 74 |
| Figure 3.20 | Rim 3 fatigue life cycles | 75 |
| Figure 3.21 | Tyre model for rolling resistance | 77 |
| Figure 3.22 | Comparison LRR & NRR tyres for RRC at different velocities | 79 |
| Figure 3.23 | Variation of rolling resistance coefficient with tyre inflation pressure | 80 |
| Figure 3.24 | Valve design-a) dimensional details, b) geometry details | 81 |
| Figure 4.1 | Initial hand sketch, sectional front view of rotors and stators arrangement for horizontal concept (NTS) | 85 |
| Figure 4.2 | Initial hand sketch, sectional front view of rotors and stators arrangement for vertical concept (NTS) | 85 |
| Figure 4.3 | Detailed half section of horizontal hand sketch concept proposal (NTS) | 86 |
| Figure 4.4 | Detailed half sectional vertical hand sketch concept proposal (NTS) | 87 |
| Figure 4.5 | Horizontal mounting concept using VR based visualisation | 90 |
| Figure 4.6 | Vertical mounting concept using VR based visualisation | 91 |
| Figure 4.7 | Stage one motor design half front sectional view | 97 |
Figure 4.8 Stage two motor design with full front sectional view
Figure 4.9 Final motor design full front sectional view
Figure 4.10 Bearing selection process
Figure 4.11 Motor tyre model for calculating bearing loads
Figure 4.12 Wheelbase model for bearing calculations
Figure 4.13 Bearing loads for hub design of an EV drivetrain
Figure 4.14 Bearing details: a) double row taper roller (inner) and b) deep groove (outer)
Figure 4.15 Shaft FE Model with a load and boundary conditions
Figure 4.16 Shaft results for maximum Von-mises stress concentrations (MPa)
Figure 4.17 Shaft results for maximum displacements (mm)
Figure 4.18 Half front section of motor FE model
Figure 4.19 Load case (40 degree) for motor assembly (transparent rim end cap for clarity)
Figure 4.20 Motor assembly results for maximum displacement (mm)
Figure 4.21 Final shaft design
Figure 5.1 Objective tree for motor optimisation
Figure 5.2 Motor cover FE model
Figure 5.3 Auto stereoscopic motion smoothening using spatial position
Figure 5.4 VR used for auto stereoscopic optimisation using 8 cameras and lights (Omni and spot lights applied) for space utilisation studies
Figure 5.5 VR used for auto stereoscopic optimisation for space utilisation studies
Figure 5.6 Virtual and augmented methodology used in this research
Figure 5.7 An example of using cyber glove II® and motion capture with EON® reality
Figure 5.8 EON® Prototype file on left and demonstration on right
Figure 5.9 Motor assembly sequence using AR tools
Figure 5.10 Motor curves for power & torque versus speed
Figure 5.11 Vehicle acceleration and velocity at time (up to t₀)
| Figure 6.1 | Drum brake schematic top view | 153 |
| Figure 6.2 | Disk brake schematic front view | 154 |
| Figure 6.3 | Disc brake 1 | 157 |
| Figure 6.4 | Disc brake 2 | 157 |
| Figure 6.5 | Disc brake 3 | 158 |
| Figure 6.6 | Disc brake 4 | 158 |
| Figure 6.7 | Disc brake 5 | 159 |
| Figure 6.8 | Disc brake 6 | 159 |
| Figure 6.9 | Brake phases | 160 |
| Figure 6.10 | Holden Barina Spark-Accelerometer data at 60-0km/h | 162 |
| Figure 6.11 | Holden Barina Spark-Accelerometer data at 100-0km/h | 163 |
| Figure 6.12 | Centre of gravity in the Holden Barina Spark | 164 |
| Figure 6.13 | Disc brake 5 FE model with load and boundary conditions | 169 |
| Figure 6.14 | Heat load input for each friction face on a disc brake | 170 |
| Figure 6.15 | Disc brake design 5, maximum temperature | 171 |
| Figure 6.16 | Disc brake 5, maximum displacement in mm | 172 |
| Figure 6.17 | Disc brake 5, maximum temperature in °C at 30 seconds | 172 |
| Figure 6.18 | Disc brake design 1 to 6, temperatures at time intervals | 173 |
| Figure 6.19 | Disc brake design 5 with 10mm and 5mm thickness | 173 |
| Figure 6.20 | Final mechanical brake system | 175 |
| Figure 6.21 | Proposed EV caliper dimensions: a) front view, b) side view (Brembo P4 30/34 C Fixed Caliper) | 177 |
| Figure 7.1 | Free body diagram of quarter-car model | 182 |
| Figure 7.2 | Block diagram of quarter-car model | 185 |
| Figure 7.3 | Velocity and displacement of sprung mass (ICE) | 188 |
| Figure 7.4 | Velocity and displacement of sprung mass (EV) | 189 |
| Figure 7.5 | Velocity and displacement of unsprung mass (ICE) | 190 |
| Figure 7.6 | Velocity and displacement of unsprung mass (EV) | 190 |
| Figure 7.7 | Velocity and displacement of driver-seat (ICE) | 191 |
| Figure 7.8 | Velocity and displacement of driver-seat (EV) | 191 |
| Figure 7.9 | Frequency response of ICE and EV vehicles using Bode Plot | 194 |
| Figure 7.10 | Suspension model of an in-wheel SRM | 197 |
Figure 7.11  Suspension FE model with load and boundary conditions  198
Figure 7.12  Material definition of suspension lugs and plates  199
Figure 7.13  High Von-mises stress concentration area in the EV suspension (Mpa)  200
Figure 7.14  Maximum displacement of lug in the EV suspension (scaled 100 times for clarity)  200
Figure 7.15  Load history curve  202
Figure 7.16  3D rain flow matrix for cycles of an in-wheel EV suspension  205
Figure 7.17  Damage plot for an in-wheel EV suspension  205
Figure 8.1  Front bay of the Holden Barina Spark ICE  210
Figure 8.2  Front bay of the Holden Barina Spark EV (front portion removed for clarity and dimensions in metre)  211
Figure 8.3  Rear bay of Holden Barina Spark EV (rear portion removed for clarity and dimensions in metre)  212
Figure 8.4  Muffler and silencer pipe in an existing Holden Barina Spark ICE  212
Figure 8.5  Lateral CG of the Holden Barina Spark ICE (dimensions in mm)  213
Figure 8.6  Longitudinal CG of the Holden Barina Spark EV (dimensions in mm)  215
Figure 8.7  Lateral CG of the Holden Barina Spark EV (dimensions in mm)  217
Figure 8.8  Forces acting on the vehicle at an inclined position  218
Figure 8.9  Variation of longitudinal force with slip (ICE & EV)  222
Figure 8.10  Motion capture experimentation for tyre servicing  224
Figure 8.11  Digital model mimicking the actual tyre change scenario  225
Figure 8.12  Discrete motions within BVH file  225
Figure 8.13  Biomechanical model front view for tyre servicing (dimension in mm and weight in Newton): a) during servicing, b) start of servicing  226
Figure 8.14  Digital mock-up for assembly sequencing-Motor and brakes were in place when wheel was dismantled  229
Appendix figures

Figure 1.1  GM Volt, motor and controller  268
Figure 1.2  Mitsubishi i-MiEV motor  268
Figure 1.3  Ford Focus electric motor  270
Figure 1.4  Honda FCEV motor  270
Figure 1.5  SIM drive  271
Figure 4.1  Rim 1 maximum Von-mises stress concentration  275
Figure 4.2  Rim 1 maximum deformation  275
Figure 4.3  Rim 1 maximum strain  276
Figure 4.4  Rim 1 lifecycles  276
Figure 4.5  Rim 2 maximum Von-mises stress concentration  277
Figure 4.6  Rim 2 maximum deformation  277
Figure 4.7  Rim 2 maximum strain  278
Figure 4.8  Rim 2 lifecycles  278
Figure 4.9  Rim 3 maximum strain  279
Figure 4.10  Rim 4 maximum Von-mises stress concentration  279
Figure 4.11  Rim 4 maximum deformation  280
Figure 4.12  Rim 4 maximum strain  280
Figure 4.13  Rim 4 lifecycles (predicted failure area enlarged)  281
Figure 4.14  Rim 5 maximum Von-mises stress concentration  281
Figure 4.15  Rim 5 maximum deformation  282
Figure 4.16  Rim 5 maximum strain  282
Figure 4.17  Rim 5 lifecycles (predicted failure area enlarged)  283
Figure 6.1  Bearing life factor for ball and roller bearings  289
Figure 9.1  Thermal expansion of the bush inside motor  294
Figure 9.2  Thermal expansion of the bush inside motor  294
Figure 11.1  Drum brake mechanisms. a) Simplex, b) Duplex, c) Duo Duplex, d) Servo, e) Duo Servo  296
Figure 11.2  Drum brake materials. 1) Grey cast iron, 2) Aluminium casting, cast iron insert, and 3) Aluminium/Ceramic casting  296
Figure 12.1  Fixed caliper: 1) brake rotor, 2) hydraulic connection, 3) brake piston and 4) bleed screws  297
Figure 12.2  Frame caliper: 1) brake rotor, 2) brake piston, 3) hydraulic connection, 4) bleed screw, 5) mounting, and 6) frame  

Figure 12.3  Fist caliper: 1) brake rotor; 2) brake piston, 3) hydraulic connection, 4) bushings, 5) mounting, and 6) frame  

Figure 14.1  Brake Testa Millennium, a) testing device (left) and b) load sensor on foot brake pedal (right)  

Figure 14.2  Holden Barina Spark-Brake test data at 60-0km/h  

Figure 14.3  Holden Barina Spark- Brake test data at 100-0km/h  

Figure 16.1  Simscape® model used in this research  

Figure 16.2  Velocity and displacement of sprung mass (EV)  

Figure 16.3  Velocity and displacement of unsprung mass (EV)  

Figure 16.4  Velocity and displacement of driver-seat mass (EV)
List of Tables

Table 1.1  National GHG Inventory May 2009  3
Table 2.1  Drivetrain comparisons  23
Table 2.2  Conventional and by-wheel drivetrain study  23
Table 2.3  In-wheel drivetrain study  29
Table 2.4  Comparisons table for four types of motor  46
Table 3.1  Small and medium cars commentary  51
Table 3.2  Wheel clearance analysis  53
Table 3.3  Comparison of different rim materials  67
Table 3.4  Summary on FE evaluations of different rim designs  73
Table 3.5  Comparison charts for rolling resistance at different velocities  78
Table 4.1  Comparison of motor concepts  91
Table 4.2  Magnetic path details for SRM concept  93
Table 5.1  Competitive analysis of motor cover materials  123
Table 5.2  Aluminium alloy properties of 6061-T6  124
Table 5.3  Design scenario for motor cover optimisation  126
Table 5.4  Specification for the in-wheel SRM  142
Table 6.1  Ordinary performance test conditions  149
Table 6.2  Fade and recovery test, brake system heating conditions  150
Table 6.3  Braking system data, small car, large car and motor cycle  161
Table 6.4  Brake test results using Brake Testa Millennium, 60-0km/h  162
Table 6.5  Typical values for thermal design, automotive brakes  166
Table 6.6  Average temperature of the Holden Barina Spark using MT100 thermometer, 100-0km/h  167
Table 6.7  Thermal variations before and after the event using MT100 digital infrared thermometer, 100-0km/h  167
Table 6.8  Disc brake 1 to 6 FE results  170
Table 7.1  Descriptions of variables used in equations  182
Table 7.2  ICE and EV specifications considered for quarter car  184
modelling

Table 8.1  Longitudinal weight variations of an EV  214
Table 8.2  Lateral weight variations of an EV  216
Table 8.3  Values of Coefficient $b_i$  221
Table 8.4  Reduction factors for NIOSH equation  228
Table 8.5  Test conditions for the vehicle  231
Table 8.6  Test results with 72 battery packs EV design  232
Table 8.7  Test results with 36 battery packs EV design  232
Table 8.8  Vehicle specification sheet  233

Appendix tables

Table 10.1  Motor characteristic derivations  295
List of symbols (in alphabetical order)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>Motor half width in millimetre or acceleration in metre per second square or numerical constant for a control point</td>
</tr>
<tr>
<td>$a_m$</td>
<td>Mean deceleration in metre per second square</td>
</tr>
<tr>
<td>$a_v$</td>
<td>Total deceleration in metre per second square</td>
</tr>
<tr>
<td>$a_{avg}$</td>
<td>Vehicle average acceleration and deceleration in metre per second square</td>
</tr>
<tr>
<td>$A$</td>
<td>Area or vehicle frontal area in metre square</td>
</tr>
<tr>
<td>$A_f$</td>
<td>Difference of the weight added to removed from the front bay in kilogram or asymmetric reduction factor</td>
</tr>
<tr>
<td>$A_l$</td>
<td>Action Limit in kilogram</td>
</tr>
<tr>
<td>$A_r$</td>
<td>Difference of the weight added and removed from the rear bay in kilogram or average range of the vehicle in kilometre</td>
</tr>
<tr>
<td>$A_{lc}$</td>
<td>Difference of the weight added and removed from the left side in kilogram</td>
</tr>
<tr>
<td>$A_{rl}$</td>
<td>Difference of the weight added and removed from the right side in kilogram</td>
</tr>
<tr>
<td>$A_{th}$</td>
<td>Thread stress area in metre square</td>
</tr>
<tr>
<td>$A_{cfm}$</td>
<td>Air flow rate in Cubic Feet per Minute</td>
</tr>
<tr>
<td>$b$</td>
<td>Magnetic flux density in tesla or numerical constant for a control point or Basquin slope or distance from the center of the body to the L5/S1 disc</td>
</tr>
<tr>
<td>$b_h$</td>
<td>Harmonic flux density in tesla</td>
</tr>
<tr>
<td>$b_i$</td>
<td>Tyre coefficient values</td>
</tr>
<tr>
<td>$B$</td>
<td>Rolling coefficient</td>
</tr>
<tr>
<td>$B_c$</td>
<td>Battery consumption in Ampere hour</td>
</tr>
<tr>
<td>$c$</td>
<td>Distance from internal muscle force to the L5/S1 disc or numerical constant for a control point</td>
</tr>
<tr>
<td>$C$</td>
<td>Rolling coefficient</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Coefficient of vehicle drag resistance</td>
</tr>
<tr>
<td>$C_f$</td>
<td>Coupling reduction factor</td>
</tr>
<tr>
<td>$C_r$</td>
<td>Coefficient of tyre rolling resistance or dynamic load rating in Newton</td>
</tr>
</tbody>
</table>
\( C_s \) Damping coefficient of the suspension in Newton second per metre
\( C_{ds} \) Damping coefficient of the seat in Newton second per metre
\( CG_{he} \) EV height of the CG from ground line in millimetre
\( d \) Major diameter or distance travelled in metre or spring coil diameter in millimetre
\( d_m \) Fully developed deceleration in metre per second square
\( d_p \) Pitch circle diameter of thread in millimetre
\( D \) Tyre diameter in millimetre square or damage percentage or rolling coefficient or shaft diameter in millimetre
\( D_f \) Distance reduction factor
\( D_t \) Vertical distance travelled by load in millimetre
\( E \) Young’s modulus in Giga Pascal or rolling coefficient
\( E_b \) Energy of battery in Joules
\( f \) Rolling resistance coefficient or Foot control in Deca Newton
\( f_0 \) Initial rolling coefficient
\( f_1 \) Final rolling coefficient of LRR
\( f_2 \) Final rolling coefficient of NRR
\( f_h \) Life factor
\( f_n \) Speed factor
\( F' \) Load in Newton
\( F_b \) Total vehicle braking force in Newton
\( F_c \) Compression force in Newton
\( F_d \) Drag force in Newton
\( F_f \) Frequency reduction factor
\( F_t \) Total force required to overcome drag and gradient force in Newton
\( F_{xr} \) Vertical force acting at an angle of 45° on rear wheel in Newton
\( F_M \) radial magnetic flux in metre square kilogram per second square
\( F_l \) Longitudinal force acting on front wheel in Newton
\( F_r \) Rolling resistance or longitudinal force acting on rear wheel in Newton
\( F_s \) Shear force of nut in Newton
\( F_x \) Longitudinal force in Newton
\( F_y \) Lateral force in Newton
$F_z$  Upward direction force at tyre ground contact or a Vertical force in Newton

$F_G$  Gradient force in Newton

$F_R$  Rolling resistance force in Newton

$F_{bf}$  Total front braking force in Newton

$F_{br}$  Total rear braking force in Newton

$F_{im}$  Internal muscle force in Newton

$F_{wf}$  Total ICE weight on front axle in kilogram

$F'_{wf}$  Total EV weight on front axle in kilogram

$F_{wr}$  Total on rear axle in kilogram

$F'_{wr}$  Total EV weight on rear axle in kilogram

$F_{wt}$  Total ICE weight in kilogram

$F'_{wt}$  Total EV weight in kilogram

$F_{wle}$  Total ICE weight on left side in kilogram

$F'_{wle}$  Total EV weight on left side in kilogram

$F_{wri}$  Total ICE weight on right side in kilogram

$F'_{wri}$  Total EV weight on right side in kilogram

$F_{avg}$  Average frequencies

$F_{max}$  Maximum frequencies

$FR_{inner}$  Inner forces in Newton

$FR_{outer}$  Outer forces in Newton

$g$  Acceleration due to gravity in metre per second square

$G$  Shear modulus of the spring in Pascal or weight of the EV in kilogram

$G_m$  Gerber mean stress correction factor

$h$  Width of rim-tyre or distance from the load to the L5/S1 disc in millimetre

$h_a$  Height of the front axle from the ground in millimetre

$h_c$  Heat transfer coefficient in watt metre square per degree Celsius

$h_l$  Heat load in joules

$h_{ai}$  Height of the front axle of the elevated EV in millimetre

$h_{cg}$  ICE height of the CG from ground line in millimetre

$H_f$  Horizontal reduction factor
\( H_0 \)  Distance from horizontal origin to inner ankle bones mid plane in millimetre

\( I \)  Mass moment of inertia in metre to the power of four

\( J \)  Polar moment of inertia in metre to the power of four

\( K \)  Spring constant in Newton per metre or load constant in kilogram or thermal resistance coefficient in metre square degree Celsius per watt

\( K_c \)  Corrosion reduction factor

\( K_f \)  Fatigue strength reduction factor

\( K_l \)  Size reduction factor

\( K_m \)  Loading mode reduction factor

\( K_n \)  Notch effects reduction factor

\( K_r \)  Reliability reduction factor

\( K_s \)  Vehicle suspension spring constant in Newton per metre

\( K_t \)  Tyre spring constant in Newton per metre or temperature reduction factor

\( K_{ds} \)  Seat suspension spring constant in Newton per metre

\( K_{pf} \)  Motor power factor at continuous rating

\( K_{freq} \)  Frequency reduction factor

\( K_{frett} \)  Fretting reduction factor

\( K_{a\text{inner}} \)  Inner axial force in Newton

\( K_{a\text{outer}} \)  Outer axial force in Newton

\( KE \)  Kinetic Energy in Joules

\( l \)  ICE Wheel base length or motor width in millimetre

\( l' \)  EV Wheel base length in millimetre

\( l_a' \)  EV wheel lateral distance in millimetre

\( l_f \)  ICE longitudinal distance of CG to front axle of wheelbase in millimetre

\( l_f' \)  EV longitudinal distance of CG to front axle of wheelbase in millimetre

\( l_i \)  Length of the reduced wheel base when an EV is on the incline in millimetre

\( l_r \)  ICE longitudinal distance CG to rear axle of wheelbase in millimetre

\( l_r' \)  EV longitudinal distance CG to rear axle of wheelbase in millimetre
\( l_{a_l} \)  \( \text{EV lateral distance of CG to the wheel center on left in millimetre} \)
\( l_{a_r} \)  \( \text{EV lateral distance of CG to the wheel center on right in millimetre} \)
\( L \)  \( \text{Wheelbase or length or linear expansion in metre} \)
\( L_i \)  \( \text{Load index} \)
\( L_1 \)  \( \text{Initial length in metre} \)
\( L_e \)  \( \text{Thread management length in millimetre} \)
\( m \) or \( m_v \)  \( \text{Mass of the vehicle in kilogram} \)
\( m_m \)  \( \text{Maximum motor mass in kilogram} \)
\( m' \)  \( \text{Weight of EV in kilogram} \)
\( M_b \)  \( \text{Braking moment in Newton} \)
\( M_e \)  \( \text{Motor efficiency} \)
\( M_s \)  \( \text{ICE sprung mass in kilogram} \)
\( M_u \)  \( \text{ICE unsprung mass in kilogram} \)
\( M_s \)  \( \text{Longitudinal force coefficient} \)
\( M_{ds} \)  \( \text{ICE Driver seat mass in kilogram} \)
\( M_{lt} \)  \( \text{Moment at L5/S1 spline in metre to the power of four} \)
\( M_{se} \)  \( \text{EV sprung mass in kilogram} \)
\( M_{ue} \)  \( \text{EV unsprung mass in kilogram} \)
\( M_{ce} \)  \( \text{Motor controller efficiency} \)
\( M(x) \)  \( \text{Maximum moment, where } x \text{ is equal to zero} \)
\( n \)  \( \text{Number of coils or motors or number of threads per inch} \)
\( n_i \)  \( \text{Number of cycles of type } i \)
\( N \)  \( \text{Number of cells or order of the curve or number of cycles} \)
\( N_a \)  \( \text{Number of accelerations} \)
\( N_i \)  \( \text{Fatigue lifetime for all occurring cycles of type } i \)
\( P \)  \( \text{Power output or motor power in kilo watt or load on the shaft in Newton} \)
\( P_a \)  \( \text{Acceleration power required in kilowatt} \)
\( P_r \)  \( \text{Power required by the vehicle in kilowatt} \)
\( P_i \)  \( \text{Average power required by vehicle in kilowatt} \)
\( r \)  \( \text{Turning radius or wheel rolling radius or spring radius in metre} \)
\( R \)  \( \text{Radius of the wheel in millimetre or ratio in millimetre square per kilogram} \)
$R_e$ Rolling radius in metre

$R_{hl}$ Ride height in metre

$R_l$ Deformed radius of the wheel in millimetre

$S$ Stopping distance in metre

$S_b$ Distance travelled between $V_0$ and $V_b$ in metre

$S_e$ Distance travelled between $V_0$ and $V_e$ in metre

$S_h$ or $S_v$ Rolling coefficient

$t$ Time or intervals in second

$t_a$ Vehicle acceleration time in second

$t_t$ Test time in second

$T_1$ Initial temperature or allowable temperature in degrees Celsius

$T_2$ Final or inlet temperature in degrees Celsius

$\Delta T$ Average temperature in degree Celsius

$T_{cond}$ Thermal conductivity in watts per metre per degree Celsius

$T_{conv}$ Thermal convection in watts per metre per degree Celsius

$T_{start}$ Start temperature in degree Celsius

$T_{finish}$ Finish temperature in degree Celsius

$v$ Velocity in metre per second

$v_{max}$ Maximum shear in Newtons or Vehicle speed in metre per second

$V$ Volume of motor in millimetre cube or cell capacity in voltage or speed in kilometre per hour or velocity in metre per second or vehicle test speed in metre per second

$V_0$ Initial vehicle speed in kilometre per hour

$V_b$ Vehicle speed at $0.8V_0$ in kilometre per hour

$V_e$ Vehicle speed at $0.1V_0$ in kilometre per hour

$V_f$ Vertical reduction factor

$V_n$ Minimum DC link voltage in Volt

$V_o$ Distance from vertical origin to inner ankle bones mid plane in millimetre

$w(x)$ Displacement in metre

$w_{max}$ Maximum deflection in millimetres

$W$ Kerb weight of the car in kilogram

$W_l$ Lifting weight in kilogram
$W_t$ Torso weight in kilogram

$x_n$ Space state variables or values of control points in $x$

$X_n$ Road input

$y_n$ Values of control points in $y$

$\alpha$ Coefficient of linear expansion per degree Celsius or gradient angle at an incline in degree or angle of incline of L5/S1 disc from horizontal plane in degree

$\sigma_u$ Ultimate tensile strength in Mega Pascal

$\sigma_s$ Longitudinal slip

$\sigma_{alt}$ Alternating stress in Mega Pascal

$\sigma_{end}$ Endurance limit in Mega Pascal

$\sigma_{max}$ Maximum stress in Mega Pascal

$\sigma_{min}$ Minimum stress in Mega Pascal

$\sigma_{ult}$ Ultimate tensile stress in Mega Pascal

$\Sigma$ Stress in Mega Pascal

$C$ Strain in Mega Pascal

$\mu_0$ Permeability in henrys per metre

$\eta_m$ Motor efficiency at continuous rating

$\Omega$ Angular velocity in radian per second

$\Omega_o$ Increased angular velocity in radian per second

$\rho$ Air density in kilograms per metre cube or material density in grams per metre cube

$\theta$ Asymmetric angle in degree

$\tau$ Shear strength of the material in Mega Pascal or motor torque in Newton metre

$\tau_m$ Maximum wheel braking torque under a fault in Newton metre

$\tau_{max}$ Motor torque in Newton metre

$v$ Poisons ratio

$\omega$ Angular speed in radian per second
# List of abbreviations (in alphabetical order)

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>2D</td>
<td>Two Dimensional</td>
</tr>
<tr>
<td>3D</td>
<td>Three Dimensional</td>
</tr>
<tr>
<td>ABS</td>
<td>Australian Bureau of Statistics</td>
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<tr>
<td>ABS</td>
<td>Anti-lock Brake System</td>
</tr>
<tr>
<td>AC</td>
<td>Alternate Current</td>
</tr>
<tr>
<td>Auto CRC</td>
<td>Commonwealth Automotive Research Council</td>
</tr>
<tr>
<td>ADR</td>
<td>Australian Design Rule</td>
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<tr>
<td>AFPM</td>
<td>Axial Flux Permanent Magnet</td>
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<tr>
<td>AR</td>
<td>Augmented Reality</td>
</tr>
<tr>
<td>AS</td>
<td>Australian Standards</td>
</tr>
<tr>
<td>ASME</td>
<td>American Society of Mechanical Engineers</td>
</tr>
<tr>
<td>ASTM</td>
<td>American Society for Testing and Materials</td>
</tr>
<tr>
<td>BDCM</td>
<td>Brushless Direct Current Motor</td>
</tr>
<tr>
<td>BMS</td>
<td>Battery Management System</td>
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<tr>
<td>BOM</td>
<td>Bill of Material</td>
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<tr>
<td>BVH</td>
<td>Bio Vision Hierarchy</td>
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<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
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<tr>
<td>CFM</td>
<td>Cubic Feet per Minute</td>
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<tr>
<td>CG</td>
<td>Centre of Gravity</td>
</tr>
<tr>
<td>cpl</td>
<td>Cents per Liter</td>
</tr>
<tr>
<td>C-SiC</td>
<td>Carbon Ceramic Matrix Composite</td>
</tr>
<tr>
<td>CSIRO</td>
<td>Commonwealth Scientific and Industrial Research Organisation</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon Dioxide</td>
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<tr>
<td>DC</td>
<td>Direct Current</td>
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<tr>
<td>DC</td>
<td>Drop Center</td>
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<tr>
<td>DFE</td>
<td>Design for Environment</td>
</tr>
<tr>
<td>DFM</td>
<td>Design for Manufacture</td>
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<tr>
<td>DOF</td>
<td>Degree of Freedom</td>
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<tr>
<td>EHB</td>
<td>Electro Hydraulic Brake</td>
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<tr>
<td>EMB</td>
<td>Electro Mechanical Brake</td>
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<tr>
<td>EN</td>
<td>Engineering</td>
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<tr>
<td>Abbreviation</td>
<td>Full Form</td>
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<tr>
<td>EPA</td>
<td>Environmental Protection Agency</td>
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<tr>
<td>ESC</td>
<td>Electronic Stability Control</td>
</tr>
<tr>
<td>ESP</td>
<td>Electronic Stability Program</td>
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<tr>
<td>ETRTO</td>
<td>European Tyre and Rim Technical Organisation</td>
</tr>
<tr>
<td>EV</td>
<td>Electric Vehicle</td>
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<tr>
<td>EWB</td>
<td>Electronic Wedge Brake</td>
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<tr>
<td>FC</td>
<td>Fuel Cell</td>
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<tr>
<td>FCEV</td>
<td>Fuel Cell Electric Vehicle</td>
</tr>
<tr>
<td>FE</td>
<td>Finite Element</td>
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<tr>
<td>FEM</td>
<td>Finite Element Methods</td>
</tr>
<tr>
<td>FOC</td>
<td>Field Orientation Control</td>
</tr>
<tr>
<td>FOS</td>
<td>Factor of Safety</td>
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<tr>
<td>GDP</td>
<td>Gross Domestic Product</td>
</tr>
<tr>
<td>GDT</td>
<td>Geometric Dimensioning and Tolerance</td>
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<tr>
<td>GHG</td>
<td>Green House Gas</td>
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<tr>
<td>GM</td>
<td>General Motors</td>
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<tr>
<td>GVM</td>
<td>Gross Vehicle Mass</td>
</tr>
<tr>
<td>HEV</td>
<td>Hybrid Electric Vehicle</td>
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<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
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<tr>
<td>IEC</td>
<td>International Electro technical Commission</td>
</tr>
<tr>
<td>IEEE</td>
<td>Institute of Electrical and Electronics Engineers</td>
</tr>
<tr>
<td>IM</td>
<td>Induction Motor</td>
</tr>
<tr>
<td>IP</td>
<td>Ingress protection</td>
</tr>
<tr>
<td>IPCC</td>
<td>Intergovernmental Panel on Climate Change</td>
</tr>
<tr>
<td>IS</td>
<td>International Standards</td>
</tr>
<tr>
<td>ISO</td>
<td>International Standard Organisation</td>
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<tr>
<td>JIS</td>
<td>Japanese Industrial Standards</td>
</tr>
<tr>
<td>KERBS</td>
<td>Kinetic Energy Recovery Brake System</td>
</tr>
<tr>
<td>L5/S1</td>
<td>Lumbar Five and Fused Sacral Vertebrae One</td>
</tr>
<tr>
<td>LiFePO₄</td>
<td>Lithium Iron Phosphate</td>
</tr>
<tr>
<td>LRR</td>
<td>Low Rolling Resistance</td>
</tr>
<tr>
<td>LSV</td>
<td>Low-Speed Vehicle</td>
</tr>
<tr>
<td>MIT</td>
<td>Massachusetts Institute of Technology</td>
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<tr>
<td>Abbreviation</td>
<td>Full Form</td>
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<tr>
<td>MC</td>
<td>Multi Combination</td>
</tr>
<tr>
<td>MPG</td>
<td>Miles per Gallon</td>
</tr>
<tr>
<td>NIOSH</td>
<td>National institute of Occupational safety and Health</td>
</tr>
<tr>
<td>NRR</td>
<td>Normal Rolling Resistance</td>
</tr>
<tr>
<td>NURBS</td>
<td>Non-Uniform Rational Bezier Spline</td>
</tr>
<tr>
<td>OEM</td>
<td>Original Equipment Manufacturer</td>
</tr>
<tr>
<td>PCD</td>
<td>Pitch Circle Diameter</td>
</tr>
<tr>
<td>PDM</td>
<td>Product Data Management</td>
</tr>
<tr>
<td>PhD</td>
<td>Doctor of Philosophy</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional Integral Derivative</td>
</tr>
<tr>
<td>PM</td>
<td>Permanent Magnet</td>
</tr>
<tr>
<td>PMM</td>
<td>Permanent Magnet Motor</td>
</tr>
<tr>
<td>R&amp;D</td>
<td>Research and Development</td>
</tr>
<tr>
<td>RFPM</td>
<td>Radial Flux Permanent Magnet</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolution Per Minute</td>
</tr>
<tr>
<td>RRC</td>
<td>Rolling Resistance Coefficient</td>
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<tr>
<td>SAE</td>
<td>Society of Automotive Engineers</td>
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<tr>
<td>SAT</td>
<td>Segmented Armature Torus</td>
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<tr>
<td>SOP</td>
<td>Standard Operating Procedure</td>
</tr>
<tr>
<td>SRM</td>
<td>Switched Reluctance Motor</td>
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<tr>
<td>SUT</td>
<td>Swinburne University of Technology</td>
</tr>
<tr>
<td>SUV</td>
<td>Sports Utility Vehicle</td>
</tr>
<tr>
<td>TF</td>
<td>Transverse Flux</td>
</tr>
<tr>
<td>TFM</td>
<td>Transverse Flux Motor</td>
</tr>
<tr>
<td>TRA</td>
<td>Tyre and Rim Association</td>
</tr>
<tr>
<td>UM</td>
<td>User Manual</td>
</tr>
<tr>
<td>VDP</td>
<td>Virtual Design Process</td>
</tr>
<tr>
<td>VP</td>
<td>Virtual prototype</td>
</tr>
<tr>
<td>VPAC</td>
<td>Victorian Partnership for Advanced Computing</td>
</tr>
<tr>
<td>VR</td>
<td>Virtual Reality</td>
</tr>
<tr>
<td>WDC</td>
<td>Wide Drop Centre</td>
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</table>
Chapter 1

Introduction

1.1 Chapter overview

This chapter introduces the background of this research, which demonstrates increasing acceptance of electric vehicles (EVs) as alternative sustainable vehicles based on four criteria, i) environmental issues, ii) government policies, iii) commercial viability, and iv) new technological opportunities. The next section covers an overview of the research problem and the need for drivetrain research. Later section emphasises the aim, objectives and the basic research methodology for the EV drivetrain. The last section includes a summary of the thesis arrangements.

1.2 Background of research

Global demands for sustainability and compelling requirements to reduce environmental impact have encouraged the transportation industry to look into ways to reduce carbon emissions. In recent years, EV development has entered a new paradigm due to environmental pollution, global warming and depletion of fossil fuels. Unlike automobiles with internal combustion engines (ICEs), EVs have the intrinsic advantage of zero emission during operation, when a renewable energy source is used for charging batteries. Amongst various green transportation solutions, the EV has been one of the most valuable technologies and it is a commercially viable solution as well. Additionally EVs have numerous advantages over fossil-fuel-powered vehicles ranging from their reduced impact on the environment to smooth and quiet operation. There is been relative reluctance for adoption of EVs as alternatives to ICE, as this arise mainly due to: i) an established EV market with large sum capital investments and the supporting infrastructure, ii) big oil companies lobbying governments, and iii) customer emotional and sensory appeal with technical advantages of ICEs such as range and high power. EVs have been seen as too expensive a decade ago, but in light of the recent fossil fuel prices and emphasis on "going green and economic sustainability," the technology has now gathered more support and attention.
The success of the EV has been dependent on performance based on established ICE technology. Globally, EVs acceptance has been gaining momentum and many factors influence potential acceptance of EVs as a sustainable transport solution in the near future. The acceptance of EVs has been influenced by environmental issues, government policies, cost viability and finally the opportunity to develop new technologies by original equipment manufacturers (OEMs, also referred to as the vehicle manufacturers). Environmental issues have encouraged government policies to subsidise green vehicles. Decade ago EVs were seen as impractical; however rising fuel prices have led to the commercial viability. These developments have created a new opportunity for all stake holders in the EV industry to develop an innovative automotive technology.

1.2.1 Environmental issues

Fossil fuels (e.g., petrol and diesel) in ICE cars produce exhaust of carbon dioxide (CO₂). CO₂, although not directly harmful to human health, has been the most significant of the greenhouse gas (GHG) emissions contributing to climate change. Among the environment hazards, climate change influenced by global warming has become a challenge today. Average global temperatures are around 0.76°C higher than the preindustrial level and they are rising rapidly (Mondal, Kumar et al. 2011). The paper identifies that the transport sector is growing at a faster rate and is positioned second, after the industrial sector, with 22% to 24% of global GHG emissions from fossil fuels.

The report by the Inter-Governmental Panel on Climate Change (IPCC), discussed future predictions about global mean temperature and sea level dependency and their implications for proposed CO₂ emission limitations (Wigley, Jain et al. 1997). The report compared the IPCC predictions IS92a-f (IPCC predictions) with NL-2% (the Netherlands 2%). The report considered only the most extreme case, in this instance NL-2% is used as a proposal gauge amongst several limitation proposals. In the IPCC range of six prediction scenarios (IS92a to 1S92f), it mainly considers the IS92c, the lowest emission scenario (based on low economic and population growth with a climate sensitivity of 1.5°C) and IS92e, the highest estimates of future emissions (based on high economic and population growth with a climate sensitivity of 4.5°C) to compare the changes in predicted rise in temperature and sea level. According to this report, by
2100, relative to the no-limitation cases, the reduction in global mean temperature increase resulting from the NL-2% limitation proposal will range between 0.1°C (i.e., from 0.7°C down to 0.6°C for IS92c) and 0.9°C (i.e., from 3.9°C down to 3.0°C for IS92e), while the reduction in sea level rise will range between 2 cm (i.e., from 12cm down to 10cm for IS92c) and 15cm (i.e., from 100cm down to 85cm for IS92e).

According to the Australian Bureau of Statistics (ABS) report, Australia, a low population country (0.3% of the world population) contributes 1.5% of total GHG emission globally (Brian 2010). This report compares carbon emission by sector in Australia (Table 1.1) and finds that transport is the fourth largest, with 13.2% of Australia’s net emissions. According to the Environmental Protection Agency (EPA) Victoria, fossil fuel combustion in the automobile industry, has been identified as the largest per capita air polluter in the world and Melbourne is the second most per capita polluted city in Australia (EPA Victoria 2010).

Table 1.1: National GHG Inventory May 2009, Source: (Brian 2010)

<table>
<thead>
<tr>
<th>Net GHG emissions by sector</th>
<th>Emissions Mt CO₂ (Emissions in million tons)</th>
<th>Percentage of total emissions</th>
<th>Percentage change in emissions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Year</td>
<td>1990</td>
<td>2007</td>
<td>1990 to 2007</td>
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<tr>
<td>Energy</td>
<td>286.4</td>
<td>408.2</td>
<td>68.4</td>
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<tr>
<td>Stationary energy</td>
<td>195.1</td>
<td>291.7</td>
<td>48.8</td>
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<tr>
<td>Transport</td>
<td>62.1</td>
<td>78.8</td>
<td>13.2</td>
</tr>
<tr>
<td>Fugitive emissions</td>
<td>29.2</td>
<td>37.7</td>
<td>6.3</td>
</tr>
<tr>
<td>Agriculture</td>
<td>86.8</td>
<td>88.1</td>
<td>14.8</td>
</tr>
<tr>
<td>Land use (change and forestry)</td>
<td>131.5</td>
<td>56.0</td>
<td>9.4</td>
</tr>
<tr>
<td>Industrial processes</td>
<td>24.1</td>
<td>30.3</td>
<td>5.1</td>
</tr>
<tr>
<td>Waste</td>
<td>18.8</td>
<td>14.6</td>
<td>2.4</td>
</tr>
<tr>
<td>Australia’s net emissions</td>
<td>547.7</td>
<td>597.2</td>
<td>100.0</td>
</tr>
</tbody>
</table>

Recent comparisons based on fuel economy using Environment Protection Agency (EPA) standards, rated the top 9 cars out of 10 as EVs or hybrid electric vehicles (HEVs) (US Department of Energy 2013). Vehicles are ranked by their combined rating (weighted by 55% on the city and 45% on the highway) and EVs are measured in Miles per Gallon equivalent (MPG) where 33.7kWh equals 1gallon of gasoline. EVs are cleaner and more ecofriendly cars, when compared to conventional ICE run cars. Comparative analysis on energy efficiency of drivetrains established a potential doubling of efficiencies using EVs and HEVs (Ahman 2001). According to Jacobson
M. Z., use of wind, solar-photovoltaic, concentrated solar power, geothermal, tidal, wave, and hydroelectric energy combined with EVs would have the advantage of reducing global warming and pollution (2009). The review evaluated nine energy sources, two liquid fuels (E-85, Corn Ethanol and Cellulosic) and three vehicle types (the EV, a fuel cell electric vehicle (FCEV) and flex-fuel vehicles with E85) to address solutions for global warming and air pollution. In summary the paper concludes that energy source is the main criterion for reduction in global warming and emissions. In Jacobson’s comparison, the EV with a wind generated energy source would be ranked top followed by a solar generated energy source. Compared to the ICE, the EV combined with renewable energy sources has the potential to reduce global warming and carbon emission.

1.2.2 Government policies

Government policies and support are vital for successful technology transition in the automotive industry. Recently, efficient renewable energy technologies have been encouraged through a series of innovative initiatives and public policies by various governments to overcome environmental hazards caused by fossil fuels (Flavin C. 2008). The report predicts growth in the renewable energy market and substantial reduction in carbon emissions in the next decade. This prediction is based on double digit market growth (capital flows of more than US$100 Billion for a National Electrical Superhighway using high-voltage in the US alone), decline in technology costs, and government policies to replace fossil fuels. Some of the government initiatives have been tax cuts on green cars, subsidies on EVs/HEVs pricing, an annual rebate on running costs of green cars, increased tax on fuel inefficient cars, tax increase on fossil fuels, government warranty on the green vehicle sales, and funding for development of infrastructures (charging stations, smart grids and renewable energy development). Government support for automotive research, supply chain, training of maintenance personnel, and educating consumers on EV technology acceptance has increased globally. Figure 1.1 is a summary of government incentives adopted across the globe to influence use of green vehicle technologies.
In this section, as case example the policy influence of the governments from Japan, China, USA, UK, and Australia are discussed. The Japanese government is mitigating energy and environmental issues by means of legislation and standards. It is supporting automotive research and development (R&D), demonstration programmes, market support (artificially created niche markets), long-term strategic plans, and easing the way for sustainable technologies (Ahman 2006). According to this paper, since 1970, the Japanese government has adopted long-term ambitious policies and strategic plans for EVs. The success of HEVs was partly attributed to the Japanese government policies; one such example is the drivetrain developed for the EV, which was used for the HEV. Ahman concluded that technology survival is dependent on policies driven by flexibility, adaptability and co-operation in technical choice (2006). Market support in the early phases of the EV development was an important move by the Japanese government for the success of HEVs.

To encourage customers for the green vehicle purchase many countries have introduced labelling systems and/or incentives based on the fuel economy (Oliver, Gallagher et al. 2009). These labelling systems encourage the purchase of car by informing customer about urban and rural driving economy. Examples of such countries include Australia, Japan, USA, Canada and China. From 2006 to 2008, the government of China modified its excise scheme for vehicles such that small cars (with less than 1.5L engine) will have a reduced excise (from 5% to as low as 1%) and high powered vehicles (with a more
than 2.5L engine) will have increased excise (8% to 12-40%). Furthermore, the Chinese government has also reduced purchase tax on small cars (with an engine smaller than 1.6L) from 10% to 5% in 2009. The paper concludes that government policies (e.g., fuel economy standards) when regulated can encourage auto companies to adopt fuel efficient technologies and result in reducing net national emissions.

Research by Diamond D., in the USA, examined the relationship from government incentive and HEV adoption patterns by the buyer (2009). This research concluded that HEV adoption is a choice driven by higher fuel prices and incentive policies. The relationship between incentive policies and market share is weaker, as the incentive benefits are not reaching the customers (e.g., dealers reaping incentive benefits and incentive payment delays by states in the USA). The paper recommends an upfront incentive policy, on behalf of the US Government to increase sales of green cars (Diamond 2009). The Obama Government, announced a plan aiming for one million advanced technology vehicles on the road by 2015 (The White House-USA January 26, 2011). The US Government plans to achieve these by providing up to US$7500 tax credit on EV purchase, investment in R&D (electric drive, batteries, and energy storage technologies) and competitive grants to communities investing in infrastructure for supporting EVs.

The UK Government in January 2011 announced a 25% discount on green vehicles (to a maximum of £5,000) (Ray Massey - Associated Newspapers Ltd - Daily Mail 18 December 2010). Some of vehicles associated with consumer incentive schemes were, Mitsubishi i-MiEV (January 2011, £28,990 to £23,990), smart for two (January 2012, £16,000 to £12,000), Peugeot iOn (January 2011, £519.16 £415 for a month lease-only), Nissan LEAF (March 2011, £28,990 to £23,900), Tata Vista (March 2011, £28,600 to £23,600), Citroen C-Zero (in early 2011, £519.16 to £415 for a month lease-only), Vauxhall Ampera (in early 2012, £33,995 to £28,995), Toyota Prius Plug-in HEV (in early 2012, £31,000 to £26,000), and Chevrolet Volt (in early 2012, £30,000 to £25,000). The UK government has ambitious targets to reduce emissions and wants to lead ultra-low carbon vehicles production.

Currently the automotive sector is the largest manufacturing industry in Australia with a contribution of nearly 1% of Australian gross domestic product (GDP), providing around 50,000 jobs and producing 320,000 vehicles annually (Federal Chamber of
Automotive Industries 2009, Commonwealth Department of Innovation Industry Science and Research 2010). On 31 October 2010, there were around 16 million vehicles registered in Australia, compared with 14.4 million registered vehicles on 31 October 2006. This added a value of A$5.8 billion annually to the economy including A$3.1 billion in exports and component sales in the automotive industry (Australian Bureau of Statistics 2010). In 2010, the Cooperative Research Centre for Advanced Automotive Technology (Auto CRC), produced a vision report covering alternative technologies and reviewing other strategies to reduce carbon emissions by 80% beyond 2020 (Automotive commonwealth research council 2010). The Victorian Electric Vehicle Trial, from the Department of Transport, Victorian Government is an initiative to reduce environmental impacts by new technology adoption (Department of Transport Victorian Government November 2012). The department has allocated EVs for trials and also established charging stations throughout Melbourne to study broader understanding adoption of EVs in the Victoria. The key findings and recommendations from these studies include the following:

- Transport makes up to 16% of Victoria’s emissions and EVs with a renewable energy source can reduce emissions by around 50% over the vehicle life time (including replacement battery). The report highlights the importance of the government role in tightening the vehicle emission standards to benefit the EV industry and improve sales.

- The full fuel cycle emission (gCO$_2$/km) includes the effect of GHG emissions from production, distribution and use of the vehicle and the electricity. The source of the electricity used to power EVs is a key issue in Victoria. Coal is main source for electricity production in Victoria. As a consequence the emissions from electricity production are amongst the highest in the world. With a current Victorian electricity average full fuel cycle emissions of the ICE is160gCO$_2$/km and the EV is 225gCO$_2$/km.

- The department forecasts an improvement in Victoria’s emissions by electricity production by mixing the coal powered with renewable energy sources. In the forecast the department also assumes 2.5% to 4.5% improvement to efficiency by the improved vehicle technology. Based on these forecasts the expected breakeven date for the full fuel cycle emissions from EV and ICE is around 2024. During the period if the energy conversion efficiency of EVs improve (at
double the rate of ICEs i.e. 4.5%), this would bring the breakeven point forward to 2018. Conversely, if ICEs improve (at double the rate of EVs i.e. 4.5%), the breakeven date would be sometime after 2030. Thus favouring EVs over ICEs for reduction in emissions.

- The report recommended recycling the rare earth elements of batteries and motors in the long term to reduce the environmental impact of the vehicle production. Also the report vouches for the EV suitability for urban driving and highlights total benefits of reducing GHG emissions when used with a renewable energy source.

- Noise and electromagnetic fields are amongst several health and amenity concerns. The studies show no difference between EVs and ICEs, with a negligible risk to human health due to electromagnetic fields. EVs provide additional benefits with near silent operation, resulting in traffic noise reduction.

### 1.2.3 Commercial viability

Historically fossil fuels have fulfilled human needs for primary energy supply for thousands of years. Traditionally mankind has reaped the benefits and transitioned primary energy source usage from solid (coal and wood) to liquid (petrol and diesel) and then to gaseous fuel (natural gas) to achieve efficiency. Since 1870, with the development of electric motors and generators, electricity has become a vital secondary energy source. Primary energy sources, such as fossil fuels have been used to develop this secondary energy source.

Current conventional fossil fuel reserves are estimated at 2000 billion barrels with a daily global consumption of 71.7 million barrels (Asif and Muneer 2007). The USA, the UK, the China, the India and the Russia use half the global energy budget. These crucial energy economies are heavy importers of fuels to sustain their energy needs. Energy security concerns have encouraged these nations to consider building alternative renewable energy sources and technologies (Robert L. Hirsch, Roger Bezdek et al. 2005, Whipple 2011). The fossil fuel dearth has also led to a rise in global fuel prices. A reflection of these is seen in Australian context with an upward trend in fuel prices as shown in Figure 1.2, per gallon using cents per litre (cpl), from July 2007 to September 2011 (Australian Competition and Consumer Commission 2011). On the other hand the
renewable energy resource produces the clean and sustainable environmental energy capacity which supports zero emission vehicles.

The commercialisation of EVs was seen as impractical a decade ago, but in light of the recent energy crisis and fossil fuel prices (the recent price of petrol was A$1.30 per litre), EVs are now seen as commercially viable. High torque, zero emissions, quietness, smooth operation, and low maintenance are some of the advantages of EVs. This intrinsic nature of EVs has led to the commercialisation of EVs by OEMs globally, e.g., Mitsubishi i-MiEV, Nissan LEAF, General Motors (GM) Volt and Toyota Prius.

![Figure 1.2: Daily retail petrol prices, 2007 to 2011—Australian cents per litre, Source: (Australian Competition and Consumer Commission 2011)]

The commercial viability of EVs is also dependent on comparison to established ICE vehicles. The key concerns for commercial viability of EVs are:

1) Price (based on life cycle costs)
2) Value (based on reliability, safety, comfort, convenience and emotive)
3) Risks (technological such as energy storage)

Economic comparisons amongst four vehicles, a conventional ICE (Toyota-Corolla), an EV (Toyota- Rav4), a HEV (Toyota- Prius), and a FCEV (Honda-CFX) were conducted (Granovskii, Dincer et al. 2006). The fuel price (based on fuel price from 1994-2004 Energy Information Administration, USA) for EVs was US$0.90, compared to ICEs
where costs were US$2.94 per 100km driving (with 45% highway and 55% city driving). Granovaskii et al., conclude that an EV with a renewable energy source is more energy efficient than HEV or ICE cars. Though EVs are energy efficient, the average price of an EV is still an issue. The life cycle assessment cost of the vehicle use is not clear to consumers. The average prices of EVs were not seen as comparable to similar powered ICES; however with technology advancements and mass production, a notable shift has been seen in recent years. Examples are the Honda Jazz (HEV) at A$22600, the Toyota Prius-C (HEV) at A$25000 and the EDAY (EV) at A$25000 (based on prices from various dealers in Melbourne). The environmental value offered by EVs is greater when compared to ICES; however there are still issues with reliability, safety and convenience to customers.

The main drawback for EVs is its limited range and longer recharging times when compared to ICES (refuelling with fossil fuels). The studies conducted reveal that consumers want vehicles to be fully capable, with more than 150km in one charge. The main handicap is the charging times and the energy density of batteries. Currently the consumers are accustomed to 5 to 10 minutes of refuelling. Hence energy storage is vital for the efficient distribution of energy sources. Compared to the ICE, the reliability factor for the EV is lower, due to low density of energy storage. Flexibility in relation to energy storage is also an issue with long charging time. Liu, et.al discussed the advancement in energy storage systems with high efficiency (e.g., Lithium iron phosphate (LiFePO₄) batteries) for EVs due to increasing concern about sustainable development (Liu, Li et al. 2010). Liu et.al reported that advanced energy storage materials, with high energy, power density, environmental friendliness, convenient and flexible storage of energy to power EVs and HEVs, is under progress. Using principles of energy storage and smart material design approaches, advanced energy storage materials (with excellent energy density) are being developed. Their practical applications will play vital roles in solving serious energy and environment problems (e.g., the EV energy storage system).

1.2.4 New opportunities

EVs were popular in the early 1900s as shown in Figure 1.3, but did not find widespread acceptance due to unreliability of the vehicle at that period compared to
ICEs (Graeme P. Maxton and John Wormald November 2004). The main reasons for EVs failure in 1900s were:

1) Range limitation of the EV
2) Abundant fossil fuel at the time (leading to cheaper prices) and easy refills for ICEs
3) Low vehicle price resulting from mass production of ICEs
4) Reliability of ICEs over EVs

Conventional ICEs have been established for a long time and hold a major market share to date with high power and range. However, their impact on the environment and a scarcity of fossil fuels are leading to challenges and an entire set of solutions and new technologies are being developed to address these 21st century challenges. The portfolio of technologies and potential solutions in the automotive industry is being broadened again. This has created an opportunity to look at different vehicles that utilise renewable energy sources, efficient transmissions, and novel materials. Price, scarcity and environmental impact of fossil fuels have created a paradigm shift in the automotive industry, resulting in new technology development (Ogden, Williams et al. 2004). Examples of such vehicles include: the GM Volt (HEV with generator to convert fossil fuel into electricity), the Toyota Prius (HEV), the Honda CFX (FCEV), the Nissan LEAF (EV) and the Mitsubishi i-Miev (EV). New opportunities in automotive industry are created in three areas, i) the manufacturing industry, ii) the infrastructure industry and iii) the fleets/maintenance support industry.

Figure 1.3: Electric vehicle sales decline during early 1900, Source: (Graeme P. Maxton and John Wormald November 2004)
The opportunities in manufacturing industries related to EVs are:

1) **Drivetrain**- a) High power density and efficiency motor, b) Motor configuration – series (trans-axle, wheel motor) or parallel (motor before or after transmission). Power trains for the HEV and the EV required new components needing a different design strategy. The challenges posed to auto companies have led to innovative design solutions. This innovation is creating opportunities for suppliers. e.g., Michelin (Michelin 2008) and Siemens (Broge 2006) with novel drivetrain products (these are further explained in chapter 2).

2) **Energy storage**- a) Electro chemical- LiFePO₄ battery, and lead acid, b) Ultra capacitor-to boost the vehicle performance and c) Mechanical battery- fly wheel or compressed air. Battery technology is developing at a rapid rate and the forecast of the LiFePO₄ batteries market is expected to reach ~US$9 billion by 2015 (Roland Berger Strategy Consultants 2011). Australian based battery technology company called “Very Small Particle Company” established in 1999, focused on developing LiFePO₄ batteries for EVs (Very Small Particle Company Ltd. 2013).

3) **Vehicle handling**- a) Sensors and controllers, b) Low resistance tyres, and c) Improved suspension design. An example of this is the Bridgestone Ecopia which is targeted to reduce fuel consumption and is suitable for EVs (Bridgestone Americas Tire Operations 2013).

4) **Light weighting**- a) Materials- Carbon fibre-composites, Aluminium and Magnesium alloys, honey combs, and composites, b) Joining and attachment-riveting, screwing, adhesives and bolting of composites. The Australian based company, Carbon Revolution claims to have brought the world’s first one-piece carbon-fibre wheel to market, known as the CR9 (Carbon Revolution 2011). The wheel is 40- 50% lighter than a conventional alloy wheel and it maintains full strength as required by OEM standards.

Another subset of this process is the development of infrastructure and energy management for EVs and HEVs.

1) **Charging infrastructure**- a) inductive and conductive charging stations, b) charging fleets (for energy supply), and c) charging station deployment (commercial and personal use). The development of charging infrastructure is
essential if EVs have to replace ICEs. Charging stations offered by Better place, Charge point and Blink are some examples in Australia. Better place offers both driver and fleet memberships within Australia for charging EVs (Better Place 2011).

2) Energy management- a) Energy generators including renewable energy, b) Energy distributors and transmitters. Current global energy needs are supplied by fossil fuels (80%), renewable (13.5%) and nuclear energies (6.5%) (Asif and Muneer 2007). According to Asif et. al the growth in the global energy demand will continue and renewable energies will play a vital role in meeting the demand. Apart from meeting specific energy needs, renewable energy sources also provide local manufacturing and job opportunities. An example of this is the Australian Solar Institute, which reported that nearly A$100 million was spent in 2011 by the Australian and State & Territory Governments on photovoltaic research (Watt, Robert Passey et al. May 2012).

These growth opportunities in EVs have led to growth and embellishment of associated industries. The maintenance and support industry will expand with growth in EVs. Examples of this are vehicle conversion markets, fleet cars, second hand car markets, maintenance and battery recycling plants. The Blade Electric Vehicle company is ingenious Australian company, which uses the Hyundai car to convert an ICE into an EV (Blade Electron May 2011). EV Power and EV works are two companies supplying EV accessories for maintenance of vehicles. The battery recycling and reuse industry is another growing support industry. According to the GM in a Volt, when the battery reaches the end of its life, only 30% or less has been used, providing a large potential for reuse applications (Isaac Leung 2011). Similar comments were made about high-performance LiFePO₄ batteries used by the Nissan LEAF. Energy Matters reports that 70-80% of residual capacity could be used for the off grid solar power application (Energy Matters Pty Ltd 2009).

1.3 Research problem

EVs will form an important part of the solution addressing the concerns previously outlined regarding carbon emissions, energy security and the ongoing need for the personal transportation. EV has the following advantages over the ICE:
1) Emissions- EV can result in zero tail pipe emissions when used with a renewable energy source. This also leads to reduced energy dependence on fossil fuels.

2) Energy conversion- The EV is approximately three times more efficient than the ICE for energy usage. This also includes the energy loss in transporting the fuel and motor/engine using the energy to run the car.

3) Torque- Electrical motors typically have high torques (e.g., the EV can run 0 to 60km within 4 seconds from Tesla Roadster)

4) Less maintenance- EVs have no oil refills required, reducing overall maintenance and are also are quite to operate as they have no mufflers

5) Simplified design- the EV has fewer moving parts than the ICE for a same sized car. It has no fuel tank, muffler, catalytic converter, and oil filters. This simplifies the design.

The key technical challenges of the EV are:

1) Drivetrains (electric motors for increased power density)

2) Energy storage (increase energy density of the battery)

3) Infrastructure (fast chargers to overcome delayed charging times)

4) Vehicle mass (reduce overall mass to increase range)

Amongst challenges listed above this research work is focused on packaging a novel high power density electric motor for a small car. The thesis initial chapters covered development of an in-wheel drivetrain for a small car and the rest of the chapters discussed implications of these changes on the vehicle design.

1.4 Aim and objectives of research

The research aims to develop a high power density motor packaging and fitment to a small car. The research objectives are:

- Design a suitable high power density motor for a small car. A high power density motor is achieved by mechanically stabilising the air gap in the motor magnetic path and maximising the space utilisation for the magnetic path.

- Develop a new brake system in accordance with Australian Design Rules (ADRs) and relocate the brake (due to housing of drivetrain inside a wheel).

- Choose a small car for the developed in-wheel drivetrain and conduct requisite studies on vehicle rides and suspension performance for the EV.
• Explore the performance sheet for weight distribution (variation in centre of gravity) and range for a developed EV.

1.5 **Scope of the research**

Motor design is done in two stages. First the magnetic path is designed to provide number of poles and stator/rotor dimensions. The next stage is designing the physical stator, the rotor, the insulation, and the copper winding to package into a small unit. The motor packaging is optimised for: i) deflection to stabilise the crucial air gap, ii) minimum mass without compromising structural rigidity, iii) space utilisation without affecting tolerances/clearances, and iv) thermal stability of motor. The current work is based on a magnetic path supplied by the CSIRO (Auto CRC C2-25 EV drivetrain program participated by CSIRO, Swinburne University, VPAC and Latrobe University). The motor electronic controller is designed by Latrobe University and is not part of this research work. The drivetrain work done in this PhD includes: i) geometry optimisations of magnetic path (stator/rotor) and other motor parts (haft, hub, motor covers), ii) mechanical performance, and iii) the vehicle fitment studies.

1.6 **Overview of methodological approach and outline**

The thesis is organised in ten chapters, chapters 2, 4, and 5 cover the development of the high power density drivetrain. The chapters 3, 6, and 7 discuss small car fitment studies with a conceived drivetrain. The chapters are arranged as follows:

• Chapter 1 includes the background of research, and review of the research problem. The chapter covers basic details of establishing growing importance of EVs and highlights key challenges of EVs. Amongst the range of possible solutions to these challenges, the research focuses on developing a high power density drivetrain for a small car.

• Chapter two covers the study of: i) the existing drivetrains, ii) the IEEE 11-2000 standard and iii) the existing motors, to establish the drivetrain configuration and motor selection. Initially, detailed analysis of the state of the art in commercial EV drivetrains and an extensive literature review on motors were conducted. These studies were carried out in order to fully comprehend relevant technologies, and to use these findings to select an appropriate drivetrain configuration and motor. An appropriate drivetrain configuration was selected based on the following objectives:
i) minimal power transmission loss (maximising the efficiency), ii) independent control of wheels, and iii) simple design. The motor selection was based on the following objectives: i) power density, ii) weight, iii) cost, iv) maintenance, v) size, and vi) torque/speed. After comparisons, the in-wheel (due to increased energy efficiency), switch reluctance motor (SRM) was selected.

- Chapter three covers vehicle selection and an appropriate rim-tyre to fit inside the selected vehicle. The vehicle selection was based on: i) vehicle mass-to-power ratios, ii) dimensional requirements, and iii) fitment studies of two identified car sizes (small, and medium). The vehicle specifications and dimensions were obtained from vehicle dealers. The vehicle selection process involved: i) examining the vehicle mass and power output and ii) physical measurements and 3D digitisation (scanning) for appropriate clearances of selected tyre with the car body. Based on this study the Holden Barina Spark was selected as the mule vehicle. The rim inner determined the motor envelope. The rim topology was designed for a compliance to “Rims and Tyre standards-Australia” to fit inside the Holden Barina Spark. It is then optimised for a low weight by comparing different rim topologies for deflection. Tyre selection for in-wheel motor was based on two main objectives: i) clearance with mudguards, and ii) low rolling resistance.

- Chapter four discusses two aspects of the motor: i) concept design and ii) detailed design. Two conceptual proposals, horizontal and vertical (based on magnetic path orientation), were conceived using CSIRO magnetic path (rotor/stator) sketches. These concepts were evaluated using virtual reality (VR) tools (schematic sketching methods), based on the following objectives: i) power density, ii) optimised magnetic path, iii) ease of assembly, detachment and manufacturability. The detailed SRM design consisted of two motor covers housing two rotors and a central rotor, positioned in the middle. The stators were seated on the hollow shaft and acted as non-rotating parts within the overall motor assembly. The shaft, motor covers and overall motor assembly were optimised using finite element method (FEM) to minimise deflections and stabilise the crucial 1mm air gap. The motor hub consisted of two bearings separating non-rotating (stators) and rotating parts (rotors). As per IEEE 11-2000 standard, the motor hub bearings were selected based on: i) the optimisation of bearing load distribution (up thrust and low thrust handling), ii) maximum life cycle, and iii) minimum vibrations and noise levels.
Finally, the motor assembly was investigated for an appropriate primary holder required to handle: i) self-release due to desired reversing of the motor typical for EVs, and ii) thrust developed within the motor during operations.

- Chapter five discusses mechanical optimisation of the motor components. Motor covers weight was minimised using FEM without compromising structural rigidity. The motor power density is also dependent on space utilisation, hence VR and augmented reality (AR) based optimisation was conducted to: i) maximise space utilisation within the wheel envelope and iii) ease of assembly and fitment. Thermal stability of the motor was established by designing appropriate forced fan cooling (retaining motor temperatures to class B, IEEE 11-2000 standard) and insulation (Class H grade, IEEE 11-2000 standard).

- In chapter six, a brake system was designed in accordance with ADR 31, 33 and 35 standards. A disc brake was selected amongst compared brake categories based on cost, weight, space required, ease of maintenance, power requirement of the actuators, and thermal/mechanical performance. Experimental tests were conducted on existing vehicles to determine braking force and temperature requirements for disc brake topology optimisation. Using the experimental test results as an input, FEM simulations were used to optimise disc brake topology for structural and thermal performance. The caliper was designed to fit within the vehicle and suspension envelope.

- Chapter seven discusses the evaluation of EV suspension based on two key aspects: i) vehicle ride and ii) fatigue failure. Quarter-car models were used for comparing vehicle rides of an in-wheel Holden Barina Spark (EV) and an ICE Holden Barina Spark (ICE). In order to accomplish these tasks, mathematical and computational simulations were devised using the MATLAB® and Simscape® programs. The simulations indicated a rough ride for the in-wheel Holden Barina Spark. Further Bode plot analysis was conducted, indicating that both EV and ICE performed within the safe frequency range. Additional studies investigated fatigue failure by examining suspension, whereby variable amplitude events generated experimentally were used as inputs to create fatigue loads. A FEM study was conducted using experimental inputs (fatigue loads) on a high stress suspension lug. The results assessed life cycle with a 3D rain flow matrix and based on Palmgren-Miner rule.
application in time domains, it was concluded that the suspension system of the in-wheel Holden Barina Spark was completely safe.

- Chapter eight examines: i) fitment of the EV ancillaries ii) the vehicle range; iii) the vehicle mass distribution; and iv) longitudinal slip and tyre change evaluations. The appropriate space/clearances for fitment of the EV ancillaries (e.g., batteries, controller, and wiring harness) and drivetrain were established. Using EPA standards drive ranges of 225km (urban drive) and 167km (high way drive) for 72 cells battery pack was calculated. The vehicle mass distribution and the centre of gravity (CG) variations determine the stability of the vehicle. Hence the vehicle mass distribution and CG variations were examined and found to be: i) 6% in longitudinal direction, ii) 0.5% in lateral direction and iii) 43% towards the ground. The CG variations in lateral and longitudinal directions did not substantially affect the vehicle ride, whilst ground variations made the EV more stable than the ICE. Comparison of EV and ICE tyre slip and longitudinal force was developed and concluded small percentile variations. Easy tyre removal during servicing of an in-wheel Holden Barina Spark was established using VR/AR based tools.

- Chapter nine summarises: i) key conclusions ii) key recommendations, iii) research novelties, and iv) evaluation of key research out comes.

- Chapter ten discusses the future scope of this research in three areas: i) EV drivetrains for potential use in similar related sectors (e.g., golf buggies and trucks), ii) contextualisation of the structured design methodology developed here in similar automotive projects and iii) EV fitment and essential studies on wheel, brakes, and suspensions which could be extended to other motors using in-wheel configuration.
Chapter 2

Selection of an EV drivetrain

2.1 Chapter overview

A detailed study of current commercial drivetrains and literature review on state of art in motors was undertaken. Using the findings from the study a novel motor design was conceived. The main focus areas of this chapter were: i) placement of motor within the vehicle (drivetrain configuration) and ii) motor selection for the drivetrain. This chapter covers the following:

- An extensive survey of EV drivetrains with a focus on the type and the design of motor used by current EV manufacturers (OEMs) was conducted to fully comprehend and discuss relevant technologies, and to use these findings to further develop the drivetrain. This study was focussed on establishing key details of current EVs: i) drivetrain configuration, ii) motor selected for EVs, iii) motor power outputs, iv) range, v) car size and vi) other design considerations (bearings, packaging of stator/rotors and motor enclosures).

- Based on the information from earlier studies on existing conventional and in-wheel drivetrains for current EVs, this study compared three configurations. The three different drivetrain configurations compared were: i) conventional (also called as indirect drive or parallel system), ii) in-wheel (also called as direct drive or series system), and iii) by-wheel (also called as direct drive or series system). An appropriate drivetrain configuration was selected based on the following objectives: i) minimal power transmission loses (maximising the motor power output), ii) independent wheel control, and iii) simple design.

- The second section in this chapter outlines motor selection for the chosen drivetrain configuration. It consolidates evaluations of earlier studies conducted on: i) EV drives/drivetrains, ii) IEEE11-2000 standard and iii) existing motor designs. The key aspects of earlier studies are discussed and the following motors are compared: i) brushed direct current motor (BDCM), ii) SRM, iii) permanent magnet motor (PMM), and iv) induction motor (IM). The motor selection was based on the following objectives: i) high power density, ii) low weight, iii) low cost, iv) low maintenance, v) small size, and vi) high torque/speed.
2.2 Drivetrain selection

In this section drivetrain configuration is defined and its three classifications are discussed. The EV drivetrain consists of an electric motor and mechanical transmission. The mounting of the motor inside the vehicle is called configuration and is discussed in this section. EV drivetrains are typically classified according to the position of the motor in relation to the wheel as: i) indirect or parallel drive (conventional) and ii) direct or series drive (by-wheel and in-wheel). The motor configuration also determines the number of motors used in the drivetrain. The following section describes three drivetrain configurations typically used in EVs and the selection of an appropriate drivetrain configuration for this research.

2.2.1 Conventional drivetrain

Conventional drivetrain (indirect drivetrain or parallel or on-board configuration) is the simplest form of EV drivetrain, which usually replaces the ICE by an electric motor. This configuration is offered as EV/HEV and is the most popular in the current commercial markets. The motor is centrally located and the transmission of the power is done through drive shaft and axles. This configuration needs the reduction gears (gear train) to manage the required speed characteristics. These are offered as front/rear wheel drive options. Figure 2.1 shows a conventional drivetrain system, where the electric motor is connected to wheels through a drive axle (transmission). Typically, the electric motor is mounted in the front/rear bonnet, similar to an ICE and the power is transmitted to both the wheels (He H. W., Sun F. C. et al. 1999). One of the disadvantages of the conventional drivetrain systems is the extent of the power losses during the transmission of power to the wheels (requiring gears, axles, stubs and transmission parts) (Caricchi, Crescimbini et al. 1996, M. Ehsani; K., M. et al. 1997, Xue, Cheng et al. 2008).

Figure 2.1: Conventional drivetrain
2.2.2 In-wheel drivetrain

The in-wheel drivetrain configuration is a typical direct drive, where the motor is directly attached in the wheel as shown in Figure 2.2. The absence of axles and gear trains makes it more efficient (low transmission losses) than the conventional drivetrain configuration discussed above (King, Jet Tseng et al. 1997, Cakir 2006). This drivetrain configuration allows independent wheel control enhancements similar to dynamic stability control, but without the vehicle acceleration reduction (e.g., cornering on high rolling resistance roads). This simplifies the design, reducing weight, cost and space requirements. Regenerative braking, antilock brake system, and stability have been incorporated more effectively as each wheel is capable of independent operation (Cakir and Sabanovic 2006). This also allows batteries to be located inside the front/rear bay area with a simplified weight distribution solution.

The in-wheel drivetrain configuration provides specific advantages: i) low transmission losses, thus maximum power output (increased energy efficiency absence of gearboxes, differentials, drive shafts and axles), ii) independent control of wheels (for speed, torque and braking requirements), and iii) simplified design (as a result of removal of mechanical systems, the component numbers were reduced compared to the ICE). The engine, gear box, transmission, fuel tank, muffler, silencer, and fuel exhaust became redundant. The motors are connected directly to wheels and batteries are housed in engine bay; as a consequence the design is simplified with management of these two parts reducing the overall weight and associated costs. Other notable advantages are i) increased ground clearance (due to removal of gear boxes and drive shafts), and ii) freedom to package batteries inside the vehicle front bay (normally used by the motor in a conventional drivetrain), as consequence improved weight distribution and ride quality. Some main disadvantages of this configuration are: i) weight of the motor on the wheel adds unsprung mass on the suspension system and result in the compromised vehicle ride conditions, and ii) packaging space within the wheel is occupied by a motor displacing conventional brake system. As a result a novel brake system redesign is required or existing brake system needs relocation if the space is available elsewhere within the vehicle.
2.2.3 By-wheel drivetrain

By-wheel drivetrain is a type of direct drive configuration, where two/four motors are installed, by the side of the driven wheels, each working independently to supply the power to drive. Generally this configuration needs stub axles for transmission of energy. Typically, this system suffers less transmission loss, compared to a conventional drivetrain, as the motors are fitted adjacent to the corresponding wheels, as shown in Figure 2.3. This configuration provides several advantages when compared to an in-wheel drivetrain: i) more space for motor packaging and ii) the drivetrain directly mounted to the chassis is open to air flow and results in better heat convection from the motor. Some main disadvantages when compared to in-wheel drivetrain are: i) more transmission losses (due to stub axles), ii) since the drivetrain is directly mounted to the chassis, it needs high performance seals for drivetrain enclosures to avoid dust/water entry.

2.2.4 Drivetrain configuration selection

In this work, the drivetrain configuration selection was based on the key criteria: i) minimal power transmission loses (maximising the motor power output), ii) independent control of wheels, and iii) simple design—as these are the key
prerequisites for a success of the EV drivetrain. Table 2.1 is a tabulated comparison of conventional, in-wheel and by-wheel drivetrain configurations based on these key criteria. The conventional drivetrain configuration has high transmission losses, independent control is not possible and as a result the overall design is complex. The by-wheel has medium transmission losses, independent control is possible and as a result the design is moderately complex. The in-wheel has low transmission losses, with independent wheel control possible and as a result design is simple. Other potential benefits with in-wheel design are: i) retrofit- it can be retrofitted easily hence can be used as standalone product for the current ICE conversions, ii) modular- the in-wheel concept allows modularity which means that it can be scaled to different vehicle sizes, and iii) vehicle stability- as the motor is housed inside the wheel the available engine space can package other ancillaries, allowing effective weight distribution and vehicle stability. In this research in-wheel drivetrain configuration was selected due to the aforementioned advantages.

Table 2.1: Drivetrain comparisons

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Conventional</th>
<th>In-wheel</th>
<th>By-wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transmission losses</td>
<td>High</td>
<td>Low</td>
<td>Medium</td>
</tr>
<tr>
<td>Independent control of wheels</td>
<td>Not possible</td>
<td>Possible</td>
<td>Possible</td>
</tr>
<tr>
<td>Design</td>
<td>Complex</td>
<td>Simple</td>
<td>Moderate</td>
</tr>
</tbody>
</table>

2.3 EV drivetrains in commercial cars

This research is focussed on designing an in-wheel drivetrain, however for the sake of completeness a simple review of conventional and by-wheel drivetrains used by OEMs is provided in Appendix 1 and summarised in Table 2.2.

Table 2.2: Conventional and by-wheel drivetrain study

<table>
<thead>
<tr>
<th>Make</th>
<th>Type of motor</th>
<th>Power (kW)</th>
<th>Torque (Nm)</th>
<th>Battery capacity (kWh)/(V)</th>
<th>Range * (km)</th>
<th>Car size</th>
</tr>
</thead>
<tbody>
<tr>
<td>GM Volt 2011</td>
<td>PMM</td>
<td>111</td>
<td>368</td>
<td>390/16</td>
<td>80</td>
<td>Large car</td>
</tr>
<tr>
<td>Mitsubishi i-MiEV</td>
<td>PMM</td>
<td>47</td>
<td>180</td>
<td>330/16</td>
<td>130</td>
<td>Small car</td>
</tr>
<tr>
<td>Nissan LEAF</td>
<td>PMM</td>
<td>80</td>
<td>280</td>
<td>480/24</td>
<td>170</td>
<td>Medium car</td>
</tr>
<tr>
<td>Toyota Prius hatchback</td>
<td>PMM</td>
<td>50</td>
<td>105</td>
<td>273.6/4.4</td>
<td>23</td>
<td>Medium car</td>
</tr>
<tr>
<td>Ford Focus</td>
<td>PMM</td>
<td>107</td>
<td>250</td>
<td>240/23</td>
<td>122</td>
<td>Medium car</td>
</tr>
<tr>
<td>Honda FCEV Clarity</td>
<td>FCEV/</td>
<td>100</td>
<td>256</td>
<td>288/20</td>
<td>386</td>
<td>Medium car</td>
</tr>
</tbody>
</table>
A brief study was conducted on the current in-wheel drivetrains from OEMs. This study enabled mapping of current technology trends to streamline design considerations for the in-wheel drivetrain. Patents were claimed for in-wheel motor designs in early 1884, by Wellington Adams, as shown in Figure 2.4 (Adams 1884). However the first functional EV and HEV credit goes to Professor Ferdinand Porsche (Figure 2.5). He created the first in-wheel drive vehicle in history during the late 1800s and exhibited it in the Paris World Exhibition in 1900 (Porsche Biography, Porsche Asia Pacific Pty Ltd 2009). The Lohner-Porsche used two generators (1.85kW) and two electric in-wheel motors (2kW) with a top speed of 35km/h offering a range of 200km (Porsche Cars North America 2013). During the early 1900s, EVs became less popular due to technological advancements in ICEs. At that point in time, i) relative prices, ii) abundant fuel, iii) high performance, and iv) easy fuel refills of ICEs made them more successful.

Figure 2.4: First EV concept, Source: (Adams 1884)
Figure 2.5: First in-wheel electric motor used late 1800s, Source: (Porsche Asia Pacific Pty Ltd 2009)

In 2006, during the Paris Motor Show, tm4 Electrodynamic systems displayed its 15kW in-wheel motor (section view, Figure 2.6) used in the CITROEN C-Metisse concept (2012). This vehicle is a HEV that combined a diesel combustion engine in front, coupled with a tm4 in-wheel motor (direct drive) at each rear wheel. In 2006, tm4 also designed a centrally mounted electric motor that provides all wheel drive capabilities to the HEV. The key findings from this motor design are: i) high power density is achieved by minimising the effective air gap and appropriate magnetic path orientation, ii) high torque density is achieved by having rotors on the outer cover of the motor. This allows magnetic flux away from the axle and rotor placed on motor circumference as shown in Figure 2.6 (maximising torques with less material), and iii) thermal management is achieved by attaching the rotor to the motor outer cover and the stator to the cooling system as shown in Figure 2.6 (as the stator reaches high temperatures in the motor).

Figure 2.6: tm4 motors, Source: (tm4 Electrodynamic Systems 2012)
Michelin developed a novel in-wheel motor concept (as shown in Figure 2.7) presented at the Paris motor show 2008. It has an ingenious arrangement of a geared motor housed inside a wheel and an electric suspension system (2008). It has a total weight of 43kg on each wheel with 7kg PMM producing a power of 30kW (steady output) and 60kW (peak output). Michelin claims an increased range for EVs with low rolling resistance tyres and ride comfort enhancements with the inbuilt electric suspension.

![Electric motor, Shock absorber, Electrical suspension](image)

**Figure 2.7: Michelin in-wheel technology, Source: (Michelin 2008)**

Siemens VDO produced the eCorner concept with an electric motor integrated into the hub (clearly intended for use with a FC or series hybrid powertrain) (Broge 2006). The motor (*Figure 2.8*) produces approximately 150Nm torque, and used a steer-by-wire system. The eCorner claims three major breakthroughs with the design: i) hydraulic shock absorbers (also called an active suspension system) replace conventional wheel suspension, ii) steer by wire supersedes mechanical steering, and iii) an electronic wedge brake (EWB) (brake-by-wire) replaces the hydraulic brake.

![Siemens eCorner](image)

**Figure 2.8: Siemens eCorner, Source: (Broge 2006)**

The Heinzmann Company develops and manufactures a wide range of drivetrains, mainly disc/hub wheel motors (*Figure 2.9*) (Heinzmann GmbH & Co. KG 2012). They offer in-wheel BDCM with different power output ranging from 12-15kW. These
motors are used in the construction of special purpose battery-powered light EVs. The following are key findings from their motor design: i) the weight of an in-wheel motor is an issue and iii) vibrations are expected as a result of using two single row deep groove ball bearings between rotor and stators (Load distribution is missing).

![Image of Heinzmann electric hub motor](image1)

**Figure 2.9: Heinzmann electric hub motor, Source: (Heinzmann GmbH & Co. KG 2012)**

Assembled Product Corporation is marketing XTi Hub Motors, as shown in **Figure 2.10 (2012)**. They come in a wide variety of sizes for different drive applications. These motors are 12-36V Brush PM Bi-directional DC motors, inside 8” Solid Rubber or 12” pneumatic integrated tyres. The key ideas from these motor designs were: i) the rim was used as a motor envelope, ii) reduction gear was used for low speed applications and iii) a shock coupler was used to withstand vibrations within the motor.

![Image of XTi Hub motors](image2)

**Figure 2.10: XTi Hub motors, Source: (Assembled Products Corporation 2012)**
The Copenhagen wheel (Figure 2.11) turns a normal bike into an electric bike with regenerative and real-time environmental sensing capabilities (Senseable City Lab MIT December 2009). The Copenhagen Wheel differs from other electric bikes, as all the components are elegantly packaged into one hub. The key highlights of these designs are: i) the wheel harvests the energy input while braking/cycling and reuses it to charge batteries, ii) it uses sensors in the wheel for collecting the required information (air/noise pollutions, congestion and road conditions), and iii) it has a modular construction (as it is designed to retrofit any bike) with the batteries housed inside the wheel over the outer motor envelope as shown in Figure 2.11 (no external wiring or bulky battery packs).

e-Traction is a self-propelled electrically powered wheel (Figure 2.12), operating at more than 90% energy efficiency, delivering torque of 7,500Nm. Key learning from this design was: i) 180° electric or mechanical steering, ii) ride-height control and iii) shock dampening, iv) high power density due to an in-wheel drivetrain (e-Traction 2011).
Protean uses PML Flight Link developed 81kW (peak)/ 64kW (continuous) PMM as an in-wheel drivetrain as shown in Figure 2.13 (Protean Electric 2012). The important learning from this motor design was space utilisation within the wheel; the motor cover shape is profiled to follow the inside rim topology. This maximises space available for the magnetic path resulting in a high power density motor.

**Figure 2.13: Protean electric hub motor, Source: (Protean Electric 2012)**

These studies indicated that a considerable amount of interest has been taken in the development of in-wheel motors by various flagship companies. A conventional drivetrain configuration is currently being adopted by OEMs and an in-wheel drivetrain is still in the conceptual stage. A comparison in *Table 2.3* represents motor details for an in-wheel drivetrain.

**Table 2.3: In-wheel drivetrain study**

<table>
<thead>
<tr>
<th>Make</th>
<th>Motor</th>
<th>Power (kW)</th>
<th>Torque (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Michelin Active Wheel</td>
<td>PMM</td>
<td>60</td>
<td>58</td>
</tr>
<tr>
<td>Siemens eCorner</td>
<td>BDCM</td>
<td>N/A</td>
<td>150</td>
</tr>
<tr>
<td>Heinzmann</td>
<td>PMM</td>
<td>12-15</td>
<td>80</td>
</tr>
<tr>
<td>XTi Hub Motors</td>
<td>Brush PMM</td>
<td>N/A</td>
<td>91.6</td>
</tr>
<tr>
<td>Protean Drive</td>
<td>PMM</td>
<td>81</td>
<td>800 (peak) 500 (cont.)</td>
</tr>
</tbody>
</table>

The following are key design considerations from this study:

- An in-wheel drivetrain has minimal transmission losses, hence it is advantageous (e.g., e-Traction). Space utilisation for the in-wheel drivetrain maximises power density (e.g., Protean electric motor). To maximise the space for the motor, the rim outer can be used as a motor envelope (e.g., XTi Hub motor). Hence motor power density is directly proportional to wheel size (e.g.,
e-Traction). Weight of motor is an issue for an in-wheel drivetrain (e.g., Heinzmann motor).

- High power density is dependent on magnetic path orientation, air gap and thermal management of motors (e.g., tm4 motor/ Heinzmann motor). High power density is achieved by mounting rotors on a motor cover, hence magnetic path orientation is crucial (e.g., maximum diameter as explained in tm4 motor). Bearings play a vital role in managing air gap; hence appropriate bearings are essential for motor performance (e.g., Heinzmann motor). Thermal management of stators requires appropriate cooling (e.g., tm4 motor), this enhances the motor power density and the long life.

- Low rolling resistance tyres can enhance EV range and hence appropriate tyre selection is vital for EVs (e.g., Michelin).

The motor is the main component within the drivetrain. In the next section, the motor selection for an in-wheel drivetrain is discussed.

2.4 Motor selection

In this section key motor aspects for mechanical design considerations are outlined. Then main motor parts and classifications are discussed. Thereby four motor classifications are compared and appropriate motor selection was done for an in-wheel drivetrain.

2.4.1 Standards for motor design

This section outlines the key motor aspects within the standard for mechanical design considerations. The IEEE 11-2000 standard, titled “IEEE Standards for Rotating Electrical Machinery for Rail and Road Vehicles-2000”, covers study on vehicles propelled by greater than kw electric motors at i) 1200m sea altitude and ii) maximum shade temperature of 0 C (IEEE 2000). The key findings from these studies on the standards that are relevant to mechanical design considerations are: i) efficiency, and ii) thermal variations within motor.

The motors efficiency is a ratio of sum of outputs/ minus losses and input. To measure the efficiency standard uses the curve is plotted by the dynamometer, for speed and efficiency against motor current, considering losses due to the wheel and transmission.
As per the standard, the continuous rating in the speed between two directions shall not exceed 3% and declared characteristics shall not vary more than +5%.

The commutator in this case bearing, should result in a total run out within 0.05mm for speeds less than 44m/s or maximum design speed (25m/s in this case explained in the next section). Vibrations in any axis at any speed (as per standard, considering upper speed limit, 25m/s), should produce less than 7.6mm/s. Recommended mean sound pressure for a self-ventilated machine should be less than 150kW continuous output at a distance of 4.5metre less than 105dBA at or below maximum design speed.

The thermal variations (also referred to as temperature rise), consider three classes in the standard: i) continuous rating, ii) one hour rating, and iii) short time overload rating. For continuous rating the motors duty cycles are performed with the pulse control (until steady state is reached) to demonstrate temperature rise within calculated continuous ratings. The one hour test or continuous rise will be examined for grade stationery field windings can reach a maximum of 180 C (10C higher is allowed for total enclosed motors) and commutators are expected to reach a maximum of 120 C ve rload tests may result in an initial temperature of 100 Cand a final of 180 C e an temperature will be used and the method uses either i) thermometer, or ii) resistance or embedded detectors for measurement. H class was used for the insulation design and Class B for stationery field windings reaching a maximum of 110 C (10 C higher is allowed for total enclosed motors). Thus arrangement provided the overload and explosion protection for the motor.

The other standards covered in motor designs are IEC 60034-5 and IEC 60034-6 for degrees of protection and methods of cooling used on the motor (IEC 60034-6 ED2.0 1991-10-01, IEC 60034-5 Ed. 4.1 2006-11). IEC 60034-5 standard is used for complete protection against harmful dust and water from flooding when motor is running (IP 56M in this research). IEC 60034-6 or equivalent AS1359:106 1990 is used for cooling of surface cover (ribbing used for extra heat transfer), air is moved inside by stator rotations and external fan driven by its own electric motor (in this research IEC 416).

2.4.2 Motor metaphors

The configuration of electric motor is critical to achieving optimum power density, however this is often compromised the available packaging space, competitive costs of
materials and manufacturing methods. The electric motor consumes electric energy and converts it into electromagnetic energy for required torque (mechanical energy). Generally an electric motor has the following main parts:

- **Magnetic path-** Magnetic path in the motors consists of rotors and stators with an air gap. The stator is the stationary part with wound coils, which receives the power supply. The rotor delivers the mechanical rotation. When the stator is excited with power supply, the rotor rotates with a magnetic flux. The air gap in an electric motor plays a vital role in generating magnetic flux (in the mechanical sense torque). The stators/rotors use poles which are laminated silicon steel with shoes (larger cross section area) on the end to sustain magnetic flux. The rotor/stator use laminations stacked in a cylindrical pattern and called the armature core. The laminations are slotted to allow the copper windings to pass through. Fibrous wedges are used to shut these slots to avoid windings plying out due to high torque (due to rotational and electromagnetic forces) in the armature core.

- **Mechanical parts-** The shaft is a vital part of the motor which acts as a load bearing element within the motor, hence structural performance is important. Bearings are generally used in motors to segregate moving (rotors)/non-moving (stators) to confirm smooth operation and extend the motor life. The motor covers ensure structural rigidity and support rotor/stator assembly for functional performance. Motor covers also enclose the magnetic path and restrict the water/dust entering the motor. Other parts such as yoke, cage, and solid back iron are used to support the rotor/stator for structural rigidity. The stator, rotor, and holders fitments, structural rigidity and clearances ensure performance of motor. The cooling lines and insulations are used for thermal management within a motor.

- **Electrical parts-** Wound copper coils are required for current to produce the required magnetic flux. The wiring is also essential to ensure that the electric supply is achieved to the motor. Commutators are used to mechanically reverse current supply through brushes to the stators. In the absence of the commutator the stators are directly induced and sensors are used for sensing the rotor position.

However motors are broadly defined based on:
- **Power supply**- Based on electric energy supplied, mainly two primary types: i) alternating current (AC) motor (e.g., IM, PMM, and SRM) and ii) direct current (DC) motor (e.g., BDCM and PMM is used with DC). It is classified as a single phase or three phase depending on the power supply. A single phase motor does not provide a rotating field (magnetic flux), but it reverses polarity (generally at 180 degrees). To start the rotation it needs a capacitor or inductor in series with the stator windings to perform the start function. Rotation in a three phase motor is achieved by three phases located at 120 degrees to provide the magnetic flux. Three phase AC motors produce high power density and hence they are frequently used in EVs.

- **Slip**- This occurs when a stator rotational field (especially in a field weakening region) is somewhat faster than rotor speed (rotor current increasingly lags behind the rotating field) then the motor is called asynchronous (e.g., IM), motors otherwise referred to as synchronous motors (e.g., PMM and SRM). The synchronous motors rotor speed is the same speed as the stator rotational field (torque proportional to the amount of current). Synchronous motors are used in EV applications due to advancements in power electronics.

- **Magnetic material**- Magnetic flux is a force produced by magnets, required to run the motor. Magnetic flux generated in a motor is dependent on the coercive force of the magnets or materials used. There are two types of magnets used in the motors: i) PMs result in direct-excitation (e.g., PMM) and ii) electro magnets result in self-excitation (e.g., IM, and SRM). The PM uses naturally available rare earth elements and exhibits high magnetic flux; hence it is used in high power density motors. On the other hand electro magnets are produced using steel and copper windings to produce the required flux. This flux is not high due to resistance and thermal losses and results low power density.

- **Brushes and commutators**- Brushed motors generally use brushes and commutators. The brush makes contact with the commutator, a device designed to transfer the electric current to the stator. A commutator (mechanical rotary switch) reverses the direction of the electric current twice every cycle, to flow through the electromagnet, so as to generate the required flux for the outside rotor. Brushless motor uses PMs or phase of coils to activate/deactivate the
stator and the required magnetic flux is obtained. Generally brushless motors require the sensor to control the rotor position.

### 2.4.3 Motor selection criteria

The aim of this study is to select an appropriate motor for the in-wheel drivetrain. Based on summary of OEM drivetrain studies in the previous section the following developments can be noted: i) OEMs predominantly adopted PMM (due to high power density and compact size), ii) IMs on the other hand were used in low cost applications and iii) SRM is still evolving. Hence in this research more focus is on PMM and SRM as potential candidates specifically due to power density and efficiency. For completeness sake all the four motors are compared in order to choose an appropriate motor. The four main motor classifications compared for key desirables are: i) BDCM, ii) IM, iii) PMM, and iv) SRM. The key desirables of in-wheel motors for this research are:

- **High power density**- Maximised motor power output for a minimised weight is a key desirable. Hence, power density in this research also represents weight and motor efficiency. Magnetic path (geometry/material/orientation), winding (geometry/material/orientation) and air gap influence the power density of the motor. Motor efficiency is a percentage ratio of mechanical energy output (generally measured using shaft output) to electrical energy input (power supply). Efficiency also depends on motor losses and it is important to keep motor losses as low as possible. Typical motor losses are: i) electromagnetic (core) losses- due to material and stator/rotor losses (due to copper or $I^2R$ losses), ii) mechanical friction losses- due to bearing and frictions in the rotating parts, and iii) air losses- air friction losses due to forced fan cooling.

- **Low weight**- The high power density motor has maximum power output for minimised weight. Amongst various factors, weight minimisation depends on structural rigidity of material and thermal stability of motor. Additionally, motors typically have vibrations causing unwanted noises. These cause structural instability; hence the motor structural components ensure the rigidity to mechanically stabilise. Hence, an appropriate trade-off for weight and performance is vital for successful motor design.
- Low cost- The availability of raw material and the level of refinement required for its use in the magnetic path are significant factors for the motor cost. This material for the magnetic path is vital for two reasons: i) long sustained availability, ii) as a consequence, stable material cost. The magnetic path material is a high cost item that dictates total motor cost. Hence low cost sustainable materials for magnetic path are vital in this research.
- Low maintenance- A long motor life cycle is desirable for EVs; hence low maintenance of the motor is imperative. Thermal stability enhances bearing and insulation life and hence low maintenance. Avoiding frictions in the motor maximises the motor life and also results in the noise reduction.
- Smaller size- The smaller size motor should fit the available packaging space in the wheel envelope and this determines its suitability for the in-wheel motor.
- High speed and torque- The ideal characteristics of a vehicle are primarily dependent on the motor torque and speed. For an EV it is important to understand the motor performance in relation to different variables (e.g., EVs requirement of a high torque at low speed) (Nanda and Kar 2006).

2.4.4 Brush and induction motors

BDCM is a DC motor, which uses brushes and a commutator to transmit the power from the static power supply to change over the pole. Brushes are made up of copper or carbon and slide with the commutator using springs (mechanical rotary switch). The commutator is a cylindrical disc stacked with windings insulated with Mica. Brushes and commutators in DC motor add extra weight/size, need a frequent maintenance (due to friction) and as result reduce the motor life expectancy. BDCM uses a mechanical commutator and brushes, hence they are bulky in construction with low efficiency, low reliability, and high maintenance requirements (Xue, Cheng et al. 2008). BDCM uses steel and copper windings for the magnetic path. BDCM speed is controlled by varying armature windings current (field weakening), unlike AC motors (by varying frequency using inverter). The controller is simple and cost effective because a separately excited BDCM allows field weakening of DC motors. BDCM has the ability to produce high torque at low speed; however high speed is an issue due to friction of commutator and brushes. To conclude, BDCM has the following characteristics based on key desirables: i) power density- low due to bulky construction, ii) weight- bulky in construction, iii)
cost- low cost as controller design is simplified and it uses steel/copper for magnetic path, iii) maintenance- high as it requires frequent maintenance due to commutator and brush frictions, iv) size- large due to presence of mechanical commutator and brushes, and v) speed/torque- moderate torque and low speed due to friction of commutator and brushes.

The IM is an asynchronous motor that uses AC supply to the stators directly and rotor functions by induction. The stator north pole when subject to the rotor south pole causes the flux, hence rotation is achieved. However unlike other motors, in IM stator field lags behind the rotor speed, thus causing induction within the motor to produce required magnetic flux. Generally IM stators use a squirrel cage or wound copper coil arrangement. A squirrel cage is a cylindrical conductive ring connected through heavy bars (generally copper/aluminium bars are used). The wound coil arrangement uses the resistor in series to reduce rotor current at the start phase. The cylindrical rotor in the IM is typically constructed using laminations with skewed parallel slots for carrying the rotor conductors which are generally embedded. The IM offers low efficiency when compared to PMM and SRM (Boglietti, Cavagnino et al. 2005, Hashemnia and Asaei 2008, Nikam, Rallabandi et al. 2012). The IM uses steel, aluminium and copper for magnetic path and hence it is low priced when compared to PMM. EVs require the high break-away torque, determined by motor’s high power density and a low weight. Compared to the PMM and the SRM, the IM is inefficient with power density and weight (Chan 2002, Xue, Cheng et al. 2008). The IM has a simple construction, it has proved reliable, it has low maintenance, and it is cost-effective. Their cost advantage over PMM and size advantage over BDCM makes them suitable for low cost/low power EV application. One such example is the use of 13kW and 25kW IMs by the Mahindra Reva NXR model, EV manufacturer from India (Blog at world press July,2010).

The IM uses field orientation control (FOC, also referred to as vector control) to extend its constant power operation, typically required in EVs. The IM generally produces higher speeds and outputs compared to the BDCM due to absence of friction within brush and commutators. An inverter and controller are used to control the speed/torque of the IM with voltage and current frequency (field-weakening). As a consequence the motor needs fewer sensors (speed and rotor position controller). Speed variations of the IM are achieved by changing the frequency of voltage and especially high torque at
lower speed. This also implies that the torque/speed is dependent on the rated maximum output of the inverter. FOC is used to break the torque and achieve constant speed; hence the motor size is increased. Due to these reasons controller design is a challenge and as a consequence results in higher costs compared to the BDCM (Xue, Cheng et al. 2008). To conclude, the IM has the following characteristics based on key desirables: i) power density- low compared to SRM/PMM, ii) weight- high due to bulky construction, iii) cost- relatively lower than PMM, higher than BDCM (controller is not cost effective), iii) maintenance- moderate, iv) size- bigger compared to SRM/PMM, and v) speed/torque- moderate torque and speed control based on rated inverter.

### 2.4.5 Permanent magnet motor

The PMM typically uses naturally available PMs (made from rare earth elements) for a magnetic path that imparts strong magnetic fields (Lorentz forces). PMs have a high coercive force resulting in persistent magnetic field. This prevents demagnetisation and results in a high residual magnetic flux. The PMs are mounted on the rotor surface or embedded in the lamination stack. This higher magnetic flux of the PM compared to electromagnets results in higher power density. The efficiency of the PM is high due to: i) reduced rotor conductor losses (absence of rotor bars) and ii) low resistance in windings (no distribution winding, hence no phase shorts and shorted end turns reduces wastage). The PMM is more efficient, light weight, compact in construction, and reliable than the BDCM, IM and SRM. The PMM offers a high power density and a compact size hence it’s a choice for EVs (M. Ehsani; K., M. et al. 1997, Zeraoulia, Benbouzid et al. 2006). By late 2000 many researchers had recommended in-wheel PMM for EVs (Caricchi, Crescimbini et al. 1994, Terashima, Ashikaga et al. 1997). The comparative study on similarities between the PMM and SRM for fault tolerance application found that with the same temperature rise, the PMM produced 29% more torque density, hence Jack, Mecrow et al., recommended it as a good choice for the EV and HEV (1996). These motors produce high torque and speed. Due to its synchronous operating nature the motor speed is precisely controlled and hence the controller design is simple.

Based on magnetic path orientation inside the PMM they are classified as: i) radial flux permanent magnet (RFPM) and ii) axial flux permanent magnet (AFPM, also called surface mounted PMM). In an AFPM motor, the magnetic flux is generated along the
axial length of the shaft/motor. The RFPM motor magnetic flux is perpendicular to the axial length of the shaft/motor. The RFPM motors are traditionally used more in industrial applications. An AFPM motor offers high aspect ratios and rotor inertia, hence higher electromagnetic flux. Due to higher electromagnetic flux of the AFPM motor, it produces high power density and hence is suitable for EVs. An AFPM 25Nm torque at 700 rpm direct drive in-wheel motor with a crucial air gap of 1mm was manufactured and tested for EV application (Rahim, Ping et al. 2007). These studies used AFPM motor for EV application and recommended further motor performance optimisation by: i) reducing air gap, ii) effective PM orientation (magnetic path), and iii) appropriate winding dimensions. The transverse flux motor (TFM) is similar to the AFPM motor except it has a complex geometry and produces magnetic flux traverse to rotations. The use of the in-wheel TFM was discussed by Baserrah et.al. (2009), who compared different flux concentrated TFM s based on FEM for magnetic path optimisation. Based on these studies Basserah et. al., established maximum torque (by axial length reduction of motor) and mechanical stability (by using split winding structure).

A six-poles AFPM motor prototype was used in an electrical scooter, which had 45Nm peak torque, 6.8kg active materials weight, and was coupled directly to the scooter rear wheel (Caricchi, Crescimbini et al. 1994, Caricchi, Crescimbini et al. 1996). This in-wheel 25kW AFPM motor (maximum output) was developed with a speed of 176km/h (maximum by field weakening control), a range of 548km per charge (based on test bench study at speed of 40km/h) and sufficient acceleration to cover 400m from a standing start in 18seconds. A novel AFPM motor with Segmented Armature Torus (SAT), and a new laminated stator topology was investigated for efficiency of an in-wheel motor (Fei, Luk et al. 2008). This experimental study used a 6kW AFPM motor that demonstrated total traction of 55Nm at 80V supplies, with efficiency over 90% and overloading capability of 200-300%, endorsing its usage as an in-wheel motor.

The cogging/ripples of the functional motor result in non-uniform rotation with audible noise and vibrations. It occurs in functional motors due to: i) attraction between motor rotor (PMs) and stator and ii) interaction of the rotor (PMs) and the copper winding due to harmonics. The studies on reducing AFPM motor cogging/ripples concluded that using PM wedges inside stator topology resulted in increasing the motor efficiency by
3% (Rahman, Patel et al. 2006). An EV embodying a directly coupled AFPM motor was developed and patented (Patel N R, Hiti et al. 2009). The patent filed claims to address cogging and low phase induction (low efficiency of motor at high vehicle speeds). In the patent filed, cogging is minimised using short pitching and motor torque is maximised using PM wedge slotting (higher permeability than air) in the motor.

Generally the PMM operates relatively cooler and hence it is low maintenance and has increased life. In the electric motor every 10°C temperature rise reduces the bearing insulation life by half and every 10°C temperature reduction doubles the bearing/insulation life expectancy. However to enable low maintenance and increase bearing life, the motor is force cooled. Typically, three ways of cooling are opted for electric motors: i) natural cooling- this is based on surrounding environment and is not very effective, ii) air cooled- this is effective when a fan is used and iii) liquid cooled-this is very effective when using a coolant circulation. Cooling also plays a vital role in increasing the efficiency and power density of the motor. The liquid cooling results in effective heat extraction from the stators and the liquid cooled aluminium ring with high thermal conductive epoxy was used to ensure quick heat extraction from the stators (Rahman, Patel et al. 2006).

The PMM has the following characteristics based on key desirables: i) power density-high power density due to use of PMs. In some cases the PMM suffers with a cogging effect; PM wedges and short pitching through the controller are used to overcome these. ii) weight- low weight due to high power density of the PMs, iii) cost- high cost due to magnetic path (discussed in the next section), iv) maintenance- low maintenance due to relatively cooler operational temperatures; cooling (forced air/liquid) can aid quick heat extraction from stators. Due to thermal stability the PMM offers low maintenance and increased motor life, v) size- small size, due to high coercive force of PMs, hence it is suitable for the in-wheel motor, and vi) speed/torque- high speed and torque characteristics. In some cases low phase induction (high torque at low speeds for EVs) of PMM can be overcome with use of PM wedges.
2.4.6 Shortfalls of permanent magnet motor

The key disadvantage of the PMM is the need for PMs which use rare earth elements. These rare earth elements have limited supply sources, hence they are subject to wild price increases and supply uncertainties. The main rare earth elements used in PMs are Neodymium Iron Boron (NdFeB, aka ‘neo’ magnets), Sintered Samarium-Cobalt (SmCo), and Samarium-Iron Nitride (SmFeN). In 2012, total production of rare earth element was ~110,000 tons (85-90% was from China) and demand was ~120,000 tons (Suzanne Shaw and Arnold Magnetic Technologies November 2012). About 90% of the rare earth elements from China are mined in Baotou, inner Mongolia and in Sichuan province (Seaman 2010, Campbell August 2008). China has been implementing a strategy to conserve its natural rare earth resources. Such examples of its strategies in 2008 are the new tariffs on export of Neodymium (15% extra) and Dysprosium (25% extra), and 50% reduction in rare earth element exports from the previous year. There was a reduction in number of approved Chinese exporters from 41 in 2007 to only 23 in 2008. The Mountain Pass in California and Mount Weld in Western Australia are other areas explored to fulfill the supply gap for Neodymium (about 18%). Most of Oxide produced today is being used in PMs and the price increase is around 10% per year. As of 2010, the Neodymium metal is estimated at US$50/kg and the Dysprosium US$155/kg. Dysprosium is found in ionic clay from Jianzxi, Southern China, and is less abundant in comparison. Dysprosium provides retention of magnetic properties at high temperatures, a key requirement in the PMM. No substitution is currently available however light rare earth Neodymium is mixed with other elements resulting in an increased size to avoid demagnetisation (due to high temperatures requirements within the motor). This largely affects the power density of the motor. Another notable disadvantage of the rare earth magnets is the fragile nature and handling them in the manufacturing environment.

Due to scarcity and cost variations of rare earth elements, a research effort is underway in three areas: i) novel materials to replace rare earth elements ii) recycling of rare earth materials and iii) development of a novel motor without the use of rare earth elements. In this research an approach was taken to develop a high power density motor without using rare earth materials (PMs). In the next section, the SRM is discussed to address short falls of the PMM.
2.4.7 Switch reluctance motor

SRMs were invented in 1838, by Davidson from Scotland; however, the first patent was filed in 1970s by Bedford and Hoftfiled. At this point in time there is no evidence of a commercially used SRM for passenger cars. However SRMs are typically used in mining trucks for loaders and dozers (LeTourneau Technologies Inc. 2012). The SRM uses the electromagnetic flux to move the rotor from one stator pole to the other stator pole. The SRM achieves rotation by the sequential energising/de-energising of the stator poles. The rotation is archived by: i) energising the stator pole winding and the nearest rotor pole is attracted, ii) then this stator pole winding is switched off and the next corresponding stator pole winding is energised, and iii) the rotor now attracts to the corresponding energised stator pole winding. This process is repeated and the rotor will follow this sequence, attempting to align rotor poles with energised stator poles. However, as the rotor and stator poles align, the stator poles switch off and the next group of stator poles switch on, continuing the rotation of the rotor. Typically the SRM generates continuous movement by consecutively switching the currents on and off, thus ensuring the poles on the rotor are continually chasing the stator current. The movement achieved is a function and depends on: i) current flowing through the winding and ii) the characteristics of the iron in the rotor. The rotor uses sensors to feedback the control system with the exact position of each pole.

The SRM has marginally lower power density than the PMM and significantly better than the IM and the BDCM (K.M. Rahman and M. Ehsani 1996). The SRM operates on reluctance torque and does not use rare earth elements for magnetic path producing efficiency equivalent to the PMM (Xue, Cheng et al. 2008). However in the SRM, weight is an issue due to absence of magnetic materials. The newly developed outer-rotor-type multipolar SRM was fitted into a vehicle and the results demonstrated 34km/h speed (Goto H., Suzuki Y. et al. 2005). Small commercial 6-4 and 8-6 poles SRMs were simulated and obtained 40% more power output at rated voltage and current compared to IMs (Rahman, Fahimi et al. 2000). Xue, Cheng et al. recommended the SRM as the most suitable for the EV drivetrain due to: i) absence of windings and PMs in rotors ii) high speed operation and iii) high reliability and safety factors (Xue, Cheng et al. 2008).
Motors for EVs have been characterised by high torque at low speed, and low torque at high speed, which suits the initial acceleration and hill climbing needs and the cruising requirements for a general automotive vehicle (Chan 2002, Zeraoulia, Benbouzid et al. 2006, Xue, Cheng et al. 2008). The synchronous SRM generates electromagnetic torque solely on the principle of reluctance. Different SRM 6-4 integration models were presented and compared for dynamic torque effects on the motor (Giesselmann 1996). The SRM characterises high-starting torque, high-speed operation capability, and a wide constant power output, making it suitable for an EV motor. The torque in the SRM is typically generated by proper positioning of rotor iron with an excited stator iron pole. The magnetic status of the machine is controlled by flux linkages in each phase with respect to time and currents using sensors embedded in the motor. In the SRM pulse positioning is achieved by closed loop design of the controller. The controller uses position sensors, a speed sensor and a torque controller which communicates through an inverter for exact positioning. Self-inductance versus rotor profile is used to sense the aligned, midway and unaligned positions to feedback the controller. SRMs have better potential for EV applications because of their simple construction, low manufacturing cost, high starting torque, field weakening capability, and reasonable torque to weight ratio and efficiency (compared to PMMs) (Wadnerkar, Tulsiram et al. 2005).

The efficiency of the motor is also dependent on the thermal management of the SRM. The outer rotor and inner rotor motors were compared in terms of thermal, dynamic behaviour to evaluate their performance in the context of the EV (Hennen and De Doncker 2007). This study concluded that effective cooling of outer rotor motor maximised efficiency by 92% and resulted in higher torque of 12.8Nm/A. Typically SRM stators rise to high temperatures due to copper and eddy current losses. Hence forced cooling is required in this region and the relative thickness of motor cover and ribs plays a vital role. The outer rib thickness of the motor cover played vital role in handling heat dissipation due to copper and eddy losses of SRM (Srinivas and Arumugam 2005). Due to absence of PMs, the windings on the rotor make the SRM relatively easy to cool. As it does not use PMs, it is less sensitive to high temperatures.

Noise and ripple effects are some of the drawbacks of SRMs, which are addressed through having a mechanically stabilised air gap and using an appropriate digital signal controller (for short pitching). The tendency to increase power density within the SRM
increases the noise and vibrations. The noise and vibrations are important as these can affect the mechanical structure response. Noise from electrical machinery has three origins: i) mechanical, ii) aerodynamic, and iii) electromagnetic forces. The mechanical noises relate to speeds and vibrations as a result of friction. In particular, bearing design and selection plays vital role in minimising mechanically influenced noise. This is discussed in detail in chapter 4, where bearing selection is addressed. The aerodynamic noises arise from air turbulence caused by rotating parts (stator/rotor/fans). Generally forced cooling is used in SRMs and this needs an external fan. The fan, sometime acts as obstruction to the airflow resulting in unwanted noise. Hence optimal cooling fan selection with low noise is important for success of the SRM (discussed in chapter 4).

Electromagnetic forces within the motor produce vibrations on the stator and hence are the main reasons for noise. The electrical current flowing in the wound coils produces magnetic field acting on the stator iron and is governed by the air gap. These magnetic fields generate flux density harmonics calculated using radial Maxwell forces. The $F_M$ is the radial magnetic flux in the air gap of the magnetic path which attracts stator and rotor. Its amplitude per unit area is called force frequency ($F_M$) is given by Equation 2.1:

$$F_M = \frac{b^2}{2\mu_0}$$  \hspace{1cm} (2.1)

The $b$ is the flux density of the stator internal surface at a given point and $\mu_0$ is the permeability ($\pi \times 10^{-7}\text{H/m}$). Also the non-stationary components within the motor produce a force dependent on the deformation mode of the stator. Force $f_{mM}$ is used to determine component amplitude ($\text{N/m}^2$). In the case of the pole pair machine, the amplitude per unit area is defined by Equation 2.2:

$$F_M = \sum_m f_{mM} = \frac{(\sum h b_h)^2}{2\mu_0}$$  \hspace{1cm} (2.2)

Where $b_h$ the harmonic flux density at pole pair number $h$ and $m$ is the mode deformation. The human audible range is within 20-16000Hz; however 1000-5000Hz is the most sensitive to humans as it coincides with the range of human speech. Pressure waves of audible frequency are mainly caused due to magnetic amplitude resulting from mode number (representing maximum or minimum frequency bands), hence this needs to be minimised. For ideal conditions the $m=0$, the deformation is uniform and when
m=1, the deformation is maximum at one point and minimal at the other. As increments of the deformation mode precede the deformation increases and results in eccentricity causing vibrations. The eventual eccentricity of the rotors due to air gap variations and bearing life can lead to audible noises within the motor. However better control is sought through an appropriate controller design for pitching the power supply. The high torque ripples at low speed were mitigated using optimum current waveform or profiling the current and acoustic noises by reducing radial pull using 24-16 SRM (Rahman and Schulz 2002, Lovatt September 1997). The stiffness and vibration analysis was investigated for 6-4 SRM and concluded: i) safe performance (1.4 factor of safety when applied with twice full load ~32kgf/mm$^2$), and ii) 6µm harmonic analysis was negligible (Srinivas and Arumugam 2004). For the SRM from a mechanical design prospective it is important to: i) have an appropriate sturdy support for stators/rotors to reduce these vibrations to a certain level and ii) minimise the pressure waves of audible frequency, mainly low amplitude and low mode number by stabilising air gap within the magnetic path. These studies on SRM indicate that:

- **Power density-** The stator contains wound coils, no magnets (rare earth elements) or wound coils in rotors; compared to the PMM the SRM has a comparable power density. The appropriate air gaps play vital role in achieving the high power density and reducing the noise. Appropriate bearing/fan selection is vital to avoid the audible noise. Forced cooling and structurally sound motor covers for thermal stability enhances the efficiency of the motor.
- **Weight-** Moderate due to electromagnets as the SRM used the steel and copper as for the magnetic path. Appropriate mechanical packaging with the low weight motor cover is vital for weight management of the motor.
- **Cost-** Moderately low due to absence of PMs in the proposed SRM. The SRM typically used steel and copper for its magnetic path and as result it is cheaper compared to the PMM.
- **Maintenance-** Low, as it operates on: a) the reluctance torque without a need of commutators, b) uses non-corrosive silicon steel for magnetic path, and c) high reliability with no single point failures.
- **Size-** Comparatively moderate as it generates reluctance torque and absence of brushes. However when compared to PMM it is still higher due to use of electro magnets for the magnetic path.
• Speed and torque- Capable of very high speeds and producing high torques. As the SRM rotation is achieved by sequentially energising and de-energising the phases.

2.4.8 Motor selection for an EV drivetrain

In this research, an extensive technology study on motors/drivetrains on OEM cars was conducted and the key findings were used for conceiving an in-wheel drivetrain. The study of OEM cars also included motor design parameters and construction, assessed by studying different configurations and specifications of EVs. From an OEM cars study, it was evident that, at present, many automotive OEMs—including GM (Volt), Mitsubishi (i-MiEV), Ford (Focus), Tesla (Roadster), Nissan (LEAF) and Toyota (Prius)—have been conducting research on a series of production EVs, and predominantly use a PMM. Other automotive suppliers and stakeholders, for example, Siemens and Michelin, have also started developing in-wheel configuration using the PMM. The IM is used in low cost/low power application and the SRM is still in the conceptual/test bench stage. Most of the car companies used small to medium size cars for EVs with an EV range of ~150-180km. The magnetic path orientation (e.g., tm4 motors), space utilisation of motor (e.g., Protean electric hub motor) and bearing designs (e.g., XTi Hub motors) are important motor design considerations highlighted in these studies.

In the second stage, an extensive study was conducted on motors to select an appropriate motor for the in-wheel drivetrain. BDCM has low power density, large weight, high maintenance requirements, and large size, hence not been appropriate for the EV. The IM is inferior to the PMM and the SRM in terms of power density, weight, maintenance, size and speed/torque profile. The SRM is advantageous to the PMM due to absence of rare earth elements for the motor magnetic path. The PMM design is based on the use of high quality PMs, which are typically expensive and scarce (as they use rare earth elements found in the nature, such as Dysprosium). Consequently, attempts have been made to use low quality magnets (using Neodymium instead of Dysprosium), which failed to meet the current requirements. Generally, magnet performance has always been a concern with the PMM, particularly in an EV motor, where extreme ambient temperatures cause significant variation in magnet strengths. The SRM typically requires a smaller air-gap than the PMM, but the cost of the magnet more than compensates for this. Nonetheless, when both motor and inverter cost are
taken into consideration, the SRM appears to have an advantage over the PMM in EV applications. From commercial vehicle application point SRM has low hardware cost and high software, design and development costs. If the software, design and development costs are translated into strong intellectual property, it is cost effective to commercialise and distribute the technology. Based on the OEM and extensive literature study, the following comparison table has been generated in order to enable the selection of the most appropriate motor for the EV.

The EV drivetrain motor selection was based on the following objectives: i) power density, ii) weight, iii) cost, iv) maintenance, v) size and vi) speed/torque. Using a ranking range from low to very high each motor is ranked as low/small, medium/moderate, and high/large. The tabulated comparisons of four motors are depicted in Table 2.4.

Table 2.4: Comparisons table for four types of motor

<table>
<thead>
<tr>
<th>Criteria</th>
<th>BDCM</th>
<th>IM</th>
<th>PMM</th>
<th>SRM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power density</td>
<td>Low</td>
<td>Medium</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td>Weight</td>
<td>Large</td>
<td>Large</td>
<td>Small</td>
<td>Medium</td>
</tr>
<tr>
<td>Cost</td>
<td>Low</td>
<td>Moderate</td>
<td>High</td>
<td>Moderate</td>
</tr>
<tr>
<td>Maintenance</td>
<td>Very high</td>
<td>Moderate</td>
<td>Low</td>
<td>Low</td>
</tr>
<tr>
<td>Size</td>
<td>Large</td>
<td>Large</td>
<td>Small</td>
<td>Moderate</td>
</tr>
<tr>
<td>Speed/torque</td>
<td>Low</td>
<td>Moderate</td>
<td>High</td>
<td>High</td>
</tr>
</tbody>
</table>

The BDCM has low power density, very high weight, very high maintenance, large size, and low speed/torque profile. The IM has medium power density, large weight, moderate cost, and moderate maintenance, large size, and moderate speed/torque profile. The PMM has been the best motor choice in terms of its high power density, low weight, small size, and low maintenance requirements. The cost of the PMM is higher due to high price arising from demand of less abundant elements used in construction of the motor magnetic path. Nonetheless, high costs due to scarcity of rare earth elements are still important concerns for the PMM. Based on objectives, the SRM is suitable for meeting all the objectives. Hence, the SRM was selected as a suitable option for the in-wheel design.
2.5 Summary

In this chapter appropriate the drivetrain configuration and motor are selected. The research draws on key findings identified in the extensive literature review conducted as well as the results on the study of commercial car drivetrains, in order to develop a suitable drivetrain configuration and motor selection. The following is a summary of the chapter:

- This chapter provided a detailed literature review and a study of the state of the art on the EV drivetrain technologies utilised by many automotive manufacturers (OEMs). The key OEMs—General Motors, Ford, Nissan, Tesla and Mitsubishi—are currently in the process of developing EVs and drivetrain components, as the key aspects of the design. Based on the OEM cars study, it is evident that many have started developing the EV as an alternative technology using the conventional configuration and the PMM. Similarly in-wheel motor are being developed by Protean, TM4, etraction, Siemens ecorner, Copenhagen wheel, XTi hub motors, and Heinzmann. These studies on cars identified key details of EVs: i) small car size is used for EV application, ii) currently the in-wheel drivetrain is still in conceptual stage, and the most commercial drivetrains are conventional configuration iii) predominantly PMMs are used in EVs, iv) around 40-60kW power output is required for a small compact size car, v) the EV offer a range of ~150-180km using ~440-480kWh LiFePO₄ batteries at various voltages and vi) other details crucial for motor design considerations are bearings, packaging of stator/rotors, and motor covers.

- Drivetrain configurations are typically designed using schematics for three types of mounting arrangements—by-wheel, in-wheel, and conventional configurations. Among the different drivetrain configurations compared, efficiency was the key determining factor. The in-wheel motor was ideal due to its benefits, with low transmission losses and simple design (redundancy of gear box components). As an in-wheel configuration it provides independent wheel control with a maximum ground clearance (since gear related components became redundant).

- The IEEE11-2000 standard pin pointed to some key mechanical design considerations: i) efficiency- The continuous rating in the speed between two directions shall not exceed 3% o and declared characteristics should not vary more than +5%, ii) thermal stability is achieved by effective motor cooling for
maintaining the stator winding within 120°C and the bearings within 0°C, iii) vibrations in any axis at any speed (considering upper speed limit, 25m/s), should produce less than 7.6mm/s. Recommended mean sound pressure for a self-ventilated machine less than 150kW continuous output, at a distance of 4.5metre should be less than 105dBA at or below maximum design speed.

- The literature review revealed that, although the concept of the SRM is not new and is already found in many industrial electrical applications, PMMs are predominantly used in EVs as they are more efficient, compact and they provide high power density. The key disadvantage of the PMM is the need of PMs which use rare earth elements for the motor and this increases motor costs. The rare earth elements are mostly sourced from China. China is currently restricting exports and implementing a strategy to conserve its natural rare earth resources. As a consequence rare earth element prices are increasing. To overcome the cost and availability issues associated with the PMM, a new material or a motor (without the use of rare earth elements) is required to sustain the EV growth. The SRM, as an alternative, uses steel and copper as base materials and, when designed with suitable mechanical rigidity, provides optimal power density.

- In this chapter, for the motor selection, four different motors were compared, namely BDCM, SRM, PMM and IM. The SRM was selected based on the key objectives: i) power density- high and comparable to PMM, ii) weight- moderate, iii) cost- low with use of electromagnets, iv) maintenance- minimum (brushless structure), v) size- moderate and vi) speed/torque- high with wide power control region.
Chapter 3
Vehicle and wheel selection

3.1 Chapter overview
This chapter covers the selection of a vehicle and an appropriate rim-tyre (wheel) to fit inside the selected vehicle. The vehicle and wheel selection is important as the wheel governs the motor envelope and vehicle governs the wheel size. The following objectives guiding the research are presented here:

- The rim size governs the motor envelope. Rim size is dependent on the vehicle size; hence the appropriate vehicle selection is important. The small car was selected as the market acceptance for EVs is greater for city commuters using small cars. The potential hybridisation involved using medium cars. Also the required vehicle performance and the dynamics limit the motor to a specific maximum mass and range. The vehicle selection was based on field studies conducted at vehicle distributors for comparisons of two currently available vehicle sizes: i) small and ii) medium cars (i.e., A and B car segments) in the market. Different aspects were considered for the vehicle selection, with objectives of examining the: i) the vehicle mass-to-power ratios, and ii) mudguard clearances. The dimensions were measured and specifications were studied for the above objectives to finalise the vehicle and the wheel selections.

- Physical measurement had limitations regarding accuracy, referencing and dismantling of permanent joints. Hence, the selected car was digitised to overcome these limitations. The vehicle digitisation (scanning) investigated appropriate space availability and clearances for a car body (mudguard clearances) with rim-tyres and brake system. Based on this study an appropriate rim-tyre and brake envelopes were established.

- Rim topology was designed in compliance to “Rims and Tyre Standards-Australia 2010” to fit inside the selected vehicle. Five rims were designed to optimise the mass and appropriate motor space, focusing on the rim standards, size, shape and materials. The rim was optimised by comparing five designs for low weight using the thermal and the static analysis, as well as following the recommendations from the SAE J2560 standard for the fatigue life cycle assessment.
• Tyres are added onto the rim perimeter, as a link between roads and rims, providing required cushioning effect. Tyre selection was based on low rolling resistance, and material. The rim is part of the SRM and the surface temperature of the SRM will affect the tyre inflation pressure. The tyre inflation pressure has an effect on rolling resistance. Hence the relationship between the tyre inflation pressure and rolling resistance for an in-wheel SRM was established using SAE J670e empirical formula. A valve was selected and the rim-tyre prototype was developed to establish the dimensional and manufacturing variances.

3.2 Vehicle selection for an in-wheel SRM

As discussed earlier, motor size is directly proportional to its power output. The motor size is influenced by the rim size which largely depends on the vehicle size. Appropriate vehicle selection was vital for ensuring the maximum motor output and the fitment. In relation to that, the following considerations were addressed:

• Vehicle mass-to-power requirement ratio is one of the prime criteria. In general, power requirements are proportional to the chosen car weight and wheel size. Different vehicles were examined for mass to power ratio in order to make the most optimal vehicle selection that would be suitable for the in-wheel SRM, under development.

• The wheel size should be as large as possible to provide sufficient space to generate the required power output. Selected vehicle with an in-wheel SRM (wheel) should provide adequate clearances and tolerances to a car body (e.g., mud guard). A dimensional and mudguard clearance study was conducted using physical measurement/digitisation methods to predetermine the wheel size and clearances.

3.2.1 Vehicle mass to power requirement ratio

The vehicle selection for the in-wheel SRM is primarily dependent on: i) the vehicle mass/size, and ii) the vehicle power requirement. The power to weight ratio ($R$) is given by the total power output ($P$) divided by the kerb weight ($W$) of the car. Considering that the vehicle will have two in-wheel SRMs and the power output ($P$) is proportional to the volume of the rim-tyres, given by Equation 3.1 as follows:

$$P \propto 2V \quad (3.1)$$
Where \( V \) is the motor volume given by Equation 3.2:

\[
V = \frac{\pi h D^2}{4} \quad (3.2)
\]

Where, \( D \) is the outside diameter of the tyre based on a rim. The power to weight ratio (\( R \)) can be written as Equation 3.3:

\[
R \propto \frac{2\pi h D^2}{4W} \quad (3.3)
\]

Considering \( h \) (width of rim-tyre) and \( \pi \) as constants in Equation 3.3, this can be rewritten as Equation 3.4:

\[
R \propto \frac{D^2}{W} \quad (3.4)
\]

Table 3.1 shows ten cars which are popular in the Australia and easily available for the fitment (through manufacturers and dealers).

<table>
<thead>
<tr>
<th>Model</th>
<th>Rim-tyre specification (Ø mm)</th>
<th>Vehicle dimension L/W/H mm</th>
<th>Wheel base mm</th>
<th>Kerb weight (W kg)</th>
<th>Ratio ( D^2/W )</th>
<th>Suspension System</th>
<th>Brake system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small car</td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Hyundai Getz 1.6 Hatch</td>
<td>175/65/TR14 (583)</td>
<td>3825/1665/1495</td>
<td>2155</td>
<td>1133</td>
<td>300</td>
<td>Front MacPherson struts, rear torsion axle</td>
<td>Front/rear disc brakes</td>
</tr>
<tr>
<td>Barina Spark 1.2, Hatch</td>
<td>205/50/R15 (586)</td>
<td>3595/1597/1522</td>
<td>2375</td>
<td>960</td>
<td>657</td>
<td>Front MacPherson struts, rear torsion beam suspension</td>
<td>Front disc brakes, rear drum</td>
</tr>
<tr>
<td>Ford Fiesta 1.4, Hatch</td>
<td>195/50/VR15 (576)</td>
<td>3950/1722/1481</td>
<td>2480</td>
<td>1121</td>
<td>296</td>
<td>Front MacPherson struts, rear twist-beam</td>
<td>Front disc brakes, rear drum</td>
</tr>
<tr>
<td>Toyota Yaris 1.5 Hatch</td>
<td>185/60/HR15 (603)</td>
<td>3785/1695/1530</td>
<td>2460</td>
<td>1045</td>
<td>348</td>
<td>Front MacPherson struts, rear torsion beam and springs</td>
<td>Front disc brakes, rear drum</td>
</tr>
<tr>
<td>Suzuki Swift 1.6 Hatch</td>
<td>195/50/R16 (601)</td>
<td>3765/1690/1510</td>
<td>2390</td>
<td>1090</td>
<td>331</td>
<td>Front MacPherson struts, rear twist-beam</td>
<td>Front disc brakes, rear drum</td>
</tr>
<tr>
<td>Medium car</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hyundai 2.4 Sonata SLX Sedan</td>
<td>215/60/VR16 (664)</td>
<td>4800/1830/1475</td>
<td>2730</td>
<td>1530</td>
<td>288</td>
<td>Front MacPherson struts, rear multi-link with anti-roll</td>
<td>Front/rear, solid disc brake</td>
</tr>
<tr>
<td>Mazda 6 2.5 Sedan</td>
<td>205/60/VR16 (652)</td>
<td>4735/1795/1440</td>
<td>2725</td>
<td>1471</td>
<td>289</td>
<td>Front double wishbone, rear multi-link</td>
<td>Front/rear, solid disc brake</td>
</tr>
<tr>
<td>Toyota Camry 2.4 Sedan</td>
<td>215/60/VR16 (664)</td>
<td>4815/1820/1480</td>
<td>2775</td>
<td>1460</td>
<td>301</td>
<td>Front and rear MacPherson struts</td>
<td>Front/rear, solid disc brake</td>
</tr>
<tr>
<td>Suzuki SX4 2.0 Hatch</td>
<td>205/50/R173 (637)</td>
<td>4135/1730/1585</td>
<td>2500</td>
<td>1215</td>
<td>334</td>
<td>Front MacPherson struts, rear torsion beam</td>
<td>Front/rear, solid disc brake</td>
</tr>
<tr>
<td>Holden Cruze-CDX 4dr Sedan</td>
<td>215/50/VR17 (647)</td>
<td>4597/1788/1477</td>
<td>2685</td>
<td>1522</td>
<td>275</td>
<td>Front MacPherson struts, rear multi-link</td>
<td>Front/rear, solid disc brake</td>
</tr>
</tbody>
</table>
Large cars were excluded from these studies, due to their inappropriateness for the EV. Theoretically, large cars require far larger power outputs than would be currently possible. Several vehicle distributors were visited for the data collection on small and medium cars. As shown in Table 3.1, vehicles were grouped according to their respective dimensions, kerb weight, rim-tyre specifications, and power to weight ratios (based on the current rim-tyre sizes), suspension systems and brake system for finalising the vehicle selection.

Medium cars have certain advantages, due to larger wheel sizes (sufficient space for motor packaging) and bigger boot/engine space (batteries and other associated ancillaries). However, they also require more power output from the motor in order to drive the vehicle (based on the vehicle weight and size). The small cars offers lower weight and enough wheel space to generate the power output. Therefore an optimum vehicle needs to be found for space and power requirements. Table 3.1 shows vehicle characteristics for the small and medium cars. The evaluation considered the following cars: i) medium cars (Hyundai Sonata, Mazda 6, Toyota Camry, Suzuki SX4 and Holden Cruze) and ii) small cars (Hyundai Getz, Holden Barina Spark, Ford Fiesta, Toyota Yaris and Suzuki Swift).

The vehicle mass to engine power ratio of all the cars is indicated in Table 3.1. Four cars offering the highest power to weight ratio were chosen as potential vehicles for further analysis. The selection includes three small cars and one medium car. The following are the details of the cars:

- Holden Barina Spark 1.2 (357)
- Toyota Yaris 1.5 (348)
- Suzuki SX4 2.0 (334)
- Suzuki Swift 1.6 (331)

### 3.2.2 Mudguard clearance requirement

Clearance between the mudguard and the wheel was studied for selected cars and key findings are presented in this section. Wheel size defines the motor envelope for the in-wheel SRM. Motor power output is directly proportional to the motor size. The “National code of practice for light vehicle construction and modification-2011” and “Guide to modifications of ot or Vehicles-2011” under the MC category permit
increasing the overall wheel diameter by 50mm and the width by 25mm (Department of Infrastructure-Victorian Government 2011, Vic Roads- Victorian Government 2011).

Dimensional fitment and clearance studies were conducted on selected cars in order to finalise the vehicle selection. Evaluations were conducted for mudguard clearances on all three sides (D, E, (B-C) and H from Figure 3.1a, cross referencing Table 3.2). The Holden Barina Spark was identified as the best vehicle size from the present study. As shown in Table 3.2, the Holden Barina Spark had maximum clearances of 64mm (D-Table 3.2), 76mm (E-Table 3.2), 99mm ((B-C)-Table 3.2), and 218mm (H-Table 3.2). This clearance study identified an increase of 50mm wheel diameter as permitted by regulations. This study selected the 1” wheel and the Holden Barina Spark as a vehicle.

Figure 3.1: Dimension clearance analysis for mud guards of the Holden Barina Spark- a) rear wheel, b) front wheel

Table 3.2: Wheel clearance analysis

<table>
<thead>
<tr>
<th>Cars</th>
<th>Tyre specs</th>
<th>W/B</th>
<th>Mud guard</th>
<th>Tyre B-C</th>
<th>Rear Tyres</th>
<th>Front Tyres</th>
<th>Width</th>
</tr>
</thead>
<tbody>
<tr>
<td>Year</td>
<td>A</td>
<td>B</td>
<td>C</td>
<td>D</td>
<td>E</td>
<td>F</td>
<td>G</td>
</tr>
<tr>
<td>Holden Barina</td>
<td>2009</td>
<td>205/50/R15</td>
<td>2375</td>
<td>685</td>
<td>586</td>
<td>64</td>
<td>76</td>
</tr>
<tr>
<td>Toyota Yaris</td>
<td>2009</td>
<td>185/60/R15</td>
<td>2500</td>
<td>625</td>
<td>603</td>
<td>22</td>
<td>55</td>
</tr>
<tr>
<td>Suzuki SX4</td>
<td>2008</td>
<td>205/50/R17</td>
<td>2743</td>
<td>705</td>
<td>637</td>
<td>68</td>
<td>58.5</td>
</tr>
<tr>
<td>Suzuki Swift</td>
<td>2006</td>
<td>195/50/R16</td>
<td>2380</td>
<td>700</td>
<td>601</td>
<td>99</td>
<td>45</td>
</tr>
</tbody>
</table>
3.3 Car digitisation for virtual design

The Holden Barina Spark was digitised as it was difficult to ascertain the exact clearances with physical measurements. The physical measurements had the following limitations:

- The “National code of practice for light vehicle construction and modification-2011” specifies that wheel must not foul any part of the body or suspension. The Holden Barina Spark outer body is a 3D complex shape and physical measurement is limited by point to point measurement. Hence foul and clearance detection is limited to low accuracies.
- Physically fitting the 1’’ wheel on the old Barina Spark for the clearance measurement is time consuming and impractical (to buy a 1’’ wheel and fit it inside the car at the distributors).
- The current Holden Barina Spark has a bearing hub connected to the chassis. It is hard to dismantle the permanent bearing hub without damaging it. Hence the wheel width space physically measured with the reference plane is not very accurate.

Moreover, the actual Holden Barina Spark CAD data was not available. Hence, further 3D digitisation was conducted on the Holden Barina Spark to examine the clearances (with more accuracy). The vehicle digitisation was used for the virtual design to achieve the following objectives:

- space and clearances for deriving the rim-tyre envelope
- space and clearances for deriving the brake system envelope
- the Holden Barina Spark fitment study (complete vehicle fitment details in chapter 8)

3.3.1 Digitisation process

The chosen Holden Barina Spark was scanned using a hand-held 3D scanner (Figure 3.2) and the point cloud surface data was obtained for different recordings as shown in Figure 3.3. Generally three or more recordings were done to capture different surfaces during the scanning. The car body was fitted with black dots or blue tags to avoid the unnecessary reflections of car outer bodies (due to smooth edges). The process used the
blue tags as location points to generate point cloud data, which was then converted into a set of 3D surfaces.

![Hand held 3D Scanner](image)

**Figure 3.2: Hand held 3D Scanner**

![Overlapped surfaces](image)

**Figure 3.3: Two or three recordings of the car surface using 3D Scanner**

As a starting point several surfaces were obtained from the scanning (Refer to Appendix 2 on how scanning works). Generally, these surfaces overlapped (Figure 3.3); however, within the software, a set of tools is available for addressing any overlapping surface. The surfaces were then cleaned based on the quality and the capture percentage. The unwanted ambiguous surfaces were deleted and only one or two surfaces were kept. Then the refining process was used, which has four stages, as shown in Figure 3.4. The four stages of the refining process included: i) point cloud (Figure 3.4a), ii) mesh data (Figure 3.4b), iii) integration (Figure 3.4c), and iv) final mesh model creation (Figure 3.4d). Firstly, a set of point cloud data was obtained, which was subsequently converted into the mesh data. Mesh data was patched using tangency control or curvature continuity methods. Next, misaligned surfaces were converted to a single surface by
integrating and aligning two or more surfaces, depending on the model complexity. The data scanned at different time intervals were meshed and integrated to form a single surface. This process resulted in cleaning up of any undesirable ambiguities in the scanned data. The final mesh refinement was achieved by cleaning and removing the unwanted data from the surface body. The scanned data was later converted into class A surfaces with the G2 curvature continuity to one another.

After the aforementioned procedure was completed, the created surfaces related to boundary surfaces were reconstructed using offsets or surface extensions. Other techniques used were bridging tangencies or creating fillets, for smooth enlarged surfaces. Non Uniform Rational Bezier Splines (NURBS), are mathematical representations of the 3-D geometry that can accurately describe any shape from simple basic primitives to the most complex 3-D organic free-form surfaces used to define the solids (e.g., automotive exterior). A NURBS surface is dependent on the curve and is defined using: i) degree of the curve, ii) control points used for curve, iii) knots (time intervals), and iv) an evaluation rule. The degree is a positive whole number defining the curves degree (e.g., quadratic, cubic): and control points usually carry the weights (defining the shape). Knots are lists of numbers defining the time curve intervals. An evaluation rule is an equation defined to connect the curves including their individual control positions. The NURB curves at knots experience kinks with sudden direction changes. To control the kinks and maintain geometric relations with the corresponding neighbouring surface, these curves are used to refine the scanned data. The NURBS refinement and control using Bezier curves and mathematical methods are explained in Appendix 3. The curvature comb was generated on the individual surface, in order to refine the required surface quality. Generally, several parameters, including curvature continuity (C values) and tangential continuity (G values), were applied on generated curves to effectively clean and fine-tune the surfaces. ‘A’ surfaces generated around mudguards were polynomial surfaces that required more than three degree quadratic equations (aforementioned in Appendix 3) to resolve. Patching, sewing, tangency and surface refinement were performed until a fine surface was obtained. Zebra stripes and curvature continuity methods were used to verify and improve the curvature comb of the surface in order to obtain the appropriate Bezier surface.
Figure 3.4: Vehicle digitisation using hand held 3D Scanner (4 stages), a) point cloud, b) mesh creation, c) integration and d) final mesh model for refinement

The required materials, textures and paint were modelled to obtain the final digital model of the Holden Barina Spark as shown in Figure 3.5.

Figure 3.5: Final digital model of the Holden Barina Spark

3.3.2 Rim-tyre envelope

The “National code of practice for light vehicle construction and modification-2011” specifies that a wheel must not foul any part of the body or suspension under any operating conditions and must be contained in the bodywork when the wheels are in the straight-ahead position (Department of Infrastructure-Victorian Government 2011). The following dimensions are determined from the digital model to define the rim-tyre envelope: i) chassis width is 1161mm, ii) wheel track is 1414mm, iii) wheel outer is 1619mm, and iii) car body outer is 1597mm. Figure 3.6 and Figure 3.7, show a 1 ”
rim-tyre fitted to digital model in front and side views respectively. The clearances at all the points were checked between rim-tyre and mudguard and were found to exceed 10mm (Figure 3.7). These studies confirmed available wheel space as 655mm diameter and 220mm width in the Holden Barina Spark.

![Figure 3.6: Digital model showing suspension and tyre details](image)

Figure 3.6: Digital model showing suspension and tyre details

![Figure 3.7: Mud guard clearance, digitised Holden Barina Spark fitted with R17 wheel](image)

Figure 3.7: Mud guard clearance, digitised Holden Barina Spark fitted with R17 wheel

### 3.3.3 Brake system space requirement

The ADR 31/02 specifies all cars to have a mechanically operated primary brake. In the ICE Holden Barina Spark, the drum brake is located inside the rear wheel. In the wheel, the SRM will be housed and the brake needs relocation. As the SRM under
development would have a brake by wire capability, a simple low braking effort system would also suffice. As a result, in this study, the brake system envelope was investigated. Using the digital model the clearance between the chassis and the suspension damper defined a new brake system space (complete brake design details in Chapter 6). As shown in Figure 3.6, it was found that ~70mm of space was available for housing a new brake system between the chassis and the suspension damper (with 60mm diameter).

### 3.4 Rim design

The wheel consists of a rim and a tyre. The rim provides the structural rigidity and the tyre provides the vehicle traction as well as cushioning when the vehicle travels on uneven surfaces. Rim and tyre are interdependent and work as a single component influencing the vehicle drive characteristic.

Rim topology consists of a cylindrical rim and a cross sectional web (also called an end cap) supporting it, as shown in Figure 3.8. The rim and end cap is constructed as: i) a single piece, ii) a two-piece, and iii) a three-piece. It is called a single piece if the end cap and the cylindrical rim are constructed as one solid piece. In a two-piece, the end cap and the cylindrical rim are constructed as two pieces. In the three-piece rim, the end cap is further divided into two pieces with a cylindrical rim. The two and three-piece rims are joined by welding, bolting, riveting, and seaming methods. The three-piece rim was not considered in this research due to poor structural performance.

The end cap is offset to the rim centreline (as shown in Figure 3.8) and based on the offset, the rim is classified as: i) zero, ii) positive, and iii) negative. In the case of the zero offset, as shown in Figure 3.8 (i.e. from the centre black line to the blue line), the end cap is closer to the rim centre line. In the positive offset, as shown in Figure 3.8 (i.e. from the green line to the blue line), the end cap is closer to the front side of the rim centreline on the positive side. The negative offset, as shown in Figure 3.8 (i.e. from the centre black line to the red line), refers to the end cap closer to the back side of the rim centre line. In the negative offset rim, the end cap is closer to the rim back side, hence the motor space becomes small. For this reason, the negative offset rim is not considered in this research. The end cap construction uses supports to provide the
structural rigidity to the rim. The following are typical end cap supports used: i) solid, ii) hollow, iii) three support, and iv) five support.

In the wheel new proposed SRM adds extra weight and thermal load. Hence, the structural performance of the rim is even more crucial. As the rim is proposed to be an integral part of the SRM, in this research the rim design was based on the following key requirements:

- Adherence to the “Rims and Tyre standards-Australia 2010” to ensure that the rim nomenclature was based on the relevant standard used in Australia. Compliance with standards (National code of practice for light vehicle construction and modification-2011, Vic roads modification guides 2011) for defining allowable limits for the custom-designed small car rim. Using the standard limits as a basis, developing the custom-designed rim that can accommodate the motor.
- Selecting the appropriate material for the custom-designed rim. The material determines structural rigidity, thermal stability, weight, and cost of rim. The material also determines manufacturing processes, ensures absence of defects, thus preventing failures.
- The key factors affecting the rim design are weight, rigidity, durability, and thermal stability. The weight optimisation was done on five rims (topologies) to select an appropriate rim based on structural rigidity, thermal stability, and life cycle assessment. The FEM was used to access these criteria.

### 3.4.1 Rim-tyre sizes

Motor power output is dependent on the rim size and the shape. Rim-tyre nomenclature defines rim shapes and sizes. In Australia, the requirement for the rim and the tyre conforms to “Rims and Tyre standards-Australia 2010”, which provides basic dimensions for the rim diameter, width, and flange shape. For the passenger car, two profiles are available: i) drop centre (DC) and ii) flat contour (with tyre bead seating retentioner). The DC rim has a well on one side of the bead areas as shown in Figure 3.8. The bead is constructed on both sides of the rim for easy dismantling of the tyre. Flat contours do not use the wells; instead the bead seating retentioner is used. The flat contour is generally used in light trucks and is not easy to disassemble during tyre
servicing, hence it was not considered in this research. The DC contours are offered in two profiles with the flange and the flange. The profile is used in this research due to the maximum rim space available on account of its shape. Car rims are used with bead profile shapes called J, JJ, K, JK, B, P and D. Here, "J" implies that the bead profile shape resembles the letter "J" (e.g., "JJ" profiles are often used for SUV wheels as they have better grip). The difference in dimensions between the bead profile stems from the rim flange shape. The J bead profile is used in most passenger cars as it offers the required grip between the tyre and the rim. The J bead profile is used in this research.

![Diagram of rim and tyre configuration](image1)

**Figure 3.8: J tyre configuration**

The passenger vehicle rim-tyre is specified as www/hh/Rdd. Here, as shown in Figure 3.9, www determines the rim-tyre width, hh refers to the percentage ratio of tyre width to tyre height ($t=\frac{www}{hh}$), and Rdd is the rim inside diameter in inches. In some instances nomenclature also covers wheel offsets, number of bolts and the PCD.

![Diagram of rim and tyre nomenclature](image2)

**Figure 3.9: Rim-Tyre nomenclature**
Tyre selection is dependent on the rim used, as both work together for vehicle functional performance. The “National code of practice for light vehicle construction and modification-2011” and “Vic roads modification guides 2011” governs the rim-tyre modifications for lightweight vehicles. Based on a clearance study from the digital model a suitable 205 J50 R17 rim-tyre was selected.

3.4.2 Rim design

The rim is an integral part of the new proposed SRM. The SRM adds extra mass and thermal loads to the rim. The available off the shelf rim is not suitable as the end cap offset may not provide adequate motor space and is also heavier. The design objective was to have a light weight rim, which provided a maximum motor space. To achieve the objectives, five rim designs were created.

As shown in Figure 3.10 a single piece rim was designed with a positive offset and a solid end cap (referred to as rim 1). The rim 1 has 11kg weight, with four central bolt holes and a valve hole as shown in Figure 3.10.

Figure 3.10: Rim 1, a positive offset single piece rim with solid end cap

As shown in Figure 3.11, the second rim was also a single piece rim design, but had a zero offset with the solid end cap (referred to as rim 2). Rim 2 has 9kg weight and relief
holes were drilled in the solid end cap for the weight reduction. The rim also has four central bolt holes and a valve hole as shown in Figure 3.11.

**Figure 3.11:** Rim 2, a zero offset single piece rim with solid end cap

As shown in Figure 3.12, a two piece rim with a positive offset using a hollow end cap was designed (referred to as rim 3). The rim 3 weighed 7kg in total and has eight bolt holes near the cylindrical rim as shown in Figure 3.12. The bolt holes are near the cylindrical rim on the end cap face. It was seam welded between the cylindrical rim and the end cap.

**Figure 3.12:** Rim 3, a positive offset two piece rim with hollow end cap
The Positive offset, single piece rim, with five support end cap is shown in Figure 3.13 (referred to as rim 4). It weighed 6kg in total and five bolts are located near the central portion of the five support end cap.

*Figure 3.13: Rim 4, a positive offset single piece rim with five support end cap*

The positive offset single piece rim, with three support end cap is shown in Figure 3.14 (referred to as rim 5). It weighed 5.5kg in total and five bolts are located near the central portion of the three support end cap.

*Figure 3.14: Rim 5, a positive offset single piece rim with three support end cap*
3.4.3 Material selection

In this research, car rims were vital for the vehicle performance. Material selection is a key parameter in the rim. The in-wheel SRM adds an extra weight on the rim and hence structural rigidity is important. The SRM is expected to reach a high temperature; hence the rim thermal stability is important. The manufacturing process governs defects; these defects result in structural failures. Hence selecting the appropriate rim material and the manufacturing process is crucial for the vehicle performance (Cura and Curti 2004). In EVs, the lighter rims provide increased range by virtue of reduced weight. A moderate cost rim is required as the design goal is to have a functional vehicle at a reasonable cost. The rim material was evaluated based on these five main criteria:

- structural rigidity
- thermal stability
- manufacturing defects
- weight
- cost

Historically, most car rims used steel. However, recently alloy rims have proved to be a better choice and are becoming a standard in the mainstream automotive market. The key advantages of alloy rims are stylish appearance, increased tensile strength, rapid heat dissipation, and light weight. Alloy rims are commonly manufactured from aluminium or magnesium alloy metals. Aluminium alloy rims are manufactured using two processes, spinning and die-casting. Other available rims use high cost lightweight materials such as chrome moly 4130, titanium, and carbon fibres. In the following section rim materials were evaluated to finalise the material selection: i) steel, ii) aluminium die cast, iii) carbon fibre, iv) magnesium, and v) aluminium spun.

A steel rim has the following characteristics: i) structural rigidity – good as it is bulky, ii) thermal stability– medium, iii) manufacturing defects– poor due to defects associated with these rims (e.g., cracking during forming operations) (Bhattacharyya, Adhikary et al. 2008), iv) weight– poor as it is too bulky, adding unnecessary weight to the vehicle, and v) cost– good as these are cheap.

An aluminium die cast rim has the following characteristics: i) structural rigidity– poor due to porosity, ii) thermal stability– medium, iii) manufacturing defects– poor
dimensional stability due to the shrinkage (Wang, Sui et al. 2011) and requires a stringent quality monitoring process for porosity, iv) weight– medium due to the aluminium density, and v) cost– medium due to the monitoring and the wastage resulting from the porosity (Hsu and Yu 2006).

A magnesium alloy rim is also known as an E-Mag, and is primarily used in high cost cars for the aesthetic appearance. The magnesium alloy rim has the following characteristics: i) structural rigidity– poor due to the inherently low structural performance (Han, Yang et al. 2012), ii) thermal stability– poor due to the low thermal coefficient, iii) manufacturing defects– poor due to gas pores in the middle of the spoke, shrinkage at the rim top and the rim-spoke junction, iv) weight– good as it is light , and v) cost– poor, as they are x-rayed and heat-tempered to ensure that there are no obvious physical flaws and to assess their durability due to defects (Wang, Li et al. 2007).

A carbon fibre rim has the following characteristics: i) structural rigidity– good, however the carbon fibre orientation plays a vital role, ii) thermal stability– good, however the carbon fibre orientation plays vital role, iii) manufacturing defects– medium due to the rigorous testing required to establish the structural rigidity, iv) weight– good as it is very light (Carbon Revolution 2011), and v) cost-poor, as they are very costly.

An aluminium spun alloy rim has the following characteristics: i) structural rigidity– good, as the aluminium alloy 6061-T6 has a 340MPa material yield strength (Merlin, Timelli et al. 2009), ii) thermal stability– good, as they possesses low elongation percentage when compared with other aluminium grades, iii) manufacturing defects– good, as it is spun it has no porosity defects (e.g., die cast alloys), iv) weight– medium due to the density of aluminium, and v) cost– medium, as they are relatively cheaper than those of magnesium and carbon fibre material.

Table 3.3 summarises rim materials with poor, medium and good scores based on material characteristics. Compared to other alloys, the aluminium spun rim has medium weight and the medium cost. The aluminium rim provides three key benefits—structural rigidity, thermal stability and low manufacturing defects. They also enhance the aesthetic appearance of the cars. Hence, the aluminium alloy 6061-T6 spun rim was selected in this study.
Table 3.3: Comparison of different rim materials

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Steel</th>
<th>Aluminium die casted</th>
<th>Magnesium</th>
<th>Carbon fibre</th>
<th>Aluminium spun</th>
</tr>
</thead>
<tbody>
<tr>
<td>Structural rigidity</td>
<td>Good</td>
<td>Poor</td>
<td>Poor</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Thermal stability</td>
<td>Medium</td>
<td>Medium</td>
<td>Poor</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Manufacturing defects</td>
<td>Medium</td>
<td>Poor</td>
<td>poor</td>
<td>Poor</td>
<td>Good</td>
</tr>
<tr>
<td>Weight</td>
<td>Poor</td>
<td>Medium</td>
<td>Good</td>
<td>Good</td>
<td>Medium</td>
</tr>
<tr>
<td>Cost</td>
<td>Good</td>
<td>Medium</td>
<td>Poor</td>
<td>Poor</td>
<td>Medium</td>
</tr>
</tbody>
</table>

3.4.4 Rim topology optimisation

In this section, five rim designs were evaluated for weight optimisation, without compromising the required thermal, structural and durability parameters. Rim is a structural component; hence it should not fail structurally under static loads and perform with required durability under service loads. To achieve the above objectives, finite element (FE) method was used to examine five rims, focusing on the Von-mises stress, the strain and the total deformation, when subjected to thermal and structural loads. A fatigue load was modelled to examine the durability of selected rims. The following were key objectives for the rim topology optimisation adopted in this research:

- **Thermal stability**: The in-wheel SRM stators are expected to reach high temperatures during the operation and hence forced cooling is planned. The planned forced cooling exhaust was in the close vicinity of the rim. Hence transient thermal analysis was modelled with an ambient temperature load of 22°C and a relative temperature increase of 70°C, which was modelled as a thermal load, with 50W/m²-°C convection coefficient.

- **Structural rigidity**: The rim is an integral part of the in-wheel SRM. The SRM has a small air gap and also adds extra mass to the rim. A structurally rigid rim topology is required to: i) handle the extra motor weight and ii) minimise deformations that affect the motor air gap. To evaluate the structural rigidity a uniformly distributed load of 8kN (one and half times the rear axle weight) was modelled at a 0 rim arc area complaint with the SAE J20 standard.

- **Fatigue analysis**: The rim topologies fatigue life was examined as a safety parameter in accordance with the SAE J2530 standard by modelling a fully reversed constant amplitude load on the rim. Keeping factor of safety (FOS) above 1.4.
### 3.4.4.1 Rim optimisation flow diagram

This section describes the FE methods for rim optimisation and Figure 3.15 depicts the flow diagram.

**Finite element methods**

**Geometric data (rim topologies):**
1) Rim 1 (11kg)- positive offset single piece rim/solid end cap
2) Rim 2 (9kg)- zero offset single piece rim/solid end cap
3) Rim 3 (7kg)- positive offset two piece rim/hollow end cap
4) Rim 4 (6kg)- positive offset single piece rim/five support end cap
5) Rim 5 (5.5kg)- positive offset single piece rim/three support end cap

**Boundary conditions**
1) The rim bolt holes were fixed
2) Rim 3 faces at end cap locations were modelled with no penetration contact patch

**Material properties**
1) Aluminium-6061T6
2) $E=69\text{GPa}$
3) $\nu=0.33$
4) $S_{ut}=310\text{MPa}$

**FE Modelling**
A tetrahedral mesh with a global element size 6 and a local element size 1 to 3 in the hole and patch areas. It is modelled with a smooth transition ratio 0.272 between elements.

**FE results**
1) Maximum Von-mises stress
2) Maximum deformation
3) Maximum strain
4) Fatigue life based on mean stress correction
5) FOS value based on mean stress to ultimate tensile strength

1) **Thermal load modelling**
A transient thermal load of $22^\circ$-$70^\circ\text{C}$ on inner rim faces with a $50\text{W/m}^2\text{C}$ convection coefficient on outer rim faces

2) **Static load modelling**
8kN uniformly distributed downwards load was modelled (towards ground zone) over a 0 arc area on rim outer face

3) **Fatigue load modelling**
Constant amplitude fully reversed load is modelled

1) To evaluate rims based on low:
   - maximum Von-mises stress, maximum deformation and maximum strain
2) To meet SAE J2530 fatigue life requirements and $F\geq 1$.

**Evaluate FE results for all five rim topologies**

**Yes**
- Optimised rim topology considered for this research

**No**
- Not considered for this research

*Figure 3.15: Rim optimisation flow diagram using FE methods*
The five rims were modelled using FE methods to optimise the rim weight. All the rims were modelled with thermal, structural and fatigue loads to optimise the topology. Initially, the transient thermal loads and then static loads were modelled to optimise the total stress, the strain and the total deformations. Also fully reversed constant amplitude loads were modelled for the fatigue life cycles and FOS of the rims.

### 3.4.4.2 Finite element modelling

The FE models were developed for all five rim topologies in the Ansys® test bench 13.1, a commercial FE software. Initially, each rim was modelled using the local and the global 3D tetrahedral mesh with 4 nodes; an example of the rim 3 is shown in Figure 3.16. Rim 3 FE model consisted of 72370 elements and 146110 nodes. The valve hole, bolt holes, and rim perimeter faces were modelled with a local tetrahedral finer mesh of 1 to 3mm. The global tetrahedral mesh of 6mm or more was modelled into the rest of the rim with a smooth transition ratio of 0.272. Transition ratio allowed a smooth flow from the local to the global mesh model within the rim topology.

![Rim 3 FE model](image)

**Figure 3.16: Rim 3 FE model**
3.4.4.3 Material and boundary conditions

The aluminium 6061-T6 engineering material was used into all five FE rim models. In order to use exact material data in the FE model, supplier data was used (ASTM International 2011). The engineering material data of Aluminium 6061-T6 material are: i) Young’s modulus \((E)\) 69GPa, ii) Poisson’s ratio \((\nu)\) 0.33, iii) coefficient of thermal expansion \((\alpha)\) \(2.6 \times 10^{-6}/\degree C\), iv) density \((\rho)\) 2.7g/cc and v) ultimate tensile strength \((\sigma_u)\) 310MPa when T6 hardened.

In order to predict the fatigue life of the rim, S-N curve for Aluminium 6061-T6 was composed in the model based on the simple methods (ASTM International 2011). It uses the relationship between alternating stresses \((\sigma_a)\) over repetitive cycles. Based on the relationship of ultimate tensile strength \((\sigma_u)\) and Von-mises stresses \((\sigma)\) distribution obtained from static study, the FE provides an estimation of fatigue life. However to compensate the effect mean stress on the true fatigue strength Good man, Gerber and Soderberg approaches are used. The mean stress correction using Gerber approach, experimentally known to be good for ductile material was used. The Gerber approach fits the curve with the horizontal axis representing the magnitude of the mean stress scaled by the value of the mean ultimate tensile strength. The vertical axis correspond the correction for fatigue strength (endurance limit). The fatigue strength reduction factor of Aluminium 6061-T6 was used as 0.52.

In the FE method the boundary conditions were modelled to simulate the physical conditions of the rim. In a car, the rim is supported through the hub at the bolt holes location as shown in Figure 3.17. The fixed support boundary conditions were used at bolt holes to restrain rotational and translation movements. Further the contact patches were required in the FE model to establish the working relation within multi body parts. Rim 3 internal faces were modelled as a contact patch with the end cap faces. As shown in Figure 3.16, the no penetration contact patch was modelled between the rim and the end cap faces. This contact relationship within the FE model ensured that at any given load, multi body parts within FE models acted as single entity avoiding self-interference.
### 3.4.4.4 Loading

*Figure 3.17* is an example of rim 3, and summarises loads and boundary conditions modelled within the FE environment. In this research three types of loads were modelled: i) the thermal load, ii) the static load, and iii) the fatigue load.

The SRM resided inside the rim and as a consequence the rim temperature was expected to rise. IEEE 11-2000 standard, titled “IEEE Standards for Rotating Electrical Machinery for Rail and Road Vehicles-2000”, specifies maximum surface temperature of enclosed ventilated motor to be restricted to 70°C (IEEE 2000). The ambient temperature of 22°C and the relative time dependent temperature rise of 70°C were modelled on the rim and the end cap internal faces. The convection coefficient of 50W/m²°C was modelled to the rim and end cap external faces.

In automotive industries, a uniform distributed load is modelled to check compliance with the SAE J2530 standard for evaluation of rims. The uniform distributed load was modelled over a 40° arc area on the outer rim faces as shown in *Figure 3.17*. The load is 1.5 times the rear axle weight of the vehicle; this scaling of load compensates for the effect of road bumps on the rim during the vehicle ride. *Equation 3.5* was used to model the load condition in accordance to SAE standard:

\[
F = 1.5F'
\]

Whereby \( F \) is the total load modelled and \( F' \) is the rear axle weight of the vehicle. The Barina Spark weight is 960kg and the two motors weight on the rear of the vehicle is 100kg. The rear axle weight \( F' \) is 5.2kN. Using *Equation 3.5* the total uniform distributed load acting on the rim was calculated as 7.8kN and it was rounded off to 8kN. The total uniform distributed downwards load of 8kN (towards the ground zone) was modelled over a 0° arc area on the outer rim face as shown by arrows in *Figure 3.17*.

The rim is subjected to cyclic loads during the vehicle travel and as a result fatigue in the rim may cause a failure. Hence the rim fatigue load was modelled in accordance with the SAE J2530 standard. The SAE J2530 standard defines the radial fatigue, the cornering fatigue and the impact fatigue test procedures. The SAE standard defines 1,850,000 minimum cycle requirements for the rim. As per the standard, the rim is required to fulfil the minimum life cycles and possess a FOS of 1.4 or more. The fully
reverse constant amplitude loading was modelled within FE environment. In the fully reverse constant amplitude modelling, the static load is reversed for a constant periods. 8kN uniform distributed load modelled over a 0 arc area on outer rim face.

22°C ambient temperature and relative temperature increase of 70°C modelled on rim and end cap inner faces. Bolt holes as fixed support.

50W/m²°C, convection coefficient modelled to rim and end cap outer faces.

Figure 3.17: Rim loads and boundary conditions

3.4.4.5 Finite element results

The FE results of all five rims is summarised in Table 3.4., with the maximum Von-mises stress, maximum total deformation, maximum strain, the FOS, and the fatigue life assessment. The FE result discussions use the rim 3 figures as an example; however the other four rim FE analysis figures are attached in Appendix 4. As shown in Table 3.4, 59MPa and 57MPa maximum Von-mises stress concentrations were observed in the solid single piece rims 1 and 2 respectively. Rims 1 and 2 weighed 11kg and 9kg respectively. Rim 3 with the hollow end cap as shown in Figure 3.18, weighed 7kg, and a 164MPa maximum Von-mises stress concentrations near rim circumference and bolt holes. Rim 4 had a 125MPa maximum Von-mises stress value and weighed 6kg. The rim 5 had a 373MPa maximum Von-mises stress concentration value and weighed 5.5kg.
Table 3.4: Summary on FE evaluations of different rim designs

<table>
<thead>
<tr>
<th>Referr ed as</th>
<th>Rim Description/ Mass (Kg)</th>
<th>Max. Von- mises stress (MPa)</th>
<th>Max. total deformation (mm)</th>
<th>Max. total strain (mm/mm)</th>
<th>FOS</th>
<th>Minimum life cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rim 1</td>
<td>Single piece construction positive offset with solid end cap (11kg)</td>
<td>59</td>
<td>1</td>
<td>0.001</td>
<td>1.5</td>
<td>100,000,000</td>
</tr>
<tr>
<td>Rim 2</td>
<td>Single piece construction zero offset with solid end cap (9kg)</td>
<td>57</td>
<td>0.6</td>
<td>0.001</td>
<td>1.5</td>
<td>100,000,000</td>
</tr>
<tr>
<td>Rim 3</td>
<td>Two piece rim with hollow end cap (7kg)</td>
<td>164</td>
<td>0.3</td>
<td>0.002</td>
<td>1.5</td>
<td>3,364,600</td>
</tr>
<tr>
<td>Rim 4</td>
<td>Five support single piece construction (6kg)</td>
<td>125</td>
<td>1.5</td>
<td>0.001</td>
<td>0.65</td>
<td>1,553,800</td>
</tr>
<tr>
<td>Rim 5</td>
<td>Three support single piece construction (5.5kg)</td>
<td>373</td>
<td>5.55</td>
<td>0.005</td>
<td>0.22</td>
<td>25,000</td>
</tr>
</tbody>
</table>

Figure 3.18: Rim 3 Von-mises stress areas (enlarged area showing high stress concentration areas, stress in MPa)

The total deformations of all rim topologies are summarised in Table 3.4. Though solid in construction, rims 1 and 2 showed total maximum deformations of 1mm and 0.6mm respectively. In contrast, the lowest maximum deformation of 0.3mm in the downward direction as shown in Figure 3.19 was found for rim 3. The maximum deformation in
the lower portion of the rim opposite to the load area was due to overhang in rim 3. Rims 1 and 2 deformed more than rim 3, due to variation in bolt hole locations. The rim 3 had bolt holes near the cylindrical rim, while rims 1 and 2 had it on the central locations. Rims 4 and 5 had total maximum deformations of the magnitude 1.5mm and 5.55mm, respectively. Strain levels, as shown in Table 3.4, were negligible for all the rims. However, rim 5 showed the highest strain of 0.005, among all rims, at the support edges near the inside diameter.

**Figure 3.19: Rim 3 deformation analysis (mm)**

The FOS is the ratio of material yield strength to the maximum Von-mises stress concentration in the rim topology. The maximum Von-mises stress in the rim topology is dependent on the cross section of the rim (which governs the weight). Generally, the FOS should be more than 1.4 in the automotive rim. Rims 4 and 5 displayed a low FOS of 0.65 and 0.22 respectively. Rims 1 to 3 displayed a FOS of 1.5 as shown in Table 3.4.

FE methods for fatigue life predicted critical failure areas due to stress concentrations. The fatigue life results for all rims are shown in Table 3.4; those for the rims 1, 2, and 3,
exceeded the SAE rim fatigue life requirements (SAE J2530 2009). As shown in Figure 3.20, rim 3 had the lowest fatigue life of 223,090 at the rim inner circumference was observed due to sharp edges (singularities, hence neglected). Elsewhere the minimum fatigue observed was 3,364,600 which surpassed the SAE standard requirements. This results indicated that these are the areas need to be improved in the design to further enhance the fatigue life of the rim. These results also indicated that the rims 4 and 5 were below the recommended SAE fatigue life limits.

**Figure 3.20: Rim 3 fatigue life cycles**

*Figures 3.18, 3.19 and 3.20 respectively depict the rim 3 maximum Von-mises stress concentration areas, maximum deformation and predicted failure area. Amongst the compared five rims, rim 3 had a median weight of 7kg and performed well under the modelled loads, with the lowest maximum deformation of 0.3mm, and the maximum Von-mises stress of 164MPa. The fatigue life of rim 3 exceeded the SAE J2530 requirements relevant for this study. These findings indicated that the hollow end cap two piece rim weighing 7kg (rim 3) was the most suitable choice for the in-wheel SRM.*
3.5 Tyre selection

In this research, tyres were studied with the objective to increase the EV range by selecting the lowest rolling resistance tyre. There are two basic tyres based on the construction i) bias ply and ii) radial ply. The bias ply tyre has a carcass made up of reinforcements running at $\theta$ to 0, while the radial has the reinforcement perpendicular to the circumferential direction. This arrangement within the radial tyre makes it circumferentially stiffer and hence the low resistance. In this research radial tyre was selected based on the aforementioned reasons.

A tyre when in motion has a contact with the ground surface. Both wheel and ground are subject to deformations and the tyre springs back to the original position once the contact area is passed. This deformation causes the rolling resistance. Sliding between the tyre-ground and aerodynamic drag are other parameters affecting the rolling resistance. The tyre velocity is dependent on the tyre deformations, and the tyre deforms every time the tyre contacts the ground surface. The contact area is called a tyre-ground zone. As the tyre enters the ground zone, the tyre slows down, leading to the tyre circumferential compression. As there is limited sliding between the tyre and the ground in this zone, the velocity at the contact point is the same as the velocity at the centre of the wheel. This is the reason for rigid wheels having higher spin speeds than pneumatic tyres under the same load. The tyre radius $R$ at the ground contact is under deformation $R_l$ and tyre elastic return causes it to have the effective rolling radius of $R_e$. Figure 3.21 shows the rotating pneumatic tyre and explains all three radii, peripheral velocity, and forces acting on it. The tyre-ground contact leads to various changes in the tyre, affecting the effective rolling radius ($R_e$). Rolling radius is defined as the ratio of the velocity ($V$) and the angular velocity ($\Omega$) of the vehicle as shown in Equation 3.6 (Genta 2006):

$$R_e = V / \Omega$$  \hspace{1cm} (3.6)
The rolling resistance is the resistance offered by the ground to the movement of the vehicle. *Equation 3.7* below represents the rolling resistance:

\[ F_r = -f \cdot F_z \]  

(3.7)

Where, \( F_r \) is rolling resistance, \( f \) is rolling resistance coefficient (RRC), and \( F_z \) is force at tyre-ground contact in upward direction. The tyre RRC depends on speeds and environmental conditions. In this research RRC was evaluated at normal dry environmental conditions. Radial tyres are characterised by lower vertical stiffness, leading to decreased loaded radius \( (R_l) \), but are circumferentially stiffer with rolling radius \( (R_e) \) values closer to rigid wheel \( R \) (Genta 2006). This leads to lower spin speeds in the contact zone than a rigid wheel. The RRC increase is non-uniform and its magnitude varies with the speed, whereby it is slow at the beginning, after which it increases at a faster rate, as given by *Equation 3.8*:

\[ f = f_0 + K \cdot V^2 \]  

(3.8)

The value of \( f_0 \) (initial rolling coefficient) varies depending on the tyre used and the constant \( K \). The comparisons of the RRC with a variable speed were conducted for the:

- Low rolling resistance (LRR) with a 0.007 initial rolling coefficient
• Normal rolling resistance (NRR) with a 0.012 initial rolling coefficient

*Table 3.5* represents both tyres for the initial and the final RRC at different car speeds. *Figure 3.22* shows the final RRC of LRR \( (f_1) \) and NRR \( (f_2) \) tyres at different velocities ranging from 10km/h to 100km/h. As observed in *Figure 3.22*, the RRC varied with the speed, whereby the increase was small at lower speeds, but increased at a much faster rate for higher vehicle speeds. It is concluded from *Figure 3.22*, that the LRR tyre RRC is much smaller and changed at a higher rate with the vehicle speed increase. In this research based on the aforementioned reasons, the LRR tyre was chosen for the vehicle. The environmental conditions affect the RRC. The LRR tyre suffers with the reduced performance in high wet conditions and subject to tyre slips. To overcome high wet performance issues of LRR tyre, different reinforcements are used. Such reinforcements typically are silica and carbon black. The various studies conducted on the performance of carbon black and silica reinforcement materials reported improvements performance of LRR tyres in wet conditions (Bornai, Touzet et al. 1997, Engehausen, Rawlinson et al. 2001, Ghosh, Sengupta et al. 2011). Nonetheless, in practice, silica is increasingly growing in popularity, compared to the carbon black, due to its lower RRC and improved wet grip performance (Ten Brinke, Debnath et al. 2003). Hence in this research, the silica based LRR tyre was selected.

*Table 3.5: Comparison charts for rolling resistance at different velocities*

<table>
<thead>
<tr>
<th>Initial RRC for LRR ( f_0 )</th>
<th>Initial RRC for NRR ( f_0 )</th>
<th>Constant ( K ), ( hr^2/km^2 ) ((1 \times e^{-07}))</th>
<th>Velocity ( V ), km/h</th>
<th>Final RRC for LRR ( f_1 ) ((1 \times e^{-02}))</th>
<th>Final RRC for NRR ( f_2 ) ((1 \times e^{-02}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>10</td>
<td>0.705</td>
<td>1.21</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>20</td>
<td>0.72</td>
<td>1.22</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>30</td>
<td>0.745</td>
<td>1.25</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>40</td>
<td>0.78</td>
<td>1.28</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>50</td>
<td>0.825</td>
<td>1.33</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>60</td>
<td>0.88</td>
<td>1.38</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>70</td>
<td>0.945</td>
<td>1.45</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>80</td>
<td>1.02</td>
<td>1.52</td>
</tr>
<tr>
<td>0.007</td>
<td>0.012</td>
<td>5</td>
<td>100</td>
<td>1.20</td>
<td>1.70</td>
</tr>
</tbody>
</table>
Rim is an integral part in the proposed in-wheel SRM and the surface temperature of the SRM will affect the tyre inflation pressure. Hence in this section the relation between the inflation pressure and the rolling resistance was examined. The empirical formula provided by SAE J670e to measure the RRC taking into account the influence of the inflation pressure is given by Equation 3.9 below (Genta 2006):

$$f = \frac{K'}{1000} \left( 5.1 + \frac{5.5 \times 10^5 + 90F_z}{p} + \frac{1100 + 0.038F_z}{p} V^2 \right)$$  \hspace{1cm} (3.9)$$

Where $K'$ has a value of 0.8 for radial tyre and the upward force $F_z$ is 4.5kN for the proposed design. The vehicle is considered to be travelling at a constant speed of 60km/h in a straight line. The recommended inflation pressure value by a placard standard for selected 205/50R17 tyre is 220632N/m$^2$ (32psi). This value was varied by 25% on either ends to examine the effect on the rolling resistance. The rolling resistance coefficient and inflation pressure was plotted as shown in Figure 3.23. The RRC decreased with the increase of tyre inflation pressure.
Figure 3.23: Variation of rolling resistance coefficient with tyre inflation pressure

It can be noted in Figure 3.23 that rolling resistance was gradually reducing with the increased pressure and after 35 psi it has steep slope in downward direction. This sudden reduction of RRC due to inflation pressure is due to wheel becoming rigid. The pressure is directly proportional to the temperature; hence RRC of LRR tyre as a result of SRM surface temperature worked in favour of the EV.

3.6 Valve selection

The valve is an important part within the wheel, as it is used to inflate the tyre. The valve selection was essential to ensure the appropriate clearances from the motor and the rim envelope. Hence, the TR 540 series valve was used, since it offered the bent profile and a long sleeve for a leverage. The bending allowed easy fitment of the motor envelope and the long sleeve extended it from the rim end cap enabling easy tyre inflation. The positive wheel endcap allowed sufficient space for the inclusion of the motor as well as providing protection from the water and the dust. Figure 3.24a, depicts the final valve TR 540 dimensional details and Figure 3.24b depicts the final valve geometric details. The valve provides a clearance of 2 mm with the motor and the rim envelope. The valve bent profile is flexible to carry the additional ±1 mm dimensional variances. The final wheel prototype was constructed at the local supplier.

The rim used aluminium 6061-T6 spun material and was seam welded to an end cap. The prototype established the motor envelope space, the wheel weight, dimensions
variations and manufacturing defects. The dimensions were within drawing tolerances and there were no manufacturing defects observed.

Figure 3.24: Valve design-a) dimensional details, b) geometry details

3.7 Summary

In this chapter, the vehicle selection, the rim design and the tyre selection were achieved for the proposed in-wheel SRM. The salient points of this chapter are summarised below:

- The vehicle selection for the motor envelope was examined in this chapter. This was achieved by comparing ten cars in two sizes—small, and medium. Initial power to weight ratio studies indicated that, amongst the cars, Holden Barina Spark, Toyota Yaris, Suzuki SX4, and Suzuki Swift were ideal. Further, these selected cars were compared for mudguard clearances by measuring physical dimensions at vehicle distributors. The Holden Barina Spark provided the required space and appropriate clearances for the R17 wheel. The Holden Barina Spark was selected based on the motor power ratio, and appropriate clearances.

- To overcome physical measurement limitations, the Holden Barina Spark was digitised using the 3D scanner. The objective of digitisation was to accurately evaluate the space required with appropriate clearances for the rim-tyre and the brake system envelopes. The digital model established: i) radial clearances exceeding 10mm at any given point on the 3D complex mudguards fitted with the 1” wheel, ii) the available wheel space as 6 m m diameter and 220mm width, and iii) ~70mm clearance available for the brake system envelope.

- The rim tyre nomenclature was selected as compliance with: i) Rims and Tyre standards-Australia 2010, ii) National code of practice for light vehicle construction and modification-2011, and iii) Vic roads modification guides 2011. Using these
The limits and the earlier mudguard clearances as a basis, the 205/50/R17 rim-tyre was selected.

- Five rims were designed based on the end cap location and topology construction. The rims designed were: i) positive offset, single piece solid rim end cap, ii) zero offset, single piece solid rim end cap, iii) positive offset, two piece hollow end cap, iv) positive offset, single piece five support end cap, and v) positive offset, single piece three support end cap.

- Rim material selection was based on comparing: i) steel, ii) aluminium die casted, iii) carbon fibre, iv) magnesium, and v) aluminium spun materials. These materials were compared for five main criteria: i) the structural rigidity, ii) the thermal stability, iii) manufacturing defects, iv) the weight, and v) the cost. The aluminium spun 6061-T6 alloy rim was selected as it has a good structural rigidity (340MPa), the thermal stability and a less porosity due to the spinning process. It was also relatively cheaper than carbon fibre and magnesium wheels.

- The key factors affecting rim were weight, rigidity, thermal stability, and durability. The weight optimisation was conducted on five rim designs using FE methods, based on criteria of structural rigidity, thermal stability, and fatigue life assessment. Among the five rims compared in the optimisation process, the two piece hollow end cap, rim 3, was selected as it had 7kg optimal weight, 164MPa maximum Von-mises stress concentration, 0.3mm maximum deformation, 1.5 FOS and 0.02 maximum strain. Also, rim 3 had 3,364,600 fatigue life exceeding SAE J2530 standard minimum life cycle requirement.

- Low rolling resistance increased the EV range. The RRC at different car speed was examined for LRR and NRR tyres. The RRC increased with speed increase. The comparisons concluded that the NRR tyre was less efficient than the LRR tyre (with 0.08 at 60km/h to 0.12 at 100km/h). The effect of the inflation pressure on the RRC was established using the SAE J670e empirical formula. The RRC decreased with the tyre inflation pressure increase (limited to the maximum allowable tyre pressure).

- Finally, the TR 540 valve was selected with the bent profile and the long sleeve. The long sleeve bent profile valve provided easy access for inflating the tyre while retaining sufficient clearance from the rim-motor envelope. The corresponding rim-tyre prototype determined the dimensional and manufacturing stability.
Chapter 4

Motor design for EV drivetrain

4.1 Chapter overview

In this chapter, the motor design is completed in two stages- i) the concept design and ii) the detailed design. The initial sections of this chapter describe the development of the in-wheel SRM concepts. The motor concepts were compared to select an appropriate concept based on the magnetic path mounting with the objective of space utilisation and air gap retention. The later section of the chapter emphasises the detailed design developed using the selected conceptual proposal. The motor detailed design covers the appropriate packaging of the shaft, the hub and the motor cover with the magnetic path. The following points describe the key objectives of this chapter in more detail:

- To develop the free hand sketches for imprinting ideas, to develop product artefacts, and to compare alternative SRM concepts. The sketch books and pencils were used to convert the magnetic path to thinking sketches for SRM concepts. The magnetic path consisted of motor rotor and stator components. To evaluate motor concept based on: i) the space utilisation, where its maximisation led to increased motor power density and then ii) optimisation of magnetic path orientation to retain an air gap of 1mm. The scope of this research was limited to packaging to achieve the high power density motor.

- To select an appropriate motor concept by using the VR based schematic sketching method. The VR tools enabled the motor concept investigation and selection based on: i) the maximised space utilisation, ii) the optimised magnetic path for stabilising 1mm gap (clearance studies) and iii) ease of the fitment and the assembly.

- To derive key magnetic path dimensional details from the finalised motor concept: i) sizes (diameter, thickness, axial lengths, width, and area) ii) air gaps (between magnetic path), and iii) poles (numbers and sizes) for the detailed motor design.

- To develop a detailed motor design using the finalised concept and the magnetic path dimensional details. The motor design achieved key objectives of: i) maximising the magnetic path space, ii) minimising deflections, iii) optimising weight, iv) manufacturability/assembly, and v) vent design/sealing.
• To select appropriate bearings, based on the required axial and radial loads withstanding capabilities. The bearings separated the rotating (rotors) and non-rotating (stators) parts within the motor. The two bearings were required in this design due to extended shaft length and as a consequence an appropriate balance of bearing loads was a prerequisite for safe handling of the vehicle.

• To minimise deflections within the motor components. The crucial 1mm air gap within the magnetic path was important for motor power density. Consequently, mechanical stiffness of shaft and all motor components were optimised to minimise deflections within the assembly.

4.2 Motor conceptual design

The concept developed in this research utilised the following two methods: i) free hand sketching for the preliminary conceptualisation, and ii) VR tools for an appropriate concept selection. Initially concept sketches were made based on the magnetics from the CSIRO/Auto CRC consortium. Then these concepts were compared using VR based schematic sketching methods to evaluate the space utilisation, the magnetic path orientation, and the assembly/fitment criteria. The final concept was identified for the detailed motor design.

4.2.1 Preliminary free hand sketches

The importance of free hand sketching for ideation and concept development in engineering design is extensively documented (Rodgers, Green et al. 2000, Company, Contero et al. 2009). The paper pencil based sketches benefit by communicating the ideas cheaply and quickly. In this section motor magnetic preliminary sketches were developed based on magnetic path supplied by the CSIRO/Auto CRC consortium. The preliminary hand sketches developed a creative orientation strategy for the magnetic path. The magnetic path, in this context, refers to the rotor-stator arrangement and the manner in which they are mounted inside the motor. The rotor is a rotating and stator is non-rotating part within the SRM. Other factors influencing the free hand sketches were the number of rotors and stators and the dimensional details. The proposed concepts were then compared based on the maximised space utilisation for the magnetic path. Using the magnetic path orientation preliminary free hand sketches were created; examples are shown in Figure 4.1 and Figure 4.2.
Initially, two sketches were made, in order to assess magnetic path orientation. Based on the magnetic path orientation, two detailed concepts were developed—horizontal and vertical. **Figure 4.1** and **Figure 4.2** represent the sketches of two rotors and a stator orientation in a way that provides the axial gaps required. To reduce the complexity...
initial sketches only were represented two rotors and a stator orientation inside the wheel envelope. *Figure 4.1* represented the horizontal concept, where the ring shaped rotor and stator were oriented concentric to each other providing a radial flux. *Figure 4.2* represented the vertical concept, where the plate shaped rotor and stator were oriented face to face with each other providing an axial flux. These ideas were subsequently presented in focus group meetings attended by CSIRO, Auto CRC, and VPAC representatives and University peers to effectively demonstrate mounting of magnetic path inside the motor.

*Figure 4.3:* Detailed half section of horizontal hand sketch concept proposal (NTS)

A further detailed hand sketch of a horizontal concept was developed; the construction is illustrated in *Figure 4.3*. Most commercial electric motors use this approach since they are not limited by space utilisation as is the case with an in-wheel SRM. In this concept, the stator was sandwiched between two concentric rotors. A stator was connected to chassis by a knuckle and rotors were attached to the rims through bolts. This concept had several drawbacks, as listed below:

- The main issue was bolting the rotating parts and non-rotating parts separately without sacrificing the space. Separating both rotor and stator required complex bearing and knuckle arrangement with an additional space. Space utilisation in this design was limited for magnetic path due to this complexity.

- The overhanging stator lamination tooth was a practical manufacturing limitation. As the stator is subject to induced magnetic flux, the maximum stator
tooth length was limited to 20-25mm. However this concept had more than 100mm stator laminations tooth length due to overhand by virtue of the design.

![Detailed half sectional vertical hand sketch concept proposal (NTS)](image)

**Figure 4.4: Detailed half sectional vertical hand sketch concept proposal (NTS)**

A detailed hand sketch of a vertical concept was developed, illustrated in *Figure 4.4*. As is shown, this concept provided more practical ways of attaching rotating and non-rotating parts by using a taper roller and a single row ball bearing inside the motor. This concept provided an opportunity to maximise space by attaching the rotor and stator directly to the rim. The rim has been a part of the SRM, whereby a detachment plate was designed to achieve easy detachment. The rotor and stator have been compacted as subassemblies, which simplified the overall concept. The stator and rotor subassemblies have been split to connect one with the rotating rim and other with the stationary shaft. This arrangement also facilitated effective segregation of rotor and stator parts. This allowed shortened stator tooth length without compromising the space utilisation. This further facilitated the opportunity for easy fitment of related subassemblies such as coil winding and forced cooling. This arrangement consisted of a hollow shaft connected to the trailing arm holding the stationary stator, with insulation placed in between. Two rotors were designed, which were mounted on the motor cover coupled directly to the rim.
4.2.2 Concept evaluation

Although the preliminary hand sketches indicated that the vertical concept was more efficient, further comparisons of both concepts were conducted using VR based tools. The VR based schematic sketching method was used based on the following four objectives:

- Maximum motor power density achieved by optimal space utilisation inside the wheel
- Optimisation of magnetic path orientation inside the motor
- Ease of assembly and sequencing for dismantling of the wheel and motor components for servicing
- Ease of manufacturability to achieve crucial 1mm air gap, thus increasing power density

The evaluations performed at this stage were based on the animation performed using schematic sketching methods using VR based graphics. These methods consisted of the frame arrangement that, if viewed in quick succession, gave the illusion of a moving picture. These are also termed as 3D sketching techniques and have advantages in evolving solutions for conceptual designs (Rahimian and Ibrahim 2011). These tools facilitated creation of 2D sketch motions, thus allowing verification of assembly techniques. Thereby, these tools were used to maximise the utilisation of the space available inside the motor. Initially, the sketches were created from vector-based curves to construct blocks. These sketches represented the outer 2D shape of the motor parts and the related components. Each block was then linked to other blocks, representing different items, such as stators and rotors in the motor. Sets of design animations were then used to define assembly protocol, which evaluated the concepts based on the objectives outlined above. The sketches provided detailed insight into the design themes and demonstrated the effectiveness of the assembling function. The demonstrations were used in focus group meetings and in consultations with multiple groups. These invaluable peer discussions provided an opportunity to fine tune and improve the SRM concepts. Moreover, the sketches enabled the assessment of the assembly function to maximise the space and achieve crucial air gap inside the motor. The aim was to represent each part as a 3D model for conceptualisation; however, as this was impractical and time consuming for creating a few series of concepts, schematic sketching was adopted in this stage. Using this approach, it was easy to construct the
alternatives in a relatively short time, saving costs and drastically reducing the design cycle of the SRM. Another advantage of the schematic conceptualisation method conceived in this research was that the sketches initially used for conceptualisation were converted into 3D parts, thus avoiding duplication of work. These sketches also offered parametric associativity and interpolation with other parts generated as blocks. These processes saved substantial time in conceptualisation of the stator-rotor and overall packaging for the in-wheel SRM. Schematic methods also provided the opportunity for visualising the final model, establishing parametric relations, interference detection and clearance analysis, which further enhanced manufacturability.

The sketches were designed based on the required dimensions and converted into blocks. Sketches were an integral part of models, allowing establishment of required associations while avoiding ambiguity in models by applying appropriate constraints. These sketched blocks were constrained with respect to relative parts providing required motions. However, the schematic sketching also suffers from some limitations, one of which is 2D presentation only, which consequently disregards the other directions (e.g., if you working in XY plane, it doesn’t tackle Y and Z directions). Applying scenes, backgrounds, textures and other elements is also not possible in the 2D environment. Consequently, model explodes and fly throughs were difficult to manage in the 2D interactive mode. Generally, 2D schematics are based on a single camera system, unlike 3D, where widgets, hoover or style cam techniques can be employed. Once the sketched blocks were constrained and path was defined for animation, functional visualisation of SRM was achieved.

For the motor conceptualisation, two proposals—horizontal and vertical—were created, as shown in Figure 4.5 and Figure 4.6. These concepts were evaluated, based on their respective ease of assembly, maximum space utilisation, optimisation of magnetic path to enhance the motor power density, and ease of manufacture so as to maintain the crucial air gap of 1mm.
Animations were created to analyse the assembly sequencing and ease of manufacture. The horizontal concept shown in Figure 4.5 consisted of a rotary rotor and overhanging stators separated by bearings. In this concept, the shaft was rotating, which is typical for most currently available commercial motors. In the horizontal motor concept, the required space was achieved with diametrical positioning of rotor-stators, as is typically done in regular motors. However, due to the shape of stators and rotors within the motor, segregation of rotating and stationary parts within the motor assembly was barely possible without compromising the available space. Assembly was also a challenge, as detachment of the motor parts and wheel became impossible, since all the parts were rotary referencing. Magnetic path dimension was an issue too, as overhang of stator without any additional support could not be manufactured.

Thus, the design moved onto the vertical motor concept shown in Figure 4.6, which had plate-like rotors attached to the motor cover, maximising the utilisation of wheel space. The rotors were linked through bearing to the stationary shaft, thus easily segregating rotating and non-rotating parts within the motor. The stator plates were sandwiched between rotors connected by a stationary shaft, which is different to the design of other currently available electric motors. The vertical motor concept thus provided more effective space utilisation as well as enabling ease of manufacturing. Space utilisation...
resulted in the increased power density due to the increased magnetic path area. The rotor stators could be packed separately, thus segregating rotating and non-rotating parts within the assembly. Next, the required conceptual subassemblies were subjected to interference detection tests, and the parts where interference was identified were refined in the assembly context. This analysis evaluated the crucial 1mm air gap within the magnetic path by assembly sequencing. Other design automation tools, for example dimension expert used for tolerance analysis, worked well with the schematic sketching methods. This approach was very time and cost efficient, as the dimensions, tolerances, and manufacturing features, were auto-inserted, once primary, secondary or tertiary planes were identified. These tolerances for stacking and alignment of stators and rotors resulted in achieving the crucial 1mm air gap within the SRM concept.

**Figure 4.6: Vertical mounting concept using VR based visualisation**

The comparison was performed on a scale of poor to high ratings for both motor designs, as shown in Table 4.1. The concepts were compared with respect to the previously identified key objectives, of space, manufacturability, assembly and magnetic path optimisation. The vertical concept met the objectives, and was thus selected for further detailed design.

**Table 4.1: Comparison of motor concepts**

<table>
<thead>
<tr>
<th>Motor Concept</th>
<th>Space Utilisation</th>
<th>Ease of manufacturability</th>
<th>Ease of assembly and detachment</th>
<th>Magnetic path optimisation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal</td>
<td>Medium</td>
<td>Poor</td>
<td>Poor</td>
<td>Medium</td>
</tr>
<tr>
<td>Vertical</td>
<td>High</td>
<td>High</td>
<td>High</td>
<td>High</td>
</tr>
</tbody>
</table>
The schematic sketching methods includes a series of animations that helped finalise the concept. Upon examination, it was evident that the following aims were accomplished during the motor concept evaluations:

- Comparison of two motor concepts led to the concept selection and the concept was finalised based on the maximum space, number of stators, rotors, assembly, and fitment. These choices resulted in increasing motor power density due to the achieved magnetic path optimisation and orientation inside the SRM.
- The assembly was refined by sequencing of rotor, stators, shaft, and wheel, thereby achieving easy detachment of individual parts. This further provided correct assembly sequencing and optimal space utilisation for the SRM.
- An analysis for clearance and tolerance on the motor related components allowed establishing ease of fitment and manufacturability by maintaining 1mm crucial gap.
- Sketches employed in this method were used for further 3D model development, avoiding duplication of work and saved substantial time and costs of design process.

### 4.2.3 Magnetic path details

Based on the finalised vertical motor concept, physical dimensions of magnetic path were determined. As mentioned in chapter 2, the SRM works on exciting the stator pole intermittently with respect to its next stator pole, to achieve required rotation between two phases. The transition from one phase to another occurs within microseconds. This in sequence results in undesirable noise especially in low phase mode as the pulse control and feedbacks becomes crucial. The undesirable noise is minimised by the following activities in the design:

- The air gap (as explained before) and structural stability within SRM magnetic path governs vibrations. Hence optimising these for reducing noises. E.g., appropriate stator/rotor thickness, motor covers with structural strength (since motor covers acts as structural support for magnetic path, the rigidity of these covers becomes crucial). The other method is by controlling the current pulse profile by digital signal processor using insulated gate bipolar transistor (this method is not in the scope of this thesis).
Table 4.2: Magnetic path details for SRM concept

<table>
<thead>
<tr>
<th>S.no.</th>
<th>Description</th>
<th>Formulas</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Motor outside diameter (mm)</td>
<td>$M_O$</td>
<td>418</td>
</tr>
<tr>
<td>2</td>
<td>Motor inside diameter (mm)</td>
<td>$M_I$</td>
<td>196</td>
</tr>
<tr>
<td>3</td>
<td>Motor length (mm)</td>
<td>$M_L$</td>
<td>194</td>
</tr>
<tr>
<td>4</td>
<td>Stator assembly length (mm)</td>
<td>$S_L$</td>
<td>68</td>
</tr>
<tr>
<td>5</td>
<td>Stator back iron length (mm)</td>
<td>$S_B$</td>
<td>10</td>
</tr>
<tr>
<td>6</td>
<td>Total centre rotor length (mm)</td>
<td>$CR_L$</td>
<td>24</td>
</tr>
<tr>
<td>7</td>
<td>Rotor length/width ratio (mm)</td>
<td>$R_{L/W} = (L/W)$</td>
<td>0.75</td>
</tr>
<tr>
<td>8</td>
<td>Rotor back iron length/width ratio (mm)</td>
<td>$R_{LBW} = (L_B/W)$</td>
<td>0.5</td>
</tr>
<tr>
<td>9</td>
<td>Number of poles (nos)</td>
<td>$N_P$</td>
<td>32</td>
</tr>
<tr>
<td>10</td>
<td>Axial length-air gap (mm)</td>
<td>$A_{GI}$</td>
<td>2</td>
</tr>
<tr>
<td>11</td>
<td>Axial length-wedge (slot mm)</td>
<td>$A_{IW}$</td>
<td>2</td>
</tr>
<tr>
<td>12</td>
<td>Axial length-insulation(slot mm)</td>
<td>$A_I$</td>
<td>2</td>
</tr>
<tr>
<td>13</td>
<td>Thickness-insulation (slot mm)</td>
<td>$T_I$</td>
<td>0.25</td>
</tr>
<tr>
<td>14</td>
<td>Coil fill factor (copper mm)</td>
<td>$C_F$</td>
<td>0.85</td>
</tr>
<tr>
<td>15</td>
<td>Linear width of tooth (mm)</td>
<td>$W_T = 2((\sin(3\pi/N_P)*(M_O+M_I))/4)$</td>
<td>10.04</td>
</tr>
<tr>
<td>16</td>
<td>Axial length of rotor (mm)</td>
<td>$L_R = W_T*(R_{LW}+R_{LBW})$</td>
<td>12.56</td>
</tr>
<tr>
<td>17</td>
<td>Axial length of winding (mm)</td>
<td>$L_W = (M_L-(2*(L_R+A_{GI})+A_{IW})+(2*T_{GI})-S_B)/2$</td>
<td>26.44</td>
</tr>
<tr>
<td>18</td>
<td>Length of coil per turn (mm)</td>
<td>$L_C = ((\pi*(M_O+M_I))/N_P*(3/2))+(2*(M_O-M_I))$</td>
<td>262.19</td>
</tr>
<tr>
<td>19</td>
<td>Area of coil (mm²)</td>
<td>$A_C = ((W_T/2)-(2*T_{GI})*L_{WB})*C_F$</td>
<td>101.65</td>
</tr>
<tr>
<td>20</td>
<td>Lamination angles (degree)</td>
<td>$L_{A} = 180/N_P$</td>
<td>5.63</td>
</tr>
<tr>
<td>21</td>
<td>Stator/rotor outside diameter (mm)</td>
<td>$SR_{O} = (M_{O}+2*(A_{GI}+W_T))$</td>
<td>389.91</td>
</tr>
<tr>
<td>22</td>
<td>Stator/rotor inside diameter (mm)</td>
<td>$SR_{I} = (M_I+2*(A_I+W_T))$</td>
<td>220.09</td>
</tr>
</tbody>
</table>

Hence, structurally thick rotor and stator dimensions were an important part of mechanical design to minimise the vibrations and noise. SRM used stator steel laminations with copper windings and rotors were steel laminations without any copper windings. The stator rotor pole numbers were the same in this design. In this design, stators and rotors produced using high quality electrical steel of grade M15 (non-oriented 2.7% silicon), was used (ASTM International 2010). The silicon steel laminations were annealed to achieve high stress bearing characteristics. Generally electron beam welding is used to for holding silicon steel laminations, which was the case in this research. In some cases glue technique is preferred; however it is costly and also results in light weight. The laminations and insulations were required to withstand:
i) electromagnetic forces, ii) thermal loads, and iii) mechanical forces. Hence high grade insulation is essential, typically i) Teflon, ii) Kapton, iii) Nomex, iv) silicon and iv) glass fibres are used. Varnish epoxies were used for copper windings to provide: i) dielectric strength, ii) high bonding, iii) high thermal resistance, and iv) corrosion resistance.

The rotor stator geometry from a vertical concept derived the magnetic path dimensions. SRM outside diameter was dependent on the selected P205/50/R17 rim size. Based on the finalised sketch the outside motor diameter, motor length, stator lengths, rotor lengths, rotor back iron details, centre rotor length, number of poles, axial lengths (air gap, wedge, and insulation), insulation thickness, and copper fill factor were determined. Then linear widths of tooth, axial length of rotor/winding, and length/area of coil per turn, lamination angle and stator outside/inside diameters were determined. Table 4.2 above represents the magnetic path dimensional details for SRM. These dimensions were derived based on the initial sketch; however these were fine-tuned further during final design. There are expected variations of 5-10% from these dimensions to final design.

### 4.3 Motor detailed design

In contrast to conventional PMMs that are typically used in the existing EV drivetrains, the SRM was selected in this research. This choice was made due to the use of non-rare earth elements for its components, simple construction, reliability and its suitability in operations under heavy load (e.g., vehicle rides). Nonetheless, the key advantages of the SRM include high power density when designed properly with a robust construction, and higher heat withstanding capabilities. Moreover, due to absence of gears, transmission loss is minimal for the selected in-wheel SRM. A detailed motor design was developed in this study based on the vertical concept finalised in the last section. Compared to PMM, SRMs need smaller air gaps within the magnetic path. To maximise power density from a mechanical approach, two factors became vital in the final design process: i) minimising the motor parts weight without compromising structural rigidity, and ii) retaining the air gap within the magnetic path, and as a consequence reducing deflections within motor parts.

Therefore, optimisation of motor stiffness to minimise weight became crucial. The bearing was designed to achieve effective load distribution and to minimise deflections.
Additionally shaft and overall assembly was optimised to minimise deflections. Thus, the motor design was developed in the following stages:

- Initial or envelope sizes were based on the old Barina Spark with a R1” wheel. Based on the concept specifications, the motor packaging was developed in a staged manner with a main objective of the high power density.
- The hub design was optimised to achieve load distribution (thrust handling) of bearings thus minimising deflections within the assembly and increasing the motor power density by retaining the crucial air gap of 1mm.
- The stability of the crucial air gap of 1mm between rotors and stators was evaluated by designing a shaft and subjecting it to a FE based deflection analysis. This was achieved by designing a hollow shaft (to minimise weight) with a larger diameter (to maximise stiffness). Further FE study on the overall motor assembly was performed to examine the deflections at magnetic path using 5kN load case.

### 4.3.1 Motor packaging

Following the conceptual stages, the initial sketch based on the finalised vertical conceptual design was further investigated. The motor design progressed in three stages and each stage was an improvement on the last one. These design stages were developed to mechanically package the motor by the following key objectives to achieve the high power density:

- Maximise the space utilisation- As an in-wheel SRM, the space utilisation was key criteria for maximising the power density. This was achieved by utilisation maximum space for magnetic path.
- Optimise the weight- The design goal was to minimise the deflections and maximise the structural rigidity, whilst optimising the motor weight. Basic inexpensive lightweight materials were used and topology optimisation was utilised. In topology optimisation, shapes and cross sections were optimised using FE methods.
- Design for assembly and manufacturability- The key elements for successful motor development; hence design was focussed to optimise the assembly and manufacturability aspects.
• Design for ingress protection (IP)- Thermal management, appropriate vents and sealing design to meet standards was also a consideration during the design process.

4.3.1.1 Stage one motor design

Using the vertical concept sketches the 3D models were created for further refinement. The stage one motor design front half sectional view is shown in Figure 4.7. The motor design consisted of a single stator with copper winding on either side, which was supported by a shaft through fibre glass mounted aluminium bushes. In the chosen design, two rotors were used on either end of the motor cover and were connected to the rim held by bolts. The motor design consisted of two main elements: i) rotating and ii) non-rotating parts, i.e. rotors and stators. Two bearings were designed on either end to support the motor covers and to keep the rotating and non-rotating parts separated. This bearing selection was a challenge as load exerted by the vehicle during cornering, turning and in any other difficult and challenging driving conditions needed management. SRMs with the chosen 1mm crucial air gap between the rotors and stators required tight tolerance for manufacturing and hence shaft design was crucial. Important factors of avoiding any water leakages inside the motor and provision for heat dissipation from the stators, brake calipers, and easy assembly for maintenance activities, were also considered.

This design had several issues identified during this process: i) bulky weight, ii) suboptimal space utilisation (due to bulkiness of the rotor/stators), and iii) assembly and manufacturing aspects (large stators and rotors were heavy to manufacture and assemble). Several potential solutions were identified in the design stage as shown in Figure 4.7. Such examples were:

• Converting motor cover shape to suite the rim outer profile, thus providing more space for the magnetic path and reduce the weight of motor cover (discussed in chapter 5). Reduce motor covers thickness by having smaller thickness covers supported by ribbing (discussed in chapter 5)

• Splitting the solid section of stators and rotors into smaller sizes (increase numbers) for ease of manufacturing and handling (discussed in chapter 5)
Replacing aluminium support at ends and using fibre glass material (class H insulation grade discussed in chapter 2) with more thickness (0.04 m-C, low thermal conductivity and very light weight)

- Increasing diameter of the shaft and using a hollow section; thus achieving required stiffness while minimising the weight

![Figure 4.7: Stage one motor design half front sectional view](image)

4.3.1.2 Stage two motor design

*Figure 4.8* shows the stage two motor design with the full front sectional view, which had three rotors and two stators (like two back-to-back motors connected together). The following were the key accomplishments of this design:

- Motor cover shape was matched to the rim profile, allowing an increased space for the magnetic path.
- To increase the motor power density by maximising space utilisation, another central rotor was added and as consequence the motor cover was split in the centre.
- The single stator was split into two stator packs positioned on either end. This modification improved the magnetic path of the motor, thus increasing its power density. Rotors and stators were packed as separate subassemblies within the motor to ease the assembling process (discussed in chapter 5, section 5.3).
- Replace aluminium supports at ends were replaced with fibre glass material
- Shaft diameter was increased and used hollow section

During this design stage several issues were identified and some examples were:

- Most of the cross sections were bulky (e.g., covers and bushes)
- Rotor and stator sections were not completely optimised for space utilisation
- Shaft and bearing designs needed optimisation
- Fibre glass was slender in the initial design stages, and as a consequence needed extra cross section to support the stator weight.

**Figure 4.8:** Stage two motor design with full front sectional view

### 4.3.1.3 Final motor design

The final motor design full sectional view is shown in *Figure 4.9* and further details are provided in *Appendix 5*, which also includes the bill of material (BOM) conceived during this research. Some parallel work was carried out, which is a general practice followed in the engineering design. Such examples are: bearing calculations, motor covers optimisation, space optimisation for magnetic path, motor assembly/manufacturing optimisation, vent design, and shaft design. However for the sake of completeness these optimisations are arranged after the final design.
The final motor design is shown in Figure 4.9 and the following were the key accomplishments of the motor packaging:

- Maximising the space utilisation– The entire available space was utilised by converting the motor outer shape to the rim envelope (discussed in chapter 5). Each of the rotor and stator were packed to form a subassembly. The stator and rotor packages were also fine-tuned for ease of assembling (discussed in chapter 5). One rotor was placed in the centre and two other rotors were held between two motor covers, which were the rotating part of the motor. The two rotors on either side were held by motor covers firmly attached through bolts. The bearings on either end provided support and acted as a buffer between the rotating and non-rotating parts of the motor. One bearing was moved to an aluminium bush from a shaft at one end to provide more flexibility during the vehicle cornering and maximise the space utilisation. A crucial gap of 1mm was retained between the motor stators and rotors. The motor was designed with a crucial air gap of 1mm, thus achieving maximum power density. Finally the sensor orientation within the motor was finalised. This is an essential part for SRMs, to accurately feedback the controller for exact speed, rotor position and torque required (as discussed in chapter 2). The sensor, which indexed the number of rotations to controller, was held with support of fibre glass on the other end of the motor cover, ensuring that the motor is being controlled by a super controller planted inside the vehicle. The sensor was designed to send a signal of motor failure, thus controlling the rotation of both motors via a feedback loop system.

- Optimising the weight– Unlike PMMs, SRMs used abundant materials (e.g., steel, copper and aluminium), which replaced rare earth elements. Since the SRM used steel, the overall weight of the motor needed to be optimized. Two approaches were used, by using basic lightweight materials (e.g., aluminium, magnesium alloys and fibre glass were used to manufacture all motor parts apart from magnetic components) and topology optimisation (e.g., shaft, motor cover, rim, and disc brakes topology optimisation explained in next chapters/sections). Most of the parts were designed as hollow structures with bigger sizes, whereby ribs and protrusions were supplemented where necessary to increase stiffness, while retaining the low weight of the motor, without compromising the crucial 1 mm air
gap. For example, motor covers were designed with 3 mm thickness and 3 mm ribs were added for extra stiffness (explained in chapter 5).

- Designing for assembly and manufacturability– A disc brake was attached to the motor cover from inside, while the rim was attached to motor cover via M12 bolts and spacers, which allowed for easy disassembly during servicing (explained in chapter 6). The rim was designed as an integral part of the motor where an easy detachment was established (explained in chapter 8). Each of the rotor and stators were packed to form a subassembly. This not only achieved easy assembly sequencing but also made the entire design modular. Manufacturability by increasing the tooth length of rotor further improved the power generation capacity. Additionally tolerance stacking and VR based simulations were conducted to refine the motor assembly for assembly optimisation and space utilisation.

- Vent and seal design– Forced fan cooling was introduced (discussed in chapter 5) and venting was provided to allow air circulation for cooling high temperature stators (compliance to IEC 60034-6 discussed in chapter 2). The shaft was hollowed for channelling wire harness and an air cool pipe was inserted, ensuring that the temperature of stators was within the optimal range. The motor covers housed two non-return valves to allow air flow generated by the forced cooling of the stators. The motor consisted of two stators positioned on a hollow shaft through aluminium bushes held by fibre glass (0.04 m-C, low thermal conductivity and very light weight) to improve the insulation of the motor. Fibre glass material offered two important aspects: i) low thermal conductivity (0.04 m-C) and very light weight (compliance to H category of IEEE 11-2000 standard discussed in chapter 2). An O-ring was attached on top of the central rotor holder, which acted as a barrier between the two motor covers, preventing any water leakage into motor during the vehicle ride (ensuring compliance with IP 56). A gland seal and protector plate was attached on end of the shaft for IP 56 protection (compliance to IEC 60034-5, discussed in chapter 2).
4.3.2 Bearing selection

Hub design in a typical electric motor consists of shaft, bearings and motor covers. In this hub design, the shaft was stationary and the stators were attached through the bush held on a shaft with spline arrangements. The motor covers with rotors rotated and the bearings were used to separate non-rotating and rotating parts within motor. These also minimised friction between the parts and improved the motor efficiency. A comprehensive bearing analysis was conducted for balancing and distributing vehicle loads for increasing power density while retaining the crucial 1 mm gap with the following objectives:

- Stabilising of deflections in order to achieve optimal motor performance (stabilise air gap).
- Reducing of noise levels to required 105dBA and vibrations less than 7.6mm/s from 4m at rated speed as per IEEE 11-2000 standard discussed in chapter 2 by selecting an appropriate bearing.
- Optimising the bearing load distribution (up thrust and low thrust handling) to ensure the vehicle ride comfort.
- Assessing bearing life cycle in ascertaining the safety of the motor.

As bearings play a vital role in load distributions and the vehicle ride comfort, their analysis was crucial for development of electric motor design. Figure 4.10 shows the bearing selection process used for hub design in this research. Bearing selection was based on: i) functional performance, ii) grease and sealants, iii) thermal management, iv) life cycle and v) mounting within the motor.

Functional performance dictated bearing required on each end based on bearing capacity to carry the required loads. In this design, bearings were used to connect the rotating motor covers and the stationery shaft, leading to reduced friction and minimised wear failures. In this case, two bearings were required for each hub, providing two supports at different ends of the shaft while the motor covers rotates around it. The first of the two bearings was positioned inside the hub and acted as the main load carrier, bearing high radial and axial loads. The second bearing—known as the outer bearing—was situated at the tip of the hub and acted as a support bearing to the more dynamic end of the shaft. Bearings can be classified based on: i) shape of the bearing (ball, roller type, or spherical) ii) position/contact of bearing inside the cage (deep groove or angular), iii) number of rows for bearing (single or double row), iv) lubrication (oil or grease), and v) protection from dust and water (lip or cage sealants). The roller bearing has more load bearing capacity than the ball or spherical type. Deep groove would allow axial and radial tolerance, thus tolerating any small deflections within overall assembly. This is typically used for load transfer application when tandem bearing arrangement is used. Increase in number of rows increases load bearing capacity of bearing. Grease type is popular in automotive applications; wet greasing is increasingly used for concealed bearings. Sealants determine the dust/water restriction and are vital for ensuring bearing performances. The outer bearing thus needed to be designed to be more dynamic, light and able to withstand radial loads and transform axial loads to inner bearing. Two bearings were selected:
1) Double tapered roller bearing- This uses roller bearing in its construction and hence is robust enough to carry both high radial and axial loads. This is used inside the shaft to carry most of the load exerted by the vehicle.

2) Deep groove ball bearing- Deep groove ball bearing was designed with clearances inside the bearing itself. This facilitates in minimising the deflections by allowing tolerance variations up to 0.02mm radially and axially. This means that load exerted on one end of the vehicle is transferred across to the other bearing.

As the bearings were sealed, wet grease (Lithium/Molybdenum sulphide grease) was used in this design. Double lip sealants were used to avoid any unwanted dust or water entering the bearings. Thermal management of the bearings is also important, due to extensive thermal radiation from the motor parts (stators). The earlier literature on SRM has shown that there is significant temperature difference between the stator and the shaft. Since the motor is designed to class B with Class H insulation (IEEE 11-2000 standard) the stationary winding temperature is expected to reach a maximum of 120°C (class B motor). The stator temperature never gets transmitted directly as insulation worked as a barrier. The inside motor temperature was 0°C with forced fan cooling (explained in chapter ). Hence, the bearings were selected based on the specific operating temperatures withstanding up to 0°C.

![Figure 4.10: Bearing selection process](image)

Each in-wheel motor contained two bearings—one for the inner and another for the
outer section of the shaft. Study on load distribution and mounting locations was essential to ensure sufficient life cycles and passenger safety. The analysis was conducted for selection of appropriate bearings capable of withstanding radial and axial loading exerted on both the outer and the inner shaft section during vehicle rides. This included the evaluation of rigidity, as well as the vehicle performance in harsh running environments. Unlike other hub designs, the challenge in the present research was the increased distance between two bearings and load distribution on either end.

**Figure 4.11: Motor tyre model for calculating bearing loads**

Based on the developed design, the following dimensions were taken as reference points and were used in the calculation of the bearing loads (*Figure 4.11*). From the schematic represented in *Figure 4.11*, the following dimensions were established: i) motor width, $l$ (195.2mm), ii) ride height, $R_H$ (241.3mm), and iii) half width, $a$ (98.5mm). Load on each wheel was based on the vehicle and the passenger weight with additional safety factor. An additional safety factor was to take into account rough operating conditions. As mentioned in FE methods for rims the vehicle weight was established as 1060kg and an additional 500kg was added to cover passengers and luggage. The gross weight then 1560kg was multiplied by a factor of 1.25, leading to a total load of 4.77kN (rounded to
5kN) on each wheel. This was used in the calculation of radial loads based on Equations 4.1 to 4.4 below:

\[ FR_{\text{inner}} = \varepsilon_1 K + \varepsilon_2 f.K \]  \hfill (4.1)

\[ FR_{\text{outer}} = (1 - \varepsilon_1) K + \varepsilon_2 f.K \]  \hfill (4.2)

\[ \varepsilon_1 = \frac{a}{l} \]  \hfill (4.3)

\[ \varepsilon_2 = \frac{R_{\text{H}}}{l} \]  \hfill (4.4)

In general practice for a commercial vehicle \( f \) is considered as 0.05, and \( \varepsilon_1 \) was calculated as 0.5046 and \( \varepsilon_2 \) as 1.236. From Equation 4.1 and Equation 4.2 the \( FR_{\text{inner}} \) was determined as 2.71kN and \( FR_{\text{outer}} \) was 2.08kN or 2.67kN respectively.

Taking a greater value as a referent centrifugal force, \( K_d \) was calculated using Equation 4.5. When the worst case scenario was considered, where the car taking its maximum wheel base right turn of 9.9metre at a velocity of 60km/h, \( K_d \) was given as:

\[ \left(\frac{K_d}{G}\right) = \frac{1}{127} \times \left(\frac{V}{r}\right) \]  \hfill (4.5)

Whereby, \( G \) was weight of the car (1060kg), \( V \) was velocity (16.67m/s) and \( r \) was turning radius (9.9m) therefore, \( K_d \) was calculated as 212.54. Thus forces acting on inner and outer bearing during the turn were calculated using Equation 4.6 and Equation 4.7:

\[ K_{\text{outer}} = \left[ 1 + 2(h/b) \left(\frac{K_d}{G}\right) \right] (5000) \]  \hfill (4.6)

\[ K_{\text{inner}} = \left[ 1 - 2(h/b) \left(\frac{K_d}{G}\right) \right] (5000) \]  \hfill (4.7)

![Figure 4.12: Wheelbase model for bearing calculations](image)

105
The following dimensions were taken directly from the vehicle: i) wheel diameter, $h$ (482.6mm) and ii) wheel base, $b$ (1414mm). Therefore, $K_{outer}$ is calculated as 5.52kN and $K_{inner}$ as 40.76kN. Hence, from Figure 4.12, axial loads ($K_{a\,outer}$ and $K_{a\,inner}$) acting on the bearings were calculated using Equation 4.8 and Equation 4.9:

$$K_{a\,outer} = (K_d/G) \left[ 1 + 2(h/b) \left( K_d/G \right) \right] K$$

$$K_{a\,inner} = (K_d/G) \left[ 1 - 2(h/b) \left( K_d/G \right) \right] K$$

Hence, $K_{a\,outer}$ was calculated as 1.22kN and $K_{a\,inner}$ as 0.98kN. The life of each bearing was determined by calculating the dynamic load rating, $C_r$ from Equation 4.10 as follows:

$$C_r = (f_h/f_n) \times (F_r)$$

Where, $f_h$ is the life factor of the bearing. In general, the passenger vehicle wheel bearing life expectancy is estimated at 21,700 hours. Thus, using this estimate and the life factor $f_h$ as 3.1 the following values were obtained: i) maximum velocity (1666.67 m.min$^{-1}$), ii) wheel diameter (0.48m), iii) wheel circumference (1.52m), iv) wheel rpm (1100rpm) and finally $f_n$ the speed factor and was calculated using the expression ($f_n=((33.3/rpm) 0.3)=0.3502$). Therefore, the dynamic load ratings for both bearings were calculated as: $C_{r\,outer}$ as 24.07kN and $C_{r\,inner}$ as 23.68kN.

This loading analysis ensured that bearing designs were safe at the roughest operating conditions. Based on the specifications given by the supplier, the recommended inner bearing (paired tapered roller bearing) can withstand $C_r$ of up to 105kN. Thus, the previously calculated value of 24.07kN was completely safe. Similarly, for the recommended outer bearing (deep groove ball bearing), the specifications indicated that it is capable of withstanding up to 27kN, which also indicated that the calculated value of 23.68kN was within the safe limits. Figure 4.13 below is a summary of loads calculated for the selected bearings to be used in the developed motor design.
Figure 4.13: Bearing loads for hub design of an EV drivetrain

The SRM hub was designed with two bearings positioned at either end, to allow load distribution and maintain a crucial gap of 1mm during the vehicle ride. One bearing was located in the inner vicinity of the shaft and designed to withstand high load characteristics. This inner bearing was very rigid and sturdy, with an extremely high reliability factor. Conversely, the other bearing was located at the outer section of the shaft and was dynamically controlled to allow radial and axial loads. This bearing was capable of withstanding any deflections in the shaft, occurring due to varying loads on the other end of the shaft. This was achieved by mounting the bearing on an aluminium bush with a small axial gap to allow small variations. The deep groove inside the bearing provided radial deflections. Calculations determined the maximum axial and radial loads acting on both bearings and, based on these results, bearings were selected from the range listed in SKF Bearings using the bearing data sheet (Appendix 6). Following data sheet two bearings were finalised: i) a tapered roller bearing paired face to face for inner wheel hub and ii) a single row deep groove ball bearing for outer hub. The bearing selected for the inner section was the double-row tapered roller bearing; more specifically, the BTHB329129ABA SKF bearing, as shown in Figure 4.14a. The more flexible single row deep groove ball bearing was selected for the outer bearing, as shown in Figure 4.14b; more specifically, the 6207 SKF bearing. According to the calculations, the bearings were completely safe to withstand the loads acting on the hub. Proper sealants and long-term lubrication bearings were adopted to ensure their prolonged usage. The key outcomes of the aforementioned bearing selection are summarised below:
• Design—Tapered roller paired face to face for inner wheel hub bearing and single row deep groove ball bearing for outer hub were selected. These bearings were validated for maximum loading conditions during the vehicle ride. More specifically, tapered roller bearing acted as main load-carrying bearing in the chosen motor design. Similarly, single row deep groove ball bearing was selected to allow radial deflections caused during the vehicle ride.

• Noise reduction—These bearings were mounted to distribute more load on the tapered roller bearing than on the deep groove ball bearing, which was achieved by preloading the tapered roller bearing, whereby the deep groove was relieved to achieve load transfer to the other end. Appropriate tolerance stacking resulted in preloading of the tapered roller and relief on the ball bearing. This minimised the friction and as a consequence resulted in reduction of noise level due to frictions.

• Design for assembly—The bearings were easy to assemble with outer shell rotating and inner shell stationary. The double row tapered roller bearing was mounted on the shaft and the single row deep groove was mounted on the aluminium bush, mounted on the shaft.

Figure 4.14: Bearing details: a) double row taper roller (inner) and b) deep groove (outer)

4.3.3 Shaft design
Shaft is an important part in this motor design, since it supported the entire motor structure and connected it to the vehicle. The motor shaft typically transmits power from the motor to the wheel. In this design, unlike other electric motors, the shaft was completely stationary and acted as the main hub support, allowing the SRM to rotate around the shaft through motor covers, rotating the wheels. In this research, the shaft was stationary as the motor casings were directly attached to the wheel, which turned the vehicle into motion. Although not transmitting power, as conventional shafts does,
its role in this particular SRM was still vital. The shaft played a critical role as part of the hub, providing structural support to the motor.

As shaft failures are common, their comprehensive analysis was an essential part of the design in the current study (Bonnett 1999, Ying, Wu et al. 2011). Shaft acts as the main backbone to the housing stators, whereby any deflections may compromise the crucial 1 mm air gap. Another important variable in this motor design is shaft hollowness, required as a relief for wire harness and providing the necessary weight reduction. The shaft was designed with larger diameter and hollow structure to maximise stiffness with a light weight design. Nonetheless, the analysis focusing on the hollow shaft strength was still critical and was conducted using two methods. Firstly, it was performed analytically, whereby the displacement, slope, moment and bending stress and shear force were determined. Next, FE methods were used to examine maximum Von-mises stress and maximum deformations. Using both methods, the results were compared and it was concluded that the shaft deformations did not compromise the crucial 1 mm air gap. In the motor design the outer bearing, which resided on the left side of the shaft, acted as a wheel support bearing for minimal loads. This bearing was selected to provide specific clearances in both axial and radial directions. Therefore, to simulate the worst case scenario for the shaft, it was considered as a cantilever.

### 4.3.3.1 Finite element modelling-shaft

The FE model was developed for a shaft in the Ansys® test bench 13.1, commercial FE software. The local and the global 3D tetrahedral mesh with 4 nodes were developed; as shown in Figure 4.15. Shaft FE model consisted of 72693 elements and 121891 nodes. The fillet/spline areas, wire harness slot and bolt slot hole were modelled with a local tetrahedral finer mesh of 0.5 to 2mm as shown in Figure 4.15. The global tetrahedral mesh of 5mm or more was modelled into the rest of the shaft with a smooth transition ratio of 0.156. The material used for shaft design was EN 26 steel, which added stiffness due to high ultimate tensile strength. The engineering data of EN 26 steel (high tensile steel) material used are i) Young’s modulus \( E \) 210GPa, ii) Poisson’s ratio \( \nu \) 0.26, iii) density \( \rho \) 7.8g/cc and iv) ultimate tensile strength \( \sigma_u \) 1080MPa.
Figure 4.15: Shaft FE Model with a load and boundary conditions

4.3.3.2 Load and boundary conditions-shaft

Figure 4.15 shows the shaft load and boundary conditions. In the FE model the boundary conditions were modelled to simulate the shaft physical conditions. In a car, the shaft is connected to the chassis trailing arm through bolt holes location as shown in Figure 4.15. The fixed support boundary conditions were used at these bolt holes to restrain rotational and translation movements.

The standard kerb weight of the vehicle was considered as 1060kg and an additional 500kg was added to cover passengers and luggage. The gross weight of 1560kg was multiplied by a factor of 1.25, leading to a total load of 4.77kN (rounded to 5kN) on each wheel. This 5kN uniform distributed load was modelled in the FE environment.

4.3.3.3 Finite element results-shaft

The FE results of maximum Von-mises stress and maximum deformation of shaft is shown in Figure 4.16 and Figure 4.17. The maximum Von-mises stress concentration in Figure 4.16 was 326MPa. Neglecting stress concentration areas, which were singularities (e.g., due to sharp corners and edges), 180-200MPa maximum Von-mises stress concentrations were observed in the shaft. The maximum deflections in the shaft (Figure 4.17) were observed as 0.70mm at the overhand area of the shaft in FE model. Similar load case (5kN) and EN 26 material was used to analytically calculate the maximum Von-mises stress and maximum deformation of shaft. It was observed from this that maximum Von-mises stress and deflections were similar to those obtained in FE model (refer Appendix 7 for shaft calculations). The deflections were of the order of 0.7mm in both methods and maximum stress value of 180MPa was generated. Based on
the material yield strength, the shaft was safe for operating within in-wheel SRM without compromising the crucial 1mm gap.

**Figure 4.16:** Shaft results for maximum Von-mises stress concentrations (MPa)

**Figure 4.17:** Shaft results for maximum displacements (mm)

### 4.3.3.4 Finite element modelling-motor

The shaft FE analysis concluded that the deflections were not compromising the crucial 1mm air gap within magnetic path; to further investigate FE study was conducted on the motor. The FE analysis on overall motor was conducted with the objective to examine the overall deflections resulting at rotors and stators inside the motor. The aim was to examine the gaps under 40 degree loading case. The FE model was developed for motor in the Ansys® test bench 13.1, commercial FE software. The local and the global 3D tetrahedral mesh with 4 nodes were developed, as shown in **Figure 4.18**. The FE model consisted of 372693 elements and 621891 nodes. The local mesh was modelled at: i) fillet/spline areas/corners, ii) wire harness slots/holes, and iii) thin sections/contact areas with a tetrahedral finer mesh of 0.5 to 3mm, as shown in **Figure 4.18**. The global tetrahedral mesh of 8mm or more was modelled into the rest of the motor parts with a smooth transition ratio of 0.426. The following material data were used for: i) EN 26
steel (shaft), nuts and bolts, ii) Aluminium 6061-T6 for rim, motor cover, motor bushes, and iii) M15 silicon steel for rotors and stators.

The engineering data of EN 26 steel (BS870 part 1, 826M40) material used are: i) Young’s modulus ($E$) 210GPa, ii) Poisson’s ratio ($\nu$) 0.26, iii) density ($\rho$) 7.8g/cc and iv) ultimate tensile strength ($\sigma_u$) 1080MPa. The engineering material data of Aluminium 6061-T6 material are i) Young’s modulus ($E$) 69GPa, ii) Poisson’s ratio ($\nu$) 0.33, iii) density ($\rho$) 2.7g/cc and iv) ultimate tensile strength ($\sigma_u$) 310MPa when T6 hardened. The engineering material data of M15 silicon steel material used are i) Young’s modulus ($E$) 122GPa, ii) Poisson’s ratio ($\nu$) 0.34, iii) density ($\rho$) 7.65g/cc and iv) ultimate tensile strength ($\sigma_u$) 490MPa.

![Figure 4.18: Half front section of motor FE model](image)

4.3.3.5 Load and boundary conditions-motor

The uniform distributed load of 5kN (calculated in previous section) was modelled over a 40° arc area on the outer rim faces as shown in Figure 4.19. The load was 5kN as the
objective was to examine deflections within the magnetic path. The in-wheel SRM has three rotors and two stators as a magnetic path.

Figure 4.19: Load case (40 degree) for motor assembly (transparent rim end cap for clarity)

The fixed support boundary conditions were used at shaft bolt holes to restrain rotational and translation movements (as shown in Figure 4.18). The no penetration contact patches were modelled between: i) central rotor and motor cover, ii) shaft and aluminium bushes, iii) stator and aluminium bushes, and iv) outer rotors and motor covers.

4.3.3.6 Finite element results-motor

Gaps between magnetic paths were kept at 1 mm, which predetermined the efficiency of the motor. As the gap narrowed, the stator touches the rotor, as the motor was subjected to failure. When the gap increases, it results in decreasing the efficiency of motor. Hence, this analysis of deflections was vital to achieve the required high motor efficiency. Motor assembly FE analysis indicated total maximum deflection of 0.025mm in the rotor-stator gap areas (as shown in Figure 4.20), which indicated that the gap was decreased by 2.5%, affecting the crucial 1 mm gap.
4.3.3.7 Nut selection for motor assembly

The motor coupling or holding fixture defines the primary holder and subjected to: i) self-release due to desired reversing of the motor typical for EVs, and ii) thrust developed within motor during operations. To avoid the self-release for both upthrust and non-reversing during motor operation, non-reverse ratchet and hold down bolts are used. In this design at one end of the motor, the entire assembly was supported by motor thrust bearing (double row tapered ball bearing discussed in previous section). The upthrust and slight radial loads caused bearings to move across axially. This entire system caused upthrust axial loads as a consequence may results in catastrophic failure during long runs. Hence upthrust and low thrust was managed with two bearings (deep groove and tapered roller discussed in previous section). The entire motor assembly, including the shaft, bearings, bushes, stator, rotors and the motor covers was held using nut with holding support (to avoid reversal failure). Hence, it was important to calculate the forces acting on the shaft ends and examine if the devised nut was capable of holding the overall assembly without affecting the crucial 1mm air gap. Calculations were thus performed to determine the shear strength of the nut in the motor assembly. This nut was responsible for holding all the magnetic components, namely rotor and stator subassemblies seated on the shaft in the motor assembly. In this section, a study was conducted to examine the loads acting on this nut without any failure. More
specifically, shear force $F_s$ of the nut was determined using Equation 4.11, as shown below:

$$F_s = \tau A_{th}$$

(4.11)

Where, $\tau$ is the shear strength of the material, which was 600MPa in this case, and $A_{th}$ is thread shear area derived from Equation 4.12, as shown below:

$$A_{th} = 0.5\pi d_p L_e$$

(4.12)

Where $A_{th}$ is the thread shear area, $L_e$ is the thread engagement length, $d_p$ represents pitch circle diameter of the thread, $d$ is the major diameter, and $P=1/ (n)$, where $n$ is the number of threads per inch. Based on those calculations, the nut used in the motor was shown to be capable of withstanding loads up to 10kN. Axial load acting on the nut was calculated as 1.22kN (bearing selection in last section). These calculations indicated that the maximum load the nut is subjected to before it shears off was far greater that the axial load acting on the motor (based on the previous calculations given in the selection on bearings). Factor of safety was more than five as during vehicle rides, the motor was subjected to dynamic loads (bouncing) and other effects (e.g., corrosion even though it is zinc plated). The aforementioned discussion and results indicate that the selected nut was capable of holding the overall motor assembly, ensuring that the crucial gap of 1mm was not compromised. Hence, selection of a nut with higher load bearing value was justified.

**4.3.3.8 Shaft final design**

Following these studies, the shaft design was finalised. The shaft was through hardened to take up the required loads without any wear. Clearance for bearings and other stator areas was dictated by shaft material specifications (EN 26) and stator bushes (Aluminium 6061-T6 alloy), selected for this design. The bearing seating was ground for fine finish and other vehicle mounting holes were created on the shaft. The final design is shown in Figure 4.21, depicting a hollow shaft with splines and ground area for seating taper roller bearings. The following were the main salient points accomplished during the shaft design:

- Design: The shaft was designed hollow, specifically to provide the necessary space for wire harness and cooling pipes. Unique spline design was
implemented to hold the stators in place during the motor operation. On one end shaft was attached to chassis using four bolts from Holden Barina Spark. On the other end nut was holding the stator and rotor subassemblies.

- Efficiency: The shaft was designed with minimal deflections and stresses, taking into consideration the crucial air gap for increased motor efficiency. This was validated using analytical and empirical methods.

![Figure 4.21: Final shaft design](image)

4.4 Summary

In this chapter, the final vehicle motor concept was developed based on the motor specification of the in-wheel SRM. Several concepts were initially considered and the final choice, based on the initial specifications and subsequent comprehensive analysis, was the Holden Barina Spark, to be used as an envelope for the motor. Two types of motor proposals were made and were evaluated based on the set criteria that helped identify the most optimal motor concept for further development. Following are the salient points that summarise the work described in previous sections:

- Hand sketches of horizontal and vertical methods of mounting magnetic path indicated that a vertical concept was more suitable for maximising space utilisation. Further investigation was carried out using VR based schematic sketching methods. Two concepts horizontal and vertical—were evaluated; based on criteria, including power density, optimised magnetic path, ease of assembly, detachment and manufacturability to retain a 1mm crucial air gap. The vertical concept was finally chosen for further refinement, as it was superior to the horizontal one, in achieving the above objectives.
• Magnetic path details such as outside motor diameter, motor length, stators lengths, rotor lengths, rotor back iron details, centre rotor length, number of poles, axial lengths (air gap, wedge, and insulation), insulation thickness, and copper fill factor were determined. Then linear widths of tooth, axial length of rotor/winding, length/area of coil per turn, lamination angle and stator outside/inside diameters were calculated.

• The motor packaging consisted of two motor covers housing two rotors and a central rotor, positioned in the middle. The stators were seated on the shaft and were acting as non-rotating parts within the overall motor assembly. The shaft was hollowed to facilitate fitting of an airflow pipe and wire harness housing inside the motor. The whole assembly was held using a nut on one end and an O-ring was used to restrict water entering into the motor. The rotating and non-rotating parts of motor were separated with two bearings on the either end.

• A SRM was used as an alternative to a PMM, as it offered an efficient alternative drivetrain system. The motor design had a 1mm air gap between the rotor and stator, which ensured high power density of the designed motor. However, adequate stiffness of the motor parts became particularly important, as the aim was to maintain the crucial 1mm air gap. The motor design constituted light shaft, rotor, stators, hub and optimised motor covers. The shaft was stationary, which is not typical for other motors of this type, and it allowed carrying stators, which were supported with bushes. The rotors were attached to the motor cover seated on bearings positioned on both ends and the central rotor was attached to the motor cover in the centre. The crucial 1mm air gap was retained between the stators and the rotors, which ensured efficient magnetic path and increased efficiency. All the related parts were designed and analysed with the aim to stabilise the air gap and prevent any increases or decreases.

• The performance study of the drivetrain hub design was performed using dynamic load bearing analysis to establish selection of bearings. The proposed motor design required two bearings for the hub, due to its rugged nature of operation during typical vehicle rides. One bearing was located in the inner vicinity of the shaft and was designed to withstand high load characteristics. This inner bearing was very rigid and sturdy, with extremely high reliability. The other bearing, located at the outer section of the shaft, was light weight and was dynamically controlled to allow
radial and axial loads. This bearing allowed any deflections in the shafts due to varying loads on the other end of the shaft during the vehicle ride. This was due to the close proximity of the wheel to this outer bearing. Once the assembly was complete, calculations were performed in order to determine the maximum axial and radial loads acting on both bearings and the optimum bearing designs were selected from SKF Bearings.

- The dynamic load ratings of the bearings were also retrieved to further validate the bearing selection. The bearing selection for motor design was crucial; since loads acted on both sides of the motor during typical vehicle rides and maintaining the air gap between the magnetic paths held by these bearings was crucial. To stabilise the 1mm air gap in the motor, which governed power density, bearing load analysis (managing thrust loads) was conducted. A good correlation between two bearings within hub design was established to ensure that bearings would not fail under worst case scenarios that can be encountered during the vehicle ride. Tapered roller paired face to face for inner wheel hub bearing (BTHB329129ABA from SKF) withstood dynamic load rating $C_r$ of up to 105kN, thus the calculated load of 24kN was completely safe. For the recommended outer bearing, the single row deep groove ball bearing (6207 from SKF) withstood dynamic load rating $C_r$ up to 27kN, which made the calculated value of 23.68kN completely safe. Bearing loads were balanced by designing two types on either ends, with a deep groove ball bearing on one end, which permitted transfer of additional radial or axial loads resulting from vehicle rides.

- The designed SRM included lightweight, optimised and robust shaft in the central system. The shaft, a backbone of the motor design, was used to support the stators through the bushes. The entire motor assembly was held on the shaft; hence, deflections and stresses the shaft was subjected to played a vital role in stabilising the crucial 1mm air gap. The deflection and stress analyses were carried out on the shaft and its performance was found to be within the allowable limits. The shaft was further analysed after light weighting using analytical and empirical methods for deflections and stresses to examine effects of deflection on the crucial air gap. The findings indicated that the shaft deflected by a maximum of 0.7mm, which was within the required allowance, without affecting the crucial air gap. Maximum stresses were 180MPa and were within the allowable yield strength for hardened EN
26 material used for manufacturing the shaft. Further study on the overall motor assembly was performed to examine the effects of different load cases, which varied from 0 to 40 degrees during a vehicle ride, on the crucial 1mm air gap. The results indicated that the air gap was affected by a maximum of 2.5%, which was within the required tolerance of the motor design.

- Finally, performance of the nut holding motor assembly at one end of the shaft was examined to determine if the selection affected a crucial 1mm air gap. The holding nut was added to minimise the reversal effect on the motor. As this nut was the only part within the motor assembly that held all the parts, it was essential to ascertain its performance for axial loads. For that reason, the thrust bearing nut was chosen and it was determined that it was capable of withstanding loads of 10kN, which was adequate for the proposed motor design.
Chapter 5

Mechanical optimisation of motor

5.1 Chapter overview

After finalising the design of the motor parts, mechanical optimisation was carried out on the motor parts with the objective of achieving high power density. The mechanical design optimisation aimed at maximising the power density by: i) minimising the motor mass without compromising structural rigidity, ii) maximising the motor space utilisation and refining the assembly structure, iii) minimising the motor temperature by designing an external cooling. The analysis of the key motor parts was aimed to achieve the following objectives of this chapter as shown in Figure 5.1:

- To increase power density minimising the overall weight of the motor is an important part of design. Thus, the aim here is to reduce the weights of individual components using appropriate material and FE analysis (e.g., reduction of motor cover weights without compromising structural rigidity).
- To maximise power density by achieving increased space utilisation, as the motor is housed within the wheel. VR/AR based approach was used to establish, i) space utilisation, and ii) ease of assembly within motor.
- In SRM, stators are expected to reach high temperatures. Thus, thermal stability of all the components is important, as thermal expansion of materials may decrease the air gaps. Evaluating the effects of temperature and designing the cooling arrangements was thus the final aim of this part of the project.
- To establish key motor characteristics for developed SRM. The motor torque, speed and velocity curves are established in the final section of this chapter.

As the motor cover holds the rotors, reduction of its weight without compromising the stiffness was vital. The motor cover stiffness ensured the rotor stiffness inside motor without affecting air gap between the magnetic components. Different materials were studied to finalise and fine-tune the material utilised for manufacturing motor covers. Based on the FE methods, using aluminium motor covers, optimum motor cover thickness was achieved. As previously noted, the motor power density was crucial aspect of motor development, since efficient
assembly sequencing was the key to achieving the optimised motor design. VR and AR techniques were used to optimise the motor assembly sequence.

![Objective tree for motor optimisation](image)

**Figure 5.1: Objective tree for motor optimisation**

As the stators were subjected to high temperatures, substantial cooling mechanism was implemented to maintain temperatures inside the motor within the acceptable range. Moreover, as stators were held with aluminium bushes, any temperature rise caused variations in their dimensions, effecting variation in the air gaps. Thus, the dimensional variations in bushes holding stators were first examined and were found to be within the required allowance. Forced cooling was introduced through the shaft, which was vented from the top of the motor using one-way valve at two locations. Lastly, key characteristics of developed SRM were derived.

### 5.2 Weight reduction of motor covers

#### 5.2.1 Material selection

The weight of the motor is indirectly proportional to its power density, with the motor covers, bushes and shaft, as well as the rotor contributing to the overall weight other than stator. Shaft acts as a backbone, carrying most of the loads; hence, in this design, EN 26 material (high tensile steel) was used for its construction, as it did not compromise the surface area or the mass (discussed in last chapter). The motor covers house three rotors and bearings, and are responsible for stiffness in the motor design. The materials investigated during the design processes of motor covers were:

- Original design with steel
- 6061-T6 aluminium alloy
- Carbon and glass fibre based composites
- Composites for barrel and aluminium material at end plates

The material selection and motor cover optimisation is important in reducing the overall weight of the motor. As shown in Table 5.1, initial studies were conducted, whereby comparative analysis of the materials that could be used for manufacturing motor parts was conducted. Each concept was scored and given poor, fair, good and very good ranging from the worst to the best, in terms of the characteristics, mainly weight, cost, manufacturability/assembly, water proofing, and thermal stability.

**Table 5.1: Competitive analysis of motor cover materials**

<table>
<thead>
<tr>
<th>Objective</th>
<th>Original steel</th>
<th>Aluminium</th>
<th>Composite</th>
<th>Composite barrels/ aluminium end caps</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight</td>
<td>poor</td>
<td>fair</td>
<td>very good</td>
<td>good</td>
</tr>
<tr>
<td>Cost</td>
<td>fair</td>
<td>fair</td>
<td>poor</td>
<td>poor</td>
</tr>
<tr>
<td>Manufacturing/assembly</td>
<td>fair</td>
<td>good</td>
<td>poor</td>
<td>poor</td>
</tr>
<tr>
<td>Ingress protection (IP 56)</td>
<td>fair</td>
<td>good</td>
<td>fair</td>
<td>poor</td>
</tr>
<tr>
<td>Thermal stability (IEC 416)</td>
<td>fair</td>
<td>good</td>
<td>fair</td>
<td>fair</td>
</tr>
</tbody>
</table>

When using steel, associated weights were an issue, as they reduced the motor efficiency. Whilst composites increased the costs of the motor and manufacturing was also a challenge. The composite barrels with aluminium end caps suffered from the restriction with respect to the joining, and were thus less waterproof when compared to other investigated materials. Aluminium designs scored the highest and were thus selected as potential candidates for the motor covers.

In this research, low cost generic materials were used for optimising the motor cover properties without compromising the overall safety. Initial studies of materials were conducted on series of aluminium alloys to evaluate the most suitable material for the motor cover design. The investigated Aluminium 6000 series consisted of Aluminium, Silicon and Magnesium composition. Among the commonly used materials in the 6000 series were 6060, 6061, 6063, 6005A, and 6082. The 6061 offers smaller composition of Silicon and Magnesium (0.4-0.6%) in its class relative to 6082 (0.7-1.2%). The 6061 series had good mechanical properties and was subjected to T6 heat treatment. Furthermore, these alloys had medium to high strength, were easy to weld and offered good resistance to corrosion, even when exposed to water. In this motor cover, one side of the motor cover was exposed to the environment, with water splashing on the
components during the vehicle ride. *Table 5.2* below shows the properties of 6061-T6 class in the aluminium material 6000 series.

*Table 5.2: Aluminium alloy properties of 6061-T6*

<table>
<thead>
<tr>
<th>Material properties</th>
<th>6061-T6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield Strength, Mpa</td>
<td>290</td>
</tr>
<tr>
<td>Tensile Strength, Mpa</td>
<td>310</td>
</tr>
<tr>
<td>Elongation, A5%</td>
<td>10</td>
</tr>
<tr>
<td>Brinell Hardness, HB</td>
<td>67</td>
</tr>
<tr>
<td>Density (kg/m$^3$)</td>
<td>2700</td>
</tr>
<tr>
<td>Young’s modulus, p a</td>
<td>69000</td>
</tr>
<tr>
<td>Coefficient of expansion 20-100 C ( C-1)</td>
<td>23E-6</td>
</tr>
<tr>
<td>Thermal conductivity 10 C (W/mK)</td>
<td>200</td>
</tr>
<tr>
<td>Electrical Conductivity (%IACS)</td>
<td>52</td>
</tr>
<tr>
<td>Melting point ( C)</td>
<td>600-655</td>
</tr>
</tbody>
</table>

After comparisons of the key material properties, 6061 with T6 treatment was used for manufacturing the bushes and motor covers. It was the most optimal choice for this design because it offered specific advantages: i) cost effective and ii) readily available, unlike other materials, and iii) good mechanical properties presented in *Table 5.2*.

### 5.2.2 Mass optimisation

Following the material selection, further FE studies were conducted to determine the required motor cover thickness.

#### 5.2.2.1 Finite element modelling

Initially, motor cover was modelled using the local and the global 3D tetrahedral mesh with 4 nodes as shown in *Figure 5.2*. FE model consisted of 81513 elements and 159965 nodes. The fillets and bolt holes were modelled with a local tetrahedral finer mesh of 2 to 5mm. The global tetrahedral mesh of 8mm or more was modelled into the rest of the motor cover with a smooth transition ratio of 0.4. Transition ratio allowed a smooth flow from the local to the global mesh model within the motor cover.

The aluminium 6061-T6 engineering material was in the FE model, supplier data was used (ASTM International 2011). The engineering material data of Aluminium 6061-T6 material are 
- i) Young’s modulus ($E$) 69GPa, 
- ii) Poisson’s ratio ($\nu$) 0.33, 
- iii) coefficient of thermal expansion ($\alpha$) $2.6 \times 10^{-6}/°C$, 
- iv) density ($\rho$) 2.7g/cc and 
- v) ultimate tensile strength ($\sigma_u$) 310MPa when T6 hardened.
5.2.2.2 Load and boundary conditions

*Figure 5.2* summarises loads and boundary conditions modelled within the FE environment. Two types of loads were modelled: i) the thermal load, and ii) the static load.

Based on IEEE 11-2000 standard, titled “IEEE Standards for Rotating Electrical Machinery for Rail and Road Vehicles-2000”, a maximum surface temperature of enclosed ventilated motor is 70°C (IEEE 2000). The ambient temperature of 22°C and the relative time dependent temperature rise of 70°C were modelled on the motor cover internal faces as shown in *Figure 5.2*. The convection coefficient of 50W/m²°C was modelled to the motor cover external faces as shown in *Figure 5.2*.

The standard kerb weight of the vehicle was considered as 1060kg and additional 500kg was added to cover passenger and luggage. The gross weight then 1560kg was multiplied with 1.25 safety factor, leading to a total load of 4.77kN (rounded to 5kN) on each wheel. This 5kN uniform distributed load was modelled in FE environment as shown in *Figure 5.2*. **Figure 5.2: Motor cover FE model**
5.2.2.3 Finite element results

Initial FE study was conducted using stress as a sensor, to evaluate rigidity with the aim of optimising the thickness of motor cover (mass). The thickness of 1.5 to 3.5mm (with 0.5 mm increment) was considered as a parameter, with an objective to keep the mass below 3kg and retaining optimum stress i.e., within permitted factor of safety (i.e., FOS>1.5). Five sets were trailed with goal of optimisation, with incremental wall thickness variation of 0.5mm, from 1.5mm to 3.5mm, as shown in Table 5.3.

Table 5.3: Design scenario for motor cover optimisation

<table>
<thead>
<tr>
<th>Design variables</th>
<th>Optimal (4)</th>
<th>Scenario 1</th>
<th>Scenario 2</th>
<th>Scenario 3</th>
<th>Scenario 4</th>
<th>Scenario 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter (mm)</td>
<td>3.00</td>
<td>1.50</td>
<td>2.00</td>
<td>2.50</td>
<td>3.00</td>
<td>3.5</td>
</tr>
<tr>
<td>Mass (gms)</td>
<td>&lt;3000</td>
<td>2923.64</td>
<td>2923.64</td>
<td>3310.96</td>
<td>2923.64</td>
<td>3310.96</td>
</tr>
<tr>
<td>Max. Von-mises stress (Pascals)</td>
<td>Monitor</td>
<td>30299.37</td>
<td>failed</td>
<td>failed</td>
<td>30299.37</td>
<td>14259.58</td>
</tr>
</tbody>
</table>

The test sets 1, 2 and 3 caused failure as the goal driven design scenario optimisation was implemented as an iterative process, whereby the final solution is found once the best fit to the set objectives is achieved. The test scenarios indicated that set 4 and 5 were more appropriate than other sets as shown in Table 5.3. Moreover, set 4 (with thickness of 2.9kg) was shown to be optimal, since the objective was to keep the weight below the 3kg target. As discussed in chapter 2, ribs serve two important aspects of motor cover: i) heat dissipation and ii) structural rigidity. Hence, the motor of 3mm thickness was designed with 3mm ribs positioned diametrically to increase stiffness across the cross section and improve heat dissipation.

5.3 Space utilisation and assembly sequencing

In this section goal of maximising power density was achieved by the following key accomplishments:

- Space utilisation: One of the key factors for motor optimisation was to increase its power density by utilising the space available inside the wheel, where the motor would be housed. Visualisation technique using virtual reality (VR) was employed to facilitate motor design, in which series of images were utilised to represent and analyse the motor design. The chosen technique used images and other animations, and was shown to be an effective way to convey both concrete and abstract ideas, allowing the assessment of the space utilised. This methods
helped in improving the subassemblies (e.g., rotor and stator packs) and over all space utilisation for magnetic path within the wheel maximised the power density.

- Easy assembly and fitment: The assembly sequence was optimised for packaging the motor components. The assembly and fitment involved using AR by effectively incorporating Cyber glove II® for interacting with VR models in real time to improve the assembly sequence (e.g., tolerance stacking rotor and stator packs) and improve the fitments (e.g., IP 56 tolerances between motor covers).

5.3.1 Virtual reality based space utilisation

The visualisation technique has found many applications in the engineering world, due to the evolution of computer graphics and animation techniques. As a result, different terms referring to visualisation have emerged in many fields of study, e.g., computer graphics, animation, VR and simulation. In this section VR based simulation for maximising space utilisation is explained.

5.3.1.1 Virtual reality modelling

The visualisation performed in this study consisted of using 3D models created for designs, whereby the correct placement of camera and lights were needed to capture the real time assembly space within EV context. Next, background and scenes (e.g., the vehicle digitised model/the rim discussed in chapter 3) were added to mimic the real life environment. The cameras were based on degrees of freedom and metaphors were used to control these. These 3D models were then attached with a specific curve, which predefined smooth motion of the assembly and interpolated between the parts. The placement of camera was dependent on the output required for accurate virtual design representation. Metaphors were predefined and included aspects, mainly, fly, orbiting, exocentric views, and scenes depending, on the specific animation requirements. Other control methods, including external widgets, e.g., style or hoover cams, were used to control the camera motions, depending on the scene required. Once camera trajectories were defined, lighting was added to the scene, based on the scene characteristics and quality of the animation required. In this VR study, the sequencing of different lights were used for the validation of the motor design. Target lights, sky light, free light and
omni light were some of the different options used in the validation of motor design in this study. Eight cameras were used for auto stereoscopic displays, where three dimensional effects were immediate and do not require any special eye ware. Once the camera was positioned, the timeline was generated for the required motion of the motor. Each motion was captured using keys, which were changed in order to vary the time line and control the motion time line. The keys were displayed at each start and end points at time intervals and were moved to control the motions. Finally, the motion was smoothened using spatial positions and timeline of each component was generated, as shown in Figure 5.3.

The depth variations and the use of two backgrounds enhanced the appearance of the objects and made them more realistic. The front image was rendered with eight cameras, which generated 3D effect on the rendered images. The scene was captured by playing the motion and the eight cameras used captured the overall motion analysis, as shown in Figure 5.4. The final generated animation consisted of rendered images, generally providing eight views of a particular object of the motor in greater depth at each particular time interval. These generated images were mixed using “mi and play assistant”, after which audio was added to improve the effects of the scenario. Since length of the complete animation exceeded five minutes, these animations were created in sequence of small subsections and the script file was finally mixed (refer to Appendix 8 for more detailed description).
Figure 5.4: VR used for auto stereoscopic optimisation using 8 cameras and lights (Omni and spot lights applied) for space utilisation studies

The auto stereoscopic used these algorithms to multiplex into one single image from the eight generated images. Generally, seven stereo pairs were used for solving these algorithms of motor assembly and disassembly, resulting in effective optimised packaging. Alioscopic display was created, as shown in Figure 5.5 representing lightweight shaft, as well as modular rotor and stator packs.

Figure 5.5: VR used for auto stereoscopic optimisation for space utilisation studies
5.3.1.2 Virtual reality results

As shown in Figure 5.5, the following were examples for improvements during design process with use of VR tools:

- The motor cover shape (outer face) was optimised with a minimal clearance of 2mm from rim inside faces. This also included retracing the end cap internal faces to match the motor outer faces.
- Motor cover was slotted to incorporate part of rotor serving as support as well maximising space for magnetic path.
- Stators were packed as subassemblies making them compact and modular for assembly process.
- Central rotor was introduced to maximise the power density (e.g., back to back motor). This resulted in two stators and three rotors within the assembly. This also facilitated split line in the motor covers for easy manufacturability.
- Wire harness and cooling passage was developed resulting in drilling appropriate cooling holes within subassemblies.
- Finally, sensor assembly was oriented within the space between the motor rotor and shaft to utilise available space.

These VR visualisation techniques had the following advantages for motor space maximisation:

- The array of images for display replaced the use of eyewear, generally required to visualise 3D. Since the motor has both electrical and mechanical parts, the 3D visualisation experience and the real time modification of the motor were easy to analyse. This allowed for both demonstration and fine-tuning of the motor in which a large group of people participated (e.g., AutoCRC/CSIRO consortium members) during presentations and resulted in modifying the design for maximising space utilisation.
- Material, textures, scenes and background mimicked real world experience, allowing easy understanding the space required for motor design within rim envelope. In particular, aesthetics and the design context with assembly sequencing were fine-tuned during this motor design stage. The content added clarity without distorting the images, thus even tiny objects, e.g., small pins and bolts were analysed, improving the overall motor design space. Creation of high
quality presentations for displays with audio aids increased the effectiveness and clarity of motor design sequencing.

- The shaft, stator, rotor packs and motor cover were examined for space in an optimised sequence. This enabled resolving clearance issues, manufacturability and the improvisation to overall motor design. Thus, the available within wheel space was utilised for maximum magnetic path, improving the motor power density.

5.3.2 Augmented reality based assembly sequencing

5.3.2.1 Virtual reality modelling

An immersive system was used for further improvisation by stereoscopic visualisations, using projectors with EON® software. This method allowed design interactions in real time by combining multiple sets of views with the applied background and scenes. Use of head-mounted glasses allowed visualisation of two views as one simplex 3D image. This classical method of visualisation is based on two stereo pairs. Special eyewear was used to differentiate left and right images created by these stereo pairs in the visualisation process. This software also enabled switching of left and right images for clarity of animation.

Figure 5.6: Virtual and augmented methodology used in this research
In this research use of EON® software was critical for space optimisation in the proposed motor design. The overall methodology used in this research for motor design optimisation is depicted in Figure 5.6. This typically consisted of importing 3D models created in other CAD software, after which interactivity, prototypes and scripts were used to build and convert them into studio future. To simulate physical scenarios, computerised 3D model that symbolised the real object created during the design stage was imported. Occasionally, 3D models have been referred to as mathematical models, since made up of a set of XYZ coordinate values that represent the surface geometry of the physical objects being depicted. After such a model was imported, the next task was to animate the model using 3D Max studio software, in which computer-based tools were specifically defined to finalise the rendered animations. This application stepped through time, frame by frame, whilst constantly providing updates of the state, event and XYZ coordinate values for each 3D model in the given simulation. Variation of degrees of accuracy was achieved through the simulation of motor objects and scenes. Motor models that were designed with a fluctuation of detail resulted in the output of the concurrent simulation. Extremely fine detail of motor models was achieved and consisted of hierarchy of subcomponents, each of which was a complex model in itself. EON® Professional was used for adding visual effects, humans, and backgrounds (e.g., car, garage etc.) to the created physical representation of the motor.

5.3.2.2 Augmented reality modelling
Cyber glove II® was used as virtual hand for interaction with motor design, thereby improvising the motor design. The Cyber glove II® used in this design stage was constructed with stretch fabric for comfort and mesh was palmed for ventilation. The 18-sensor cyber glove II® system included open fingertips, which allowed easy grasping of objects and improving motor design critical issues. It used batteries and was connected to the overall assembly. Initially, the position and orientation of the forearm was defined as an object identified in EON® software. Based on the mountings, e.g., InterSense, Polhemus, and Ascension, six degrees of freedom (DOF), motion tracking sensors detected the motion and interfaced with the motor design.
Figure 5.7: An example of using cyber glove II® and motion capture with EON® reality

Motion capture consisted of using 20-camera system (infrared) using Arena® software. The motion capture process used camera calibration, whereby easy skeleton creation based on physique of personnel allowed recording of multiple actors and real time motion capture. Generally, reflective markers placed on actors and props were used, as this created a cloud of 3D points. Experimentation for motion capture is illustrated in Figure 5.7, where a reflective clothed human is performing motor assembly sequencing. Motion capture clothing had reflective markers, allowing the camera to capture images that represent real time human motion. These 3D points were subsequently labelled and mapped to a skeleton solver for tracking full body motion. Captured actor and prop data was exported to 3D max studio using standard 6 DOF digital file format for further 3D animation and analysis.

The designed motion data were integrated into 3D animation using 3D max studio. The motion capture data were exported as bio vision hierarchy (BVH) file from Arena® software where the motions were edited in order to eliminate noise from the captured data, mostly due to inconsistencies in the flow in the motion path. Generally, the affected data sets were broken into small segments in the edit section to improve the flow of motion. These discrete areas were identified and processed to remove unwanted noise. The data purging process consisted of highlighting the area of motion and editing the curvature in order to produce smooth curves that represent fluid motion. The BVH files were subsequently assigned as skeletons within EON® prototype files. This allowed real time simulation of an Avtar through the skeleton linked to the motion data.
of the actors. The captured data was integrated into the design for assembly feasibility studies.

5.3.2.3 Augmented reality results

Next, the simulation was created, as the engaged motor design required compelling and interactive experience, which enabled virtual validation of the motor assembly. It also allowed for the demonstration and exploration of the conceptualisation of different motor subassemblies. These designs were routed through a series of scripts for required simulations. Simulation tree and property managers were used to edit nodes of individual objects to connect and route them appropriately for required simulations. These EON® simulations were used for interactive motor design walk-through using the latcher system for display and a tracker system, i.e. Cyber glove II® and motion capture, described above, were used for interactions. An example of wheel simulation file with prototype tree is shown in Figure 5.8.

![Figure 5.8: EON® Prototype file on left and demonstration on right](image)

EON® software used the concept of prototypes, which were simply reusable assets or objects. Prototypes were akin to codes that were packaged into components and held in a software catalogue, so that they could be referenced and utilised by other applications, if required. The next step was to add grips with the notions with aim of 3D sound for a
VR-based simulation. The volume in EON® Reality was a combined effect of the following three volume values:

- Volume field (from the direct sound property node)
- Attenuation due to distance
- Attenuation due to sound cones

The three values given above were multiplied in order to determine the final volume level that was applied at any instance within the simulation demonstration of the motor. The volume field was used during demonstrations of the motor simulation, whereby the motor design walkthrough and interactions was performed in real time. This in particular was to focus on individual motor parts, by assigning different volume tone to mimic the live environment. Similar studies were conducted for arrangements and interactions of the motor, whereby the digitised vehicle model was used for the improvisation of assembly techniques. Electric wheel was mounted on a digitised vehicle and the Cyber glove II® was used for interactive demonstration, as illustrated in Figure 5.8. Fine-tuned motor assembly is shown in Figure 5.9.

![Image: Motor assembly sequence using AR tools]

**Figure 5.9: Motor assembly sequence using AR tools**

Motor design was fine-tuned and the following assembly sequence was established using AR tools (as per Figure 5.9):

- Step 1: Lightweight shaft
- Step 2: Outer subassembly, consisting of the following:
  - Motor outer cover
  - Rotor attached using bolts
Stator packs (stators, support palates, bushes, bolts)
- Outer bush
- Sensor sub assembly
- O-ring

- **Step 3**: Centre rotor subassembly, consisting of the following:
  - Centre rotor attached using bolts
  - Support palates, bushes, bolts

- **Step 4**: Inner subassembly, consisting of the following:
  - Motor inner cover
  - Rotor attached using bolts
  - Stator packs (stators, support palates, bushes, outer bush, bolts)
  - Inner bush

- **Step 5**: Brake subassembly, consisting of the following:
  - Brake caliper
  - Disc brake
  - Hydraulic circuit
  - Attaching bolts

The following were salient points achieved using AR tools (EON® software and the Cyber glove II® system):

- High quality stereoscopic 3D with superior image quality provided clear presentation of the proposed motor design. The Cyber glove II® provided an opportunity to interact with motor designs and parts in real time. This arrangement also allowed visualisation of the motor and the digitised vehicle, and enabled fine-tuning of the overall assembly.

- The motor designs using these systems proved to be reliable for effective simulations. Multiple people shared and interactive 3D stereo display which effectively demonstrated complex motor designs. Interaction based on interactive devices (e.g., Cyber glove II®), that enabled a walkthrough of the designs, helped create an easy dismantling protocol for the motor, as this was practiced and fine-tuned within a digitised vehicle in real time. This approach verified the effectiveness of the motor assembly function.
• Immersive viewing through passive or active stereoscopic techniques enabled visualisation of environments that were difficult to visit in ‘real life’, e.g., design subassemblies within the motor assembly. Of particular interest for this design were stator-rotor subassemblies, improving assembly sequencing, clearances and dimensional fitments of the within-wheel envelope. These tools also optimised motor assembly and within-wheel space utilisation for motor power density improvement.

5.4 Thermal stability analysis and cooling design

One of the characteristics of the SRM is a significant temperature increase inside the stators during operation. The temperature rise has three main implications for the motor: i) it reduces efficiency, ii) it reduces the torque, and iii) it affects air gaps, which can substantiate motor cease or failure. The temperature rise in motor affect the efficiency of the motor, hence it becomes vital to manage the temperature rise. The electromagnetic performance of the motor is directly related to thermal stability of the motor. The magnetic path is affected if temperature rise is not ventilated and results in resistance increase, leading to copper losses causing a variation in torques. Clearances and air gaps between the stator and rotors were crucial in this design; as any thermal expansion affects the crucial 1mm air gap. The H class insulation was used for key components of the SRM, which is suitable for very heavy performance at high ambient temperature at a higher speed. These key insulation was used on: i) copper wire insulation, ii) polyester sheets (low thermal conductivity) to insulate stator slots, and iii) insulation paint (varnish) on wound coils. To manage the motor thermally, the following were examined and designed:

• Based on the IEEE 11-2000 standard discussed in chapter 2, the H class insulator was used capable of handling up to 180°C. H class insulation was used between the stator and aluminium bushes to minimise heat transfer from stator to the aluminium bushes and other motor parts. However when overloaded the heat through convection and radiation causes some effects on the aluminium bushes resulting in deflections in crucial 1mm air gap area. The worst case scenario was examined for effects of temperature rise on aluminium bushes using analytical calculations.

• The motor was designed with a surface temperature of 70°C and the stator was 120°C (class B motor, the IEEE 11-2000 standard). The forced cooling was
supplemented in compliance with IEC416, by selecting an appropriate fan to minimise the stator temperature inside the SRM.

5.4.1 Thermal stability analysis

The motor was designed to IEC416 standards, which meant that it was a totally enclosed motor with forced fan cooling to ventilate the heat. Hence thermal insulation material (with a thermal conductivity of 0.02-0.8W/m·°C) was used within the stator and bushes to avoid conduction. The thermal conductivity $T_{\text{cond}}$ is calculated using Equation 5.1:

$$T_{\text{cond}} = \frac{L}{K \times A} \ [\degree \text{C}/\text{W}]$$  \hspace{1cm} (5.1)

Where $L$ is the length, $A$ is the area and $K$ is the thermal resistance coefficient of the material. Fibre glass material (0.0 m·°C low thermal conductivity) was used to separate the stators and bushes within the motor to reduce conduction within motor parts. Aluminium bushes were attached to the stator and acted as supports, mounting the stator to the shaft. Although an insulation shield was used between bushes and stators the thermal expansion within bushes due to convection and radiation affected the crucial 1mm air gap. As a result, any deflection in bushes causes deflections in the stator, thus affecting the crucial 1mm air gap and affecting the motor power density and motor’s performance.

Simple analytical calculation was conducted to evaluate the effect of thermal expansion when the motor was operational and overloaded. Each electric motor contained two bushes: i) an outer bush (Bush 1) and ii) an inner bush (Bush 2). The flux produced during the rotation emitted an excessive amount of heat, which was calculated as 250°C (1.4 factor of safety based on 180°C insulation). The heat dissipated within the motor was substantial and the analysis on the thermal expansion bushes implicated motor performance, as aluminium material was used in their manufacture. Aluminium is characterised by a high thermal expansion coefficient. Hence, calculations were performed to determine the maximum expansion of bushes in the worst case scenario. Thermal expansion calculation is given by Equation 5.2:

$$L = a \times L_1 (T_2 - T_1)$$  \hspace{1cm} (5.2)
Where $L$ is the linear expansion, $L_1$ is the initial length, $\alpha$ is the coefficient of linear expansion of the material, $T_1$ represents the initial temperature, and $T_2$ is the maximum temperature. The initial length $L_1$ varied according to the location of the stator holder, due to its complex geometry. The coefficient of linear expansion of aluminium was taken as $23 \times 10^{-6}/^\circ C$. The initial ambient temperature for this case was taken as an average room temperature $22^\circ C$, while the maximum temperature to which the stator holder was exposed was calculated to be $250^\circ C$, as a worst case scenario. Therefore, using Equation 5.2, the linear expansion can be calculated and for safety reasons, it was assumed that the entire body of the holder was expanding. The results obtained indicate that the greatest expansion yielded an addition of 0.53% to the outermost diameter of the bush, which was 0.93mm (refer to Appendix 9 for the calculated dimensions of the aluminium bushes). These evaluations determined that the bush design and the proposed material were safe with respect to the temperature conditions.

### 5.4.2 Forced cooling design

Forced cooling from a fan to the stator was introduced to reduce temperatures of the stators and associated components. Convection is the heat transfer mode between a surface and a fluid, the two ways possible are: i) natural or ii) forced. The convection results in laminar flow at lower velocities followed by turbulent flow. A Reynolds number is used to define the relationship between laminar and turbulent flow. The thermal convection $T_{conv}$ is calculated using Equation 5.3:

$$T_{conv} = \frac{1}{Ah_c} [^\circ C/W]$$

(5.3)

Where $A$ is the area and $h_c$ is the heat transfer coefficient (for forced air cooling it is 50-350W/m-$^\circ$C). Radiation accounts for heat transfer by electromagnetic waves inside the motor. For radiation effects, the Stefan-Boltzmann equation is used to calculate radiation exchange between the surfaces. Inside the electric motor, thermal convections resulting from radiations were: i) 8.5W/m-$^\circ$C between copper-stator, ii) 6.5W/m-$^\circ$C between end winding and the ring, iii) 5.5W/m-$^\circ$C between ring and ambient.

Determining the most suitable cooling fan depends on: i) type of fan ii) air flow rate, iii) air pressure, and iv) noise level (IEEE 11-2000 standard). A small fan was employed to force cool the stators inside the SRM. The first three types were investigated amongst four available fans: i) axial propelled fan, ii) centrifugal fan, iii) cross flow fan and iv)
tower fan. Axial propelled and centrifugal fans were the most suitable as they were suited to the space availability and cooling requirements. The cross flow and tower fans were heavy and air flow circulation with ducting pipe was also an issue. An axial fan is typically used next to the motor and hence was not suitable due to space limitations inside the wheel. Also the motor was completely enclosed in compliance with IP 56 for avoiding any water or dust inside. Generally the rpm of small fans is 3500 and axial fans can produce smaller output with higher air flow. On the other hand the centrifugal fan produces smaller output at lower air flow resulting in higher air pressure. In this case ducting pipe was used to ventilate the motor, since the high air pressure was required to achieve the required air flow to the stator. Hence the centrifugal fan was selected, which was mounted beneath the car (near fuel tank, as this area becomes redundant in an EV). The stators were cooled by a ducting pipe ventilated through a hollow shaft. The selected centrifugal fan constitutes the following other accessories: i) mesh/felt filter, ii) finger guard, iii) mounting bracket and iv) duct joint and v) duct pipe.

Based on the centrifugal fan the airflow rate was determined to select an appropriate size. The air flow rate $A_{CFM}$ is usually measured in Cubic Feet per Minute (CFM). In order to calculate the $A_{CFM}$ required for this SRM, Equation 5.4 was used, as shown below:

$$A_{CFM} = \frac{W \times K}{(T_1 - T_2)}$$

(5.4)

Where $W$ is watts dissipated, $K$ is constant, $T_1$ ($^\circ$F) is allowable temperature and $T_2$ is inlet temperature. In this case, for standard air conditions, the constant $K$ was taken as 3.16, which was the specific heat of air at standard conditions. The input power for each motor was 15kW (discussed in the next section) and an assumption for power losses were: i) stator core losses, ii) stator coil losses, iii) rotor losses iv) mechanical losses, and v) stray load losses. The first three losses resulting from magnetics accounted for 75-80% of total losses. Mechanical losses (due to bearing friction, though not substantial, has some implications during high speeds) and stray losses (pulsation due to change of reluctance of the magnetic path of teeth during the movement of rotor teeth with respect to stator teeth) account for the rest of losses. A total loss of 10% was calculated and hence dissipated heat accounted to 1500W (based on 15kW power output motor). Next, the allowable operating temperature was set to ($T_1$) 50$^\circ$C, while the inlet
air was taken to be at the ambient temperature of \( T_2 \) 20°C. Therefore, the temperature increase calculated based on these values was 30°C (102°F). Using Equation 5.4, a minimum of ~46CFM fan was required to dissipate the required heat inside the stator.

Air pressure losses arise from: i) flow path (duct pipe), ii) internal components of motor and iii) outlet shape (valve on the motor). It accounts for square of air flow and represented by quadratic curve called resistance curve. In this case the air flow of 2.0\( \text{m}^3/\text{min} \) was required and a small duct pipe of 16mm was used inside the hollow shaft to ventilate the stators. The expected air pressure to achieve the air flow inside the motor was 200pa (based on supplier resistance curves). The noise level within or any component connected to motor was required to be less than 105dBA at 4.5metres operating at rated speed (the IEEE 11-2000 standard discussed in chapter 2).

Therefore, the centrifugal fan selected for this motor was required to produce: i) air flow rate of 48CFM, ii) air pressure of 200pa, iii) a noise level lower than 105dBA. However, the final selection was based on compliance or a higher value fan. The most common cooling fan that fitted this criterion was selected, i.e. one that had specified diameter of 150mm with 134mm width (based on space available) and capable of 81.2CFM air flow rate. With the fan an appropriate mesh filter, felt filter and finger guard were also chosen. The following were the details of the chosen fan (Oriental MB1040-B): i) air pressure 81.2CFM, ii) maximum static pressure 206pa, iii) airflow 2.3\( \text{m}^3/\text{min} \), iv) noise level 60dBA, and v) RPM 2750.

5.5 Motor key characteristics

The motor key characteristics were developed based on the selected vehicle (including wheel discussed in chapter 3) and motor designed (discussed in chapter 4). The selected vehicle and wheel specifications were used to determine motor torque and power with respect to the vehicle speed. This comprehensive analysis allowed an overall optimum solution to be found for the complete drivetrain for the selected Holden Barina Spark.

The starting point to define motor characteristics was the specification that defined limits and weights for design parameters and/or derived quantities. Given the above, the specifications considered based on the Holden Barina Spark and designed motor are defined in Table 5.4. It should be noted, however, that the tabulated values were taken as conservative values of the Holden Barina Spark and 205/50/R17 rolling resistance.
tyre (e.g., the vehicle mass of 1200kg instead of 1060kg, motor power factor of 0.75 instead of 0.9).

**Table 5.4: Specification for the in-wheel SRM**

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification considered</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass ((m))</td>
<td>1200kg (total mass including extra weight of batteries and other ancillaries explained in chapter 8)</td>
</tr>
<tr>
<td>Acceleration due to gravity ((g))</td>
<td>9.81</td>
</tr>
<tr>
<td>Density of air ((\rho))</td>
<td>1.225kg/m³</td>
</tr>
<tr>
<td>Coefficient of rolling resistance ((C_r))</td>
<td>0.012 (maximum LRR coefficient taken from chapter 3)</td>
</tr>
<tr>
<td>Vehicle maximum speed ((v_{max}))</td>
<td>38.8m/s (140km/h)</td>
</tr>
<tr>
<td>Vehicle acceleration time ((t_a))</td>
<td>12s (0 to 22.22m/s)</td>
</tr>
<tr>
<td>Vehicle average acceleration and deceleration (under regeneration) ((a_{avg}))</td>
<td>1.82m/s² (0 to 22.22m/s in 12 seconds)</td>
</tr>
<tr>
<td>Vehicle frontal area ((A))</td>
<td>2.53m² (Holden Barina Spark 1695mm ×1490mm)</td>
</tr>
<tr>
<td>Coefficient of drag resistance ((C_d))</td>
<td>0.33</td>
</tr>
<tr>
<td>Gradient angle 2% ((\alpha))</td>
<td>1.0 degree (Tan (\alpha = 0.2))</td>
</tr>
<tr>
<td>(\cos \alpha)</td>
<td>0.97</td>
</tr>
<tr>
<td>(\sin \alpha)</td>
<td>0.24</td>
</tr>
<tr>
<td>Wheel rolling radius ((r))</td>
<td>0.3184m (205/50/R17)</td>
</tr>
<tr>
<td>Number of motors ((n))</td>
<td>2 (rear wheel drive)</td>
</tr>
<tr>
<td>Motor efficiency at continuous rating ((\eta_m))</td>
<td>0.9</td>
</tr>
<tr>
<td>Maximum motor mass ((m_m))</td>
<td>40kg</td>
</tr>
<tr>
<td>Maximum wheel braking torque under a fault ((\tau_m))</td>
<td>-78Nm (1/3 of the wheel locking torque in the wet)</td>
</tr>
<tr>
<td>Minimum voltage ((V_{m}))</td>
<td>234V</td>
</tr>
<tr>
<td>Motor power factor at continuous rating ((K_{pf}))</td>
<td>0.75</td>
</tr>
</tbody>
</table>

When the vehicle is in motion, the longitudinal force required was given by *Equation 5.5*:

\[
F_X = F_R + F_D + F_G
\]  

(5.5)

Where \(F_R\) was the rolling resistance force required of the vehicle mass \(m\) with a rolling resistance coefficient \(C_r\) at an acceleration due to gravity \(g\) on an incline of \(\alpha\), given by *Equation 5.6*:

\[
F_R = C_r \cdot m \cdot g \cdot \cos \alpha
\]  

(5.6)
$F_D$ was the drag force to overcome aerodynamic resistance offered by a vehicle at speed $v_{max}$ through air of density $\rho$ with Coefficient of drag resistance $C_d$ offered by the vehicle frontal area $A$, given by Equation 5.7:

$$F_D = 0.5. \rho. C_d. A. v_{max}^2 \quad (5.7)$$

$F_G$ was the gradient resistance force of the vehicle mass $m$, and an acceleration due to gravity $g$ on an incline of $\alpha$, given by Equation 5.8:

$$F_G = m. g. \sin \alpha \quad (5.8)$$

The vehicle motion can be translated into a single motor’s load point at straight motion given that $\alpha = 0$. In vehicle straight line motion, there is no gradient force as the value of $\sin \alpha$ tends to 0 and rolling resistance force is $C_r m g$ as the value of $\cos \alpha$ tends to 1. Then motor speed $\omega$ was calculated using the following Equation 5.9:

$$\omega = \frac{v_{max}}{r} \quad (5.9)$$

Similarly in the motor torque $\tau$ was calculated using power factor at continuous rating with the following Equation 5.10:

$$\tau = \frac{F_x r}{n} \quad (5.10)$$

From Equation 5.11, the motor power $P$ per motor was calculated as:

$$P = \frac{F_x v}{n} \quad (5.11)$$

Using equation at no acceleration when the vehicle is at maximum speed in a straight line, the continuous ratings from Equations 5.5 to 5.11 at $\alpha = 0$ and $v = V_{max}$ for the motor were calculated as 122rad/s vehicle speed, 145Nm maximum torque, and 17.5kW maximum power. The motor torque at gradient ($\tau'$) has an upper limit imposed by the available mass and a lower limit given by the maximum gradient ($\sin \alpha = 0.24$ and $\cos \alpha = 0.97$). The maximum torque using gradient was calculated using Equations 5.5 to 5.11, as 593Nm. Starting at $t = 0, v = 0$, and $\alpha = 0$ up to $t = t_\alpha$ the values were generated for maximum speed $v_{max}$. The vehicle power and torque were tabulated (calculations table in Appendix 10) with vehicle speed to obtain the motor characteristic curves as shown in Figure 5.10.
Figure 5.10: Motor curves for power & torque versus speed

As per Figure 5.10, the total motor power at continuous rating is 24kW (both motors) at 60% of the rated speed (2100 rpm) with a peak torque of 96Nm. For vehicle motion, a motor's peak and continuous requirements are described by its maximum torque, which it holds at a base speed, after which the torque declines following a constant power characteristic, until maximum speed is reached, as shown in Figure 5.10. Similarly based on the constant and peak value of speed and torque, motor acceleration and velocity were plotted against acceleration time. Figure 5.11 shows an average acceleration value of 1.8m/s² and maximum acceleration value of 2.31m/s². Also acceleration is constant after the vehicle reaches maximum speed, followed by drop in the average acceleration as shown in Figure 5.11. Based on these motor power characteristic studies, two motors of 15kW at rear wheel were determined for the selected Holden Barina Spark.

Figure 5.11: Vehicle acceleration and velocity at time (up to tₘ)
5.6 Summary

The motor design and relevant parts were finalised, as described in this chapter. It consisted of three rotors, two stators and a central shaft supported with unique hub design. The motor was optimised for reduced weight, space, assembly, and specification sheet was accomplished. The salient points addressed in this chapter are:

- The motor weight was reduced by designing motor covers with optimal thickness without compromising their stiffness. Since the motor cover housed the rotors on either ends, any deflections due to load would cause issues within the overall motor design. Initially motor cover material of Aluminium 6061-T6 was finalised by comparing different materials. Then the motor cover was optimised using FE methods by optimising the mass. Five sets were tested, with different wall thicknesses ranging from 1.5mm to 3.5mm with the objective of achieving the factor of safety of minimum 1.5 for a minimum thickness. This test resulted in the final selection of motor thickness of 3mm with a rib protrusion of 3mm for the chosen motor cover.

- The VR and AR optimisation of motor parts was performed to stabilise the air gap and increase the motor power density by space utilisation. Using the final motor design, its parts were optimised using stereoscopic and auto stereoscopic displays for space utilisation. This system used most of the space within the rim and allowed better fitment of subassemblies within-wheel, whereby space was a very precious commodity. Finally, AR techniques were used on the key motor parts using Cyber glove II® and EON® interactive tools to interact in real time, thus expediting the final stages of motor design. These techniques achieved easy assembly of motor subassemblies, magnetic paths in particular. Each of the rotors and stators were packed to form the subassembly within the motor assembly and the crucial 1mm air gap was retained.

- The temperatures within the motor, in particular those that stators were subjected to, can increase to 250°C (a 1.4 factor of safety based on 180°C insulation). The aluminium bushes held these stators in position and any deflection due to the increase in temperature would result in compromising the 1mm air gap. H class (IEEE 11-2000 standard) insulation was used for: i) copper wire insulation, ii) polyester sheets (low thermal conductivity) insulation for stator slots, iii) insulation paint (varnish) to wound stator and iv) stators and bushes. Furthermore calculations
were performed to assess the effects with an assumption of insulation failure in between stators and bushes. These effects indicated that the greatest expansion was 0.53%, which was within the allowable limits. Consequently, the material selection was deemed safe for these bushes. Finally, forced fan cooling was chosen, Oriental (MB1040-B), which had specifications: i) air pressure 81.2CFM, ii) maximum static pressure 206pa, iii) airflow 2.3m3/min, iv) noise level 60dBA, and v) RPM 2750.

• Using the vehicle rolling resistance force (based on selected 20550R17 tyre), drag force to overcome the vehicle drag (based on the selected Holden Barina Spark), and gradient force (at 25%), the total longitudinal force was calculated at different speeds. The motor power and torque at different speeds were plotted. The torque and motor power at continuous rating were 24kW (both motors) at 60% of the rated speed (2100 rpm) with a peak torque of 96Nm. Similarly, based on the constant and peak value of the speed and the torque of the motor, the vehicle acceleration and the velocity were plotted against the acceleration time. From the plot, average acceleration value of 1.8m/s² and maximum acceleration value of 2.31m/s² were observed. Based on these motor power characteristic studies, two motors of 15kW power at rear wheels were established for the selected Holden Barina Spark.
Chapter 6

Primary mechanical braking system

6.1 Chapter overview

A new mechanical brake design was required for the in-wheel SRM, since the motor occupied the space generally used by the brake system in the Holden Barina Spark. This chapter focuses on the development of a low cost primary mechanical braking system. The following are detailed in this chapter:

- The brake system was designed in accordance with ADR standards. These standards were studied for brake performance during hot and cold tests. The standards were gauged and parameters were utilised for establishing compliance to ADR.

- The goal of this chapter was to develop a primary mechanical braking system with the existing electronic brake within the Holden Barina spark and the developed motor controller. Hence mechanical brake configurations were compared for selecting an appropriate brake with a caliper for an in-wheel SRM to fit within ~70mm space. Hence mechanical brakes were compared and low cost material was selected.

- An understanding of the impact of thermal changes and structural responses was important in the selection of the disc brake. Depending on the size of thermal changes and the materials involved, temperature changes could lead to warping and many other unwanted consequences. For example, friction as applied in brake systems creates heat in some areas of the disc structure. These temperature changes lead to warping, which in turn causes unwanted noise. The thermal and structural properties of a rim-mounted disc brake were investigated with a view to finalise a design for the in-wheel SRM.

- The disc brake topologies were optimised using FE methods by studying the six designs for a as low temperature as possible, minimising temperature changes and their impact on structures. In addition, statutory braking requirements for passenger vehicles and EVs with regenerative braking were also used for designs. Experimental tests were conducted on the Holden Barina Spark and compared with a large car and motorcycles to determine braking force and temperature requirements.
• FE methods were used to evaluate and optimise disc brake designs with respect to structural and thermal performance, and integration studies. Accordingly an optimised disc brake design was selected.
• A caliper suitably fitting in the ~70mm space available (as discussed in chapter 3) between the chassis and the suspension damper (with 60mm diameter) was finalised for the Holden Barina Spark.

6.2 Brake design considerations

6.2.1 Performance standards for brake design

A study was conducted of existing regulations to determine the testing conditions for brake designs. In Australia, standards on braking system for vehicles are categorised based on vehicle types: i) passenger cars use ADR 31, ii) motor cycles use ADR 33 and iii) heavy vehicles use ADR 35 standards. In this section, ADR 31/02 specific to passenger cars brake performance is discussed (Department of Infrastructure-Victorian Government 2009). The findings from ADR 31/02 were used to establish the brake performance for the Holden Barina Spark.

In the ADR 31/02 standard, the following conditions were essentials to be met for all tests: (i) tyres cold at the start of the test, (ii) road surface to have good adhesion, (iii) no wind is considered, (iv) actuation force on the pedal not greater than the prescribed maximum, (v) wheels not able to lock above 15km/h, (vi) vehicle not able move outside a . m lane, and (vii) yaw angle of ≤1 °. The following additional conditions were required for EVs with motors permanently connected: tests must be performed with electric motors connected, and for vehicles fitted with regenerative braking system functioning, as ABS tests were performed on a road surface with poor adhesion. Additional conditions specific to vehicles operating regenerative braking included the capacity to apply the mechanical brake in the case of a break in the wiring of the electric transmission or after the start switch was turned off. Where there was a failure of energy source for regenerative braking, the system was capable of delivering full brake control (Choi and Lee 2004). The designed SRM used brake by wire with a motor controller; however as per the ADR 31/02, primary mechanical braking is a mandatory requirement for all vehicles sold in Australia. Hence a mechanical brake design was essential for the developed in-wheel SRM.
The brake performance determining factors were stopping distance and mean fully developed deceleration. As per ADR 31/02, stopping distance was measured from when the driver first actuated the braking system. Initial speed had to be within 98% of the prescribed speed. The measuring equipment had to have an accuracy of +/-1% of mean fully developed deceleration $d_m$ was calculated using the following Equation 6.1:

$$d_m = \frac{V_b^2 - V_d^2}{25.92(S_e - S_b)}$$

(6.1)

Where, $V_0$ was the initial vehicle speed in km/h, $V_b$ was vehicle speed at 0.8 $V_0$ in km/h, $V_e$ was the vehicle speed at 0.1 $V_o$ in km/h, $S_b$ was the distance travelled between $V_0$ and $V_b$ in metres, $S_e$ was the distance travelled between $V_0$ and $V_e$ in metres.

6.2.1.1 Ordinary performance test with cold brakes

This is also known as “Type-0 test” and is a general performance test with cold brakes, performed laden and repeated unladen, with the temperature of the brakes between 65°C and 100°C. The vehicle was required to achieve satisfactory results in both mean fully developed deceleration and stopping distance. The conditions for the test are set out in Table 6.1 below. In Table 6.1, $V$ was the test speed in km/h, $S$ was the stopping distance in metre; $d_m$ was the mean fully developed deceleration in m/s$^2$, $f$ was the force applied to foot control in daN, and $V_{max}$ was the maximum speed of the vehicle in km/h.

**Table 6.1: Ordinary performance test conditions**

<table>
<thead>
<tr>
<th>S no</th>
<th>Description of test</th>
<th>Acronyms</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A)</td>
<td>Type-0 test with motor disconnected</td>
<td>$V=$</td>
<td>100km/h</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$S &lt;$</td>
<td>$0.1V + 0.0060V^2$ (m)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$d_m &gt;$</td>
<td>6.43m/s$^2$</td>
</tr>
<tr>
<td>(B)</td>
<td>Type-0 test with motor connected</td>
<td>$V=$</td>
<td>80% $V_{max} &lt; 160$km/h</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$S &lt;$</td>
<td>$0.1V + 0.0067V^2$ (m)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$d_m &gt;$</td>
<td>5.76m/s$^2$</td>
</tr>
<tr>
<td></td>
<td>Foot control</td>
<td>$f=$</td>
<td>6.5 - 50daN</td>
</tr>
</tbody>
</table>
6.2.1.2 Fade and recovery test for hot braking

Fade and recovery test for hot braking is also known as “Type-I test”. For the fade test, the brake system was heated by repeated actuation with the vehicle fully laden, under the conditions shown in Table 6.2 below, at the end of which the Type-0 test was repeated with the motor disconnected. The vehicle must achieve 75% of cold performance, or stopping distance \(0.1V + 0.0080V^2\), mean fully developed deceleration of 4.82m/s\(^2\) (i.e., 75% of 6.43m/s\(^2\)). EVs with regenerative braking were not required to use the system during the test and the motor was connected in the highest gear.

Table 6.2: Fade and recovery test, brake system heating conditions

<table>
<thead>
<tr>
<th>(V_1) (km/h)</th>
<th>(V_2) (km/h)</th>
<th>(\Delta t) (sec)</th>
<th>(n)</th>
<th>(d_m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>80% (V_{max}) &lt; 120km/h</td>
<td>0.5(V_1)</td>
<td>45</td>
<td>15</td>
<td>3</td>
</tr>
</tbody>
</table>

For recovery performance, the following procedures were performed; i) four stops from 50km/h, motor connected \(d_m\) 3m/s\(^2\), ii) 1.5km between each cycle. Between stops, it was necessary to accelerate to 50km/h and maintain static speed until the next stop. To test recovery performance, the Type-0 test was repeated from Table 6.1. Results were required to be \(\geq 0\%\) and \(\leq 150\%\) of cold performance results. ADR 31/02 testing requirements for passenger car brake systems were summarised as follows:

- **Cold performance:** ADR 31/02 Type-0 cold braking tests require the braking system to achieve a deceleration of 6.43m/s\(^2\) with the motor disconnected and 5.76m/s\(^2\) with the motor connected. From 35km/h the vehicle must stop within 12.5m and achieve peak deceleration of at least 5.76m/s\(^2\).
- **Hot performance:** For the ADR Type-I hot braking fade test, the vehicle must produce 75% of the Type-0 deceleration result or at least 4.82m/s\(^2\). For the ADR Type-I hot braking recovery test, the vehicle must produce between 70% and 150% of the Type-0 test.

6.2.2 Brake design parameters

A review of the existing literature on the development of mechanical braking systems relevant to EVs was conducted to evaluate current brake design parameters. Much of the research conducted to date was on the behaviour of automotive disc brake rotors and friction materials focused on optimisation of desirable performance characteristics, such as: i) improved thermal performance, ii) durability of friction materials, and iii) the
reduction of undesirable characteristics such as noise and vibration through structural/thermal rigidity (McPhee and Johnson 2008, Triches Junior, Gerges et al. 2008, Nishiwaki, Fujioka et al. 2009). McPhee and Johnson also concluded from the experiment that in order to increase convection heat flow mechanisms in vented disc brakes needs topology characterisation and optimisation (2008).

The braking system is of critical importance from a safety perspective and its correct function is highly regarded by customers (Breuer and Karlheinz 2008). Apart from the obvious safety considerations, structural integrity of the brake system is critical to product’s success. As much as 80% of the effort in braking system design and testing focuses on maximising driver comfort (Breuer and Karlheinz 2008). Vibration and noise, manifested as brake squeal and judder, is not acceptable to the customer, with customer surveys showing that brake noise is considered to be a critical quality indication (Triches Junior, Gerges et al. 2008). Warranty costs in the US resulting from noisy brakes have been estimated at US$ one billion per annum (Breuer and Karlheinz 2008). Low frequency vibrations (0-1000Hz) are the result of disc brake thickness variation causing incomplete contact with the friction material surface, or from non-uniform expansion of the disc brake under a heavy load, which prevents uniform pressure distribution, localised increased load and uneven wear. This phenomenon is known as frictionally excited thermo elastic instability and it causes wear resulting in the material damage. The driver experiences these vibrations as judders in the steering wheel. Vibration in the frequency 1000-3000Hz has been attributed to a stick-slip mechanism and is experienced as a howl or groan (Breuer and Karlheinz 2008). High frequency vibration is the result of physical, geometric or dynamic instability, producing frequencies above 3kHz that affect an area of high sensitivity of the human ear, and that are experienced as an unpleasant whine (Breuer and Karlheinz 2008). Earlier studies on use of different compositions and tribology of friction materials on brake pads in dry and wet conditions demonstrated noise control (Bijwe 1997, Eriksson, Bergman et al. 2002, Chan, Stachowiak et al. 2004). However brake comfort to a large extent is driven by the disc design itself.

Considerable study has been conducted on the behaviour and performance of motorcycles during braking. Optimal motorcycle braking strategies were investigated by Sharp (2009). Motorcycles featuring a rim mounted disc brake design have been in
production since 2004 (Buell Motorcycles 2004). The brake systems used in these designs were light weight discs. The light weight is achieved by improving surface area for heat absorption with a larger diameter and thinner structures.

### 6.2.3 Mechanical brake selection

Amongst several brakes and caliper design configurations used in passenger cars, the following is broader classification: i) friction, ii) pump, and iii) electromagnetic. Friction brakes are the most common and offer friction between the rotating parts and stationary parts using a pad to decelerate the vehicle. The fuel pump braking is used in ICE vehicles where it is part of the engine and the fuel flow is stopped to decelerate the vehicle. In electromagnetic braking, motor polarity is reversed, whilst converting the heat energy of brakes to store into batteries. An EV or a HEV with a regenerative braking or kinetic energy recovery brake system (KERBS) is further hybridised by integration with motor controllers. Often in a vehicle combination different classifications are used and are additionally supplemented by an electronic controller and sensors to communicate with other vehicle subsystems (e.g., steering, suspension). This system is also known as brake by wire and is an example of a system using an electronic assistance or a control to increase the efficiency. Each car manufacturer uses different terminology for the brake system with a specific controller design. Some the common terminologies used are: i) electronic stability controller (ESC) or electronic stability program (ESP), ii) anti-lock braking system (ABS), and iii) electro hydraulic brake (EHB) or electronic mechanical braking (EMB). In ESC and ABS the electronic component of the braking system has the potential to integrate with other electronic control systems such as motor and suspension, to act as a traction and stability control system during deceleration.

The selected Holden Barina Spark was supplied with ABS and moreover the motor controller was designed to integrate regenerative braking and ABS capabilities. Hence, the primary goal of this chapter was to develop a primary mechanical braking system. As a consequence two mechanical friction brake configurations were considered for selecting an appropriate design suitable for an in-wheel SRM, mainly: i) drum and ii) disc.
6.2.3.1 Drum brake

The drum brake consists of brake shoes mounted to the stub axle and a brake drum mounted to the axle (shown in Figure 6.1). The brake shoes are pressed radially against the friction lining on the inside of the drum by hydraulically actuated cylinders. After braking, a return spring releases the shoes from the drum.

Among the drum brake mechanisms available, the most commonly used types are: i) simplex, ii) duplex, iii) duo duplex, iv) servo, v) duo servo; figures are attached in Appendix 11, Figure 11.1 (Breuer and Karlheinz 2008). In the simplex drum brake design, the brake shoe in the forward direction of rotation provides 65% of the brake force generated, and that in the rear direction provides the remainder. The front shoe may have a thicker lining to compensate for the uneven wear. Both shoes are anchored to the same point. The brake factor generated by this design was in the order of 2 to 2.3. This design, generally found on the rear of lightweight passenger cars, provides specific advantages by incorporating the service brake and the parking brake and is cost-effective. In the duplex drum brake design, the two shoes are identical, each with their own mounting point and actuating cylinder. This design exhibits a self-actuating effect and generates a brake factor between 2.5 and 3.5. Difficulties in modulating this braking force and incorporating a parking brake make the implementation of this design less common. Duo Servo drum brakes feature the two brake shoes arranged in series to produce a very high brake factor (between 3.5 and 6.5) and are suited to small- and medium-sized commercial vehicles. Wear was taken up by simple manual adjustment of the brake at the required intervals.

As shown in Appendix 11, Figure 11.2, materials used for the drum brake are: i) grey cast iron, ii) aluminium casting, cast iron insert, and iii) aluminium/ceramic cast composites (Breuer and Karlheinz 2008). In cost sensitive cases, the use of the drum is
common as grey cast iron is used. Where weight was a consideration, aluminium outer with cast iron inserts were used, or ceramic aluminium cast composites in some cases. For this research, low cost and easily maintained drum brakes were advantageous. However the lack of space within the rim, due to the housing of the electric motor made it difficult to use drum brake design, hence disc brake was further investigated.

6.2.3.2 Disc brake

Disc brakes feature a disc or rotor mounted on the axle, and a caliper containing hydraulically actuated cylinders provides a braking torque via an axial clamping force on the friction material (shown in Figure 6.2). Disc brakes exhibit smooth, predictable ease of access to friction material for maintenance, and reliable mechanical and thermal performance. For these reasons they are almost universally used at the front of passenger cars.

![Disc brake schematic front view](image)

**Figure 6.2: Disk brake schematic front view**

The disc brake systems are classified based on calipers used as: i) fixed (Appendix 12, Figure12.1), ii) frame (Appendix 12, Figure12.2) and iii) fist (Appendix 12, Figure12.3). Fixed calipers have opposed cylinders arranged on both sides of the rotor and the caliper housing is fixed rigidly. Frame calipers have one brake cylinder on the inboard side of the rotor which transmits a clamping force to the outer side of the rotor via a ventilated frame which allows cooling air to reach brake pads. The resulting low operating temperature is a benefit of this design. The fist caliper also has one piston on the inboard side. It is a compact design suitable for use where space is limited such as front wheel drive cars. Calipers are generally cast from spheroidal graphite cast iron or aluminium. Based on the space availability of smaller than ~70mm from an earlier study
on the Holden Barina Spark (chapter 3), the fist caliper was selected (explained in detail in a section 6.5 of this chapter).

Nearly 90% of the kinetic energy is converted to heat during braking by the hydraulic friction brake and is absorbed by the disc; hence the disc has a thick cross section and serrations (shown in Figure 6.2). This energy is transferred via convection to the surrounding air. High performance applications such as racing, or heavy use such as downhill braking while towing, mean that the discs can reach temperatures of up to 700ºC (Breuer and Karlheinz 2008). Ventilation slots or serrations cast into discs and cross drilling aid the cooling process. Passenger car discs are predominantly cast from grey cast iron GCI15-GCI25 grades. Motor cycle discs use 410 and 420 stainless steel grades. Carbon ceramic matrix composite (C-SiC) discs have been developed for high performance applications such as racing. This material offers many benefits, including a service life of up to 300,000km, 60% reduction in weight, high corrosion and temperature resistance. Research on carbon and silicon carbon composite friction materials was conducted and they were reported as having “e cellent tribological properties”. It also detailed a manufacturing method directed at reducing cost and increasing uptake across the industry (Li, Xiao et al. 2010). It was reported that further work was under way in various institutes dedicated to reducing the cost of these disc brakes, yet cost remains a factor limiting their widespread adaption (Li, Xiao et al. 2010). Though C-SiC discs offered an attractive possibility of reducing unsprung weight with a rotating mass in the order of 4kg per disc, the high cost factor limited their use in the research. This is also the reason these rotors have not found wide acceptance in the market. 410 or 420 grade stainless steel was relatively low cost material with wide availability, good thermal properties and good corrosion resistance.

Choosing between the drum and disc, a disc brake made from low cost steel was more suitable as it was able to fit inside the available space. The drum on the other hand has to be located inside the motor where there was no space available. Use of a composite would add substantial cost to the braking system. A compact fist caliper design was required based on available space. Hence, the disc brake with 420 grade stainless steel was further developed.

One of the primary goals of the disc design was the reduction of the rotating mass and thus the gyroscopic force, and consequently a reduced force was required to actuate
steering. Reduction of mass was more of a concern than reduction of the gyroscopic force in the case of the in-wheel electric motor, due to the relatively high ratio of unsprung weight to total mass and the increased rotating mass of the motor. The concept designs developed a low cost primary hydraulic braking system, specifically, dimensional optimisation of the disc brake component of the system.

6.2.4 Disc brake concepts

The disc brake was investigated to develop an optimised concept based on key findings from earlier study and summarised as follows:

- It is a prudent safety consideration to require that the service brake has a mechanical actuation and hydraulic delivery mechanism, in compliance with the ADR 31/02 regulations in the event of a failure of the electrical braking system.
- As the selected Holden Barina Spark was supplied with an ABS and the motor controller was designed to integrate regenerative braking, hence the primary aim was to develop a mechanical brake system to integrate with the vehicle. Between the drum and disc brake, a disc brake made from low cost steel was more suitable for an in-wheel application due to the ~70mm available space within the Holden Barina Spark.
- Minimising frictionally excited thermo elastic instability in passenger vehicles is a key factor in brake comfort. From earlier study on the literature, thermal expansion has been found to be a key factor affecting the friction material contact ratio. Vibrations in vehicle disc brakes have been reduced by use of appropriate thickness materials for the disc. Motorcycles have successfully implemented a rim mounted disc brake system with a larger diameter and thinner structure. Further study was needed to develop disc concepts and to determine suitability of the disc concept for an in-wheel application.

The disc brake was designed as an integral part of the new proposed SRM. The disc brake weight and thermal handling was a challenge. The available off the shelf disc brakes were not suitable to use due to space limitation and heavier designs. The design objective was to have a light weight disc brake, which provided maximum heat absorption. To achieve the objectives, six disc brake designs were created with a larger diameter and thinner structure. This improved the surface area to maximise stiffness and improve heat absorption during braking. Six conceptual disc brake designs were created for topology optimisation for heat absorption.
As shown in Figure 6.3 disc brake 1 was a simple circular design with six tabs for external bolt holes. Disc brake 1 had a 4.41kg weight and used 420 stainless steel grade material.

![Disc brake 1](image1.png)

Figure 6.3: Disc brake 1

As shown in Figure 6.4 disc brake 2 was designed with six filleted tabs for external bolt holes and a larger surface area than disc brake 1. It was relieved with 18 holes on the circular faces and used 420 stainless steel grade material. The total weight of disc brake 2 was 4.47kg.

![Disc brake 2](image2.png)

Figure 6.4: Disc brake 2

As shown in Figure 6.5 disc brake 3 was designed with six protruded tabs for external bolt holes and a larger surface area than disc brake 2. It was relieved with 54 holes on the circular faces to minimise the weight increase. Disc brake 3 had 4.10kg weight and used 420 stainless steel grade material.
As shown in Figure 6.6 disc brake 4 was designed with six protruded tabs for external bolt holes and was relieved radially as compared to disc brake 3. Disc brake 4 had 4.10kg weight and used 420 stainless steel grade material.

As shown in Figure 6.7 disc brake 5 was designed with six protruded tabs for external bolt holes and has an increased width radially compared to disc brake 4. Disc brake 5 had 5.2kg weight and used 420 stainless steel grade material.
As shown in Figure 6.8 disc brake 6 was designed with six protruded tabs for external bolt holes and was the same geometry as disc brake 5 with additional cooling holes. Disc brake 6 had 4.13kg weight and used 420 stainless steel grade material.

6.3 Brake ADR compliance

With a given design it was important to determine that the brake forces of a selected vehicle met ADR guidelines. In order to examine the compliance, the following were accomplished in this section: i) determining the required brake force from the Holden Barina Spark through experimentation and ii) using experimental results, establishing the brake force distribution of the Holden Barina Spark and compliance to ADR.
6.3.1 Brake force experimentation

In the case of cars, the actuation force is the initial braking force generated by pressure on a pedal, amplified by the lever ratio of the pedal pivot point to master cylinder actuation piston shaft pivot point, summed with additional force provided from a vacuum booster. The delivery system converts actuation force to hydraulic pressure in a twin cylinder master cylinder, which, via two independent circuits, is transmitted to wheel brake cylinders, the function of which is to provide clamping pressure on the friction surfaces of the brake system. The friction surfaces create a retarding torque that is transferred to the road via the tyres. Figure 6.9 demonstrates different phases, where the braking speed in the car has been plotted with respect to time intervals. Speed, pedal force and braking were also plotted at time intervals. The following are the braking phases of the car: i) $V_0$ is when an obstacle is met, ii) $t_c$ is when operator perceives the obstacle, iii) $t_{sh}$ is when a decision is made, iv) $t_r$ is reaction time, v) $t_{Act}$ is time required to develop perceptible deceleration vi) $t_{Act, sw}$ is actuation time and pedal force development time vii) $t_{res}$ is the brake system response time, viii) $t_{sw}$ is force development time – from the brake system response time until development of 75% maximum braking force, and ix) $t_f$ is the maximum braking time, release of the brake system at standstill.

![Brake phases diagram](image)

Figure 6.9: Brake phases

The proposed disc brake was required to meet the required brake force. In order to determine required brake force, data was gathered on the configuration and performance of existing vehicles, mainly the Holden Barina Spark. Additionally a large car and a motorcycle data were also compared with the Holden Barina Spark data. The large car represented the highest weight and the motor cycle represented the lowest weight.
Vehicles representative of these segments were identified and studied for brake performance experimentally. The brake system components were measured for the Holden Barina Spark (small car), the Holden Captiva (large car), and the Buell (motorcycle) as listed in Table 6.3.

Table 6.3: Braking system data, small car, large car and motor cycle

<table>
<thead>
<tr>
<th>Segment</th>
<th>Small Car</th>
<th>Large Car</th>
<th>Motorcycle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Make and model</td>
<td>Holden Barina Spark</td>
<td>Holden Captiva</td>
<td>Buell</td>
</tr>
<tr>
<td>Total mass (kg)</td>
<td>960</td>
<td>1890</td>
<td>280</td>
</tr>
<tr>
<td>L2 (brake pedal pivot to pedal in mm)</td>
<td>320</td>
<td>360</td>
<td>180</td>
</tr>
<tr>
<td>L1 (brake pedal pivot to pivot axis in mm)</td>
<td>80</td>
<td>100</td>
<td>10</td>
</tr>
<tr>
<td>Pedal ratio</td>
<td>4</td>
<td>3.6</td>
<td>18</td>
</tr>
<tr>
<td>Brake booster diaphragm diameter (mm)</td>
<td>200</td>
<td>240</td>
<td>N/A</td>
</tr>
<tr>
<td>Master cylinder piston (mm)</td>
<td>25</td>
<td>25</td>
<td>16</td>
</tr>
<tr>
<td>Front wheel cylinder (mm)</td>
<td>45</td>
<td>35</td>
<td>25</td>
</tr>
<tr>
<td>No of front wheel cylinders</td>
<td>1</td>
<td>2</td>
<td>8</td>
</tr>
<tr>
<td>Front disc brake diameter D (mm)/ radius of friction surface</td>
<td>250/45</td>
<td>280/55</td>
<td>385/30</td>
</tr>
<tr>
<td>Front disc brake thickness</td>
<td>8x2</td>
<td>10x2</td>
<td>5</td>
</tr>
<tr>
<td>Rear wheel cylinder</td>
<td>drum</td>
<td>40</td>
<td>drum</td>
</tr>
<tr>
<td>No of rear wheel cylinders</td>
<td>0</td>
<td>1</td>
<td>N/A</td>
</tr>
<tr>
<td>Rear disc brake diameter d (mm)</td>
<td>drum</td>
<td>disc</td>
<td>N/A</td>
</tr>
</tbody>
</table>

The testing of these three vehicles was conducted using two testing devices: i) Brake Testa Millennium (as shown in Appendix 13 Figure 13.1) and ii) an accelerometer. For testing with a Brake Testa Millennium, a sensor was placed on the foot brake pedal to record the actuation force. The test sequence was activated when a load was sensed. The device recorded various braking force values generated in this process and printed as a report. Accelerometer data was also collected to compare the results from the Brake Testa Millennium. Three tests were conducted, as shown in Table 6.4, for the Spark and Captiva, and the mean value was used in the development of the new brake design. The Spark was used to conduct experiments at 60km/h and 100km/h to stopping. The examples of Spark averaged at 60km/h and averaged at 100km/h were shown to demonstrate brake force generated. For the Spark, the accelerometer and Brake Testa Millennium data were in the range of 0.85 to 0.9g for 60km/h to stopping (Figure 6.10 and Brake Testa Millennium data in Appendix 13, Figure 13.2). Table 6.4 represents the three tests carried out to examine the mean braking force required for the study from an average of 60km/h.
Table 6.4: Brake test results using Brake Testa Millennium, 60-0km/h

<table>
<thead>
<tr>
<th>Make of vehicle</th>
<th>Test cycle</th>
<th>Pedal maximum (N)</th>
<th>Maximum deceleration (G)</th>
<th>Average deceleration (G)</th>
<th>Test Duration (S)</th>
<th>Speed of test (km/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holden Barina Spark</td>
<td>Test 1</td>
<td>143</td>
<td>0.91</td>
<td>0.62</td>
<td>2.8</td>
<td>62</td>
</tr>
<tr>
<td></td>
<td>Test 2</td>
<td>123</td>
<td>0.88</td>
<td>0.55</td>
<td>2.5</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td>Test 3</td>
<td>119</td>
<td>0.91</td>
<td>0.63</td>
<td>2.6</td>
<td>58</td>
</tr>
<tr>
<td></td>
<td>Mean</td>
<td>128</td>
<td>0.9</td>
<td>0.6</td>
<td>2.6</td>
<td>57</td>
</tr>
<tr>
<td>Holden Captiva</td>
<td>Test 1</td>
<td>132</td>
<td>0.84</td>
<td>0.65</td>
<td>2.8</td>
<td>67</td>
</tr>
<tr>
<td></td>
<td>Test 2</td>
<td>193</td>
<td>0.99</td>
<td>0.71</td>
<td>2.6</td>
<td>66</td>
</tr>
<tr>
<td></td>
<td>Test 3</td>
<td>266</td>
<td>1</td>
<td>0.76</td>
<td>2.5</td>
<td>69</td>
</tr>
<tr>
<td></td>
<td>Mean</td>
<td>197</td>
<td>0.94</td>
<td>0.71</td>
<td>2.6</td>
<td>67</td>
</tr>
</tbody>
</table>

Figure 6.10: Holden Barina Spark-Accelerometer data at 60-0km/h

For the Spark, the accelerometer and Brake Testa Millennium data were in the range of 0.8 to 0.9g for 100km/h to zero (Figure 6.11 and Brake Testa Millennium data in Appendix 13, Figure 13.3).
Figure 6.11: Holden Barina Spark-Accelerometer data at 100-0km/h

The deceleration (brake force) and time values calculated by the accelerometer and Brake Testa Millennium at 60km/h and 100km/h were found to be within 10% of deceleration value for all the vehicles. By testing with different methods and different vehicles, it was ensured that the values were identical with only slight adjustments to cover vehicle and driver reactions. The maximum deceleration developed by each vehicle was calculated using 0.85g or 8.428m/s$^2$ which were above the ADR 31/02 cold and hot test requirements.

6.3.2 Brake ADR compliance study

Theoretical calculations were made to verify that the proposed rim rotor mounted brake design would meet ADR requirements. Earlier performance standards study discussed in this chapter (Australian Design Rule 31/02 - Brake Systems for Passenger Cars) was used. This specifies design and performance requirements based on testing procedures required in order to gain certification on Australian roads for passenger vehicles. The braking forces were required to achieve a peak deceleration of 5.8m/s$^2$. The following Equations 6.2 to 6.6, were used to calculate the braking force distribution at the front and rear of the vehicle at peak deceleration of $\geq$ .8m s$^2$. Hence, $F_b$ is total braking force in Newtons (N) given by Equation 6.2:

$$F_b = m_v, a_v$$  \hspace{1cm} (6.2)
Where, 1560kg was total vehicle mass \((m_v)\), and total deceleration was 5.8m/s^2 \((a_v)\). Then total braking force required was 9.05kN. The selected vehicle has been manufactured and marketed by General Motors in various markets as the ‘atiz’ (in the European market) and ‘Spark’ (in the Australian market). Configured with an ICE, it had a total mass of 960kg. The removal of the ICE and associated systems balanced the installation of batteries and electric motors made vehicle mass with passenger’s weight 1560kg. Wheelbase and CG dimensions are shown in Figure 6.12. With an in-wheel SRM and related components an additional 6% variation was observed in the longitudinal direction (discussed in chapter 8). As this 6% variation was small, it was neglected for brake force distribution calculations.

![Figure 6.12: Centre of gravity in the Holden Barina Spark (GM 2012)](image)

Calculation of front/rear wheel braking force bias was based on front/rear weight bias of the vehicle and on the CG as shown in Figure 6.12. It was calculated using Equation 6.3 as follows:

\[
\frac{F_{wr}}{F_{wt}} = \frac{l_f}{l} \tag{6.3}
\]

Where, \(l_f\) is distance of CG to front axle of wheelbase in mm (975), \(l\) is wheelbase length in mm (2375), \(F_{wt}\) is total weight of the vehicle in kg, \(F_{wr}\) is weight distribution on rear axle in kg (640), \(F_{wf}\) is weight distribution on front axle in kg (920). Hence the weight on front was calculated using Equation 6.4:

\[
F_{wf} = F_{wt} - F_{wr} \tag{6.4}
\]

The braking force front wheels were determined using Equation 6.5:
Similarly braking force rear wheels were determined using Equation 6.6:

\[
\frac{F_{br}}{F_b} = \frac{F_{wr}}{F_{wt}}
\]  

(6.6)

From the above calculations it was evident that a braking force of 1.9kN (3.71kN on both rear wheels) was required at each of the rear wheels and 2.7kN (5.34kN on both front wheels) was required at each of the front wheels to achieve a peak deceleration of 5.8m/s\(^2\) (as per ADR) as measured at the tyre-road interface. These calculated values were used as inputs for the design of the disc brakes. Vehicle deceleration calculations were important to ensure that these designs met the performance criteria set by ADR. Stopping distance was regulated by mean fully developed deceleration, which was calculated according to the difference in the velocities at the distance at specific time intervals. As per the ADR, the vehicle decelerates from 35kmh-0kmh within 12.5m; mean deceleration was calculated as per Equation 6.7:

\[
S_{dev} = \frac{V_b^2}{2a_m}
\]  

(6.7)

Hence mean fully developed deceleration was required between both the required peak deceleration (5.8m/s\(^2\)) and mean deceleration (3.8m/s\(^2\)) range, and hence it followed that if both other conditions were met this condition was achieved. Thus, using ADR 31/02, \(a_m\) mean fully developed deceleration in m/s\(^2\) was calculated by the following Equation 6.8:

\[
2a_m(S_e - S_b) = V_b^2 - V_e^2
\]  

(6.8)

Where, \(V_0\) was the initial vehicle speed in 35km/h, \(V_b\) was the vehicle speed at 0.8\(V_0\) i.e., 28km/h, \(V_e\) was the vehicle speed at 0.1\(V_0\) i.e., 3.5km/h, \(S_b\) was the distance travelled between \(V_0\) and \(V_b\) in metres i.e., 2.93m, \(S_e\) was the distance travelled between \(V_0\) and \(V_e\) in metres i.e., 9.97m. Hence average deceleration (\(a_m\)) was calculated as 4.2m/s\(^2\). This value was more than the required mean deceleration of 3.8m/s\(^2\).
6.4 Brake disc optimisation

6.4.1 Brake thermal experimentation

From the earlier brake design parameter it is evident that thermal performance of the disc brake is critical for the design. During braking, kinetic energy is converted into heat via friction and transferred from the friction surface into the disc brake and to a lesser degree, into the friction material, into the caliper and brake fluid, and into the surrounding environment via convection. The capacity of the disc brake to absorb this heat energy has a direct impact on the performance of the system, and is proportional to the thermal mass. Larger disc brakes have a greater capacity to absorb heat energy and consequently the system experiences less ‘fading’, as a decrease in the frictional force at the friction surface results in temperature rises. Stored heat radiates to the brake fluid via the caliper, further reducing the capacity to deliver braking power. It was therefore essential to correctly design the thermal mass to balance the heat flow in and out to maintain the system at a safe operating temperature. A standard test involves hard braking from 100-0km/h ten times with acceleration back to 100km/h between brake applications. The objective is to maintain a temperature below 700°C with no loss of braking performance over the test, as shown in Table 6.5 (Breuer and Karlheinz 2008).

Table 6.5: Typical values for thermal design, automotive brakes

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Disc brake surface temperature</th>
<th>Brake fluid temperature Mountain pass descent</th>
<th>Critical temperature Brake hoses</th>
<th>Critical temperature aluminium alloys</th>
<th>Brake fluid after high power stops</th>
</tr>
</thead>
<tbody>
<tr>
<td>Target Value</td>
<td>&lt;600 C</td>
<td>&lt;180 C</td>
<td>&lt;150 C</td>
<td>&lt;180 C</td>
<td>&lt;150 C</td>
</tr>
</tbody>
</table>

The disc brake temperature was recorded before and after each test using a Micro Temp MT100 digital thermometer operating in the temperature range -27° to 230°C, and increments of 0.1°C. Table 6.6 shows the calculations of effective mean temperatures for a small car, using a thermometer for the three experiments. Two sets of tests were conducted one from 60-0km/h and the other at 100-0km/h. As shown in Table 6.7, a Holden Barina Spark averaged a rate of 59°C in comparison with a large car and a motor cycle with approximately 30°C variance. At 60-0km/h, similar results were found with a slight decrease in temperature range.
Table 6.6: Average temperature of the Holden Barina Spark using MT100 thermometer, 100-0km/h

<table>
<thead>
<tr>
<th>Experiments</th>
<th>$T_{\text{start}}$ °C</th>
<th>$T_{\text{finish}}$ °C</th>
<th>$\Delta T$ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiment 1</td>
<td>20</td>
<td>73.6</td>
<td>53.6</td>
</tr>
<tr>
<td>Experiment 2</td>
<td>25</td>
<td>84.8</td>
<td>59.8</td>
</tr>
<tr>
<td>Experiment 3</td>
<td>30</td>
<td>93.3</td>
<td>63.3</td>
</tr>
<tr>
<td>Avg. temperatures</td>
<td>25.0</td>
<td>83.9</td>
<td>58.9</td>
</tr>
</tbody>
</table>

Table 6.7: Thermal variations before and after the event using MT100 digital infrared thermometer, 100-0km/h

<table>
<thead>
<tr>
<th>Type</th>
<th>Small car</th>
<th>SUV</th>
<th>Motor Cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{start}}$ °C</td>
<td>25</td>
<td>42.2</td>
<td>65</td>
</tr>
<tr>
<td>$T_{\text{finish}}$ °C</td>
<td>84</td>
<td>72</td>
<td>96.4</td>
</tr>
<tr>
<td>$\Delta T$ °C</td>
<td>59</td>
<td>29.8</td>
<td>31.4</td>
</tr>
</tbody>
</table>

The disc brake converted kinetic energy via friction into heat. Hence the total kinetic energy $KE$ was calculated as 627,200J, as shown in Equation 6.9:

$$KE = \frac{1}{2} m_v v^2$$

(6.9)

The mass of vehicle $m_v$ was 1560kg (from earlier calculations). In the case of the front and rear weight, which had a bias of 59:41, this energy was spread over the braking system front and rear in the same proportion, giving 370,038J (front wheels) and 257,152J (rear wheels). Consequently, the load for each front wheel was 185,024J and for each rear wheel 128,576J.

The results graph on Figure 6.10 and Figure 6.11, obtained during tests of the Holden Barina Spark, employing an electronically controlled braking system to maximise braking force, shows that the brake force developed from zero at $t_0$ to maximum at approximately and $t$ had the following sequences: i) at 1.0 second as the braking system was pressurised, ii) continuing on a level until approximately 2.5 seconds, and iii) falling off sharply to zero at 3.0 seconds. This was considered close to an ideal graph and was used as a template to develop a thermal load graph for a 100-0km/h braking event, a condition which more closely simulated an emergency braking event. The period of the 100-0km/h braking event was calculated from Equation 6.10:

$$V = V_0 + a \cdot t$$

(6.10)
Where $V$ was final velocity (0), $V_0$ was initial velocity (27.8m/s) and $a$ mean maximum developed braking force which was -8.8m/s$^2$ (from the Holden Barina Spark, 60-0km/h test results). Solving for $t$, the period was found to be 3.15 seconds. As seen in the test results graph, in reality the braking force was not applied instantaneously at maximum level, but ramped up at the start of the event and down afterwards.

### 6.4.2 Finite element methods

A FE method was used on the six disc brake designs to optimize and select an appropriate brake based on thermal performance under heat load conditions simulating 100-0km/h hard braking. Transient thermal analysis was conducted using the Ansys® test bench 13.1 with an ambient temperature of 22°C for the surrounding air.

#### 6.4.2.1 Finite element modelling

The FE models were developed for all six disc topologies in the Ansys® test bench 13.1, commercial FE software. Initially, all discs were modelled using the local and the global 3D tetrahedral mesh (solid mesh) with 4 nodes; an example of the brake disc 5 is shown in Figure 6.13. The disc 5 FE model consisted of 800 elements and 6442 nodes. The bolt holes, and fillet areas were modelled with a local tetrahedral finer mesh of 0.5 to 2mm. The global tetrahedral mesh of 5mm or more was modelled into the rest of disc 5 with a smooth transition ratio of 0.32. Transition ratio allowed a smooth flow from the local to the global mesh within the FE model.

#### 6.4.2.2 Material and boundary conditions

In the FE model the boundary conditions were modelled to simulate the disc brake 5 physical conditions. In this design, the disc brake was connected to the motor at bolt holes location as shown in Figure 6.13. The fixed support boundary conditions were used at these bolt holes to restrain rotational and translation movements.

The engineering data of 420 grade steel material (AS 2837-1986) used are i) Young’s modulus ($E$) 200GPa, ii) Poisson’s ratio ($v$) 0.28, iii) density ($\rho$) 7.8g/cc and iv) ultimate tensile strength ($\sigma_u$) 860MPa.
Figure 6.13: Disc brake 5 FE model with load and boundary conditions

6.4.2.3 Loading

Figure 6.13 shows the disc brake 5 example showing load conditions. The following equation was used as an idealised heat load. Ansys® test bench 13.1 replicated the load application condition found in the earlier experiments and the following is a representation, as shown in Equation 6.11:

When $x > 0 \geq 1$ then $y = \sin x \frac{hl}{(2t-1)}$  \hspace{1cm} (6.11)

Where $x$ and $y$ are heat load variants in Watts, $hl$ is heat load in Joules and $t$ is time in seconds. The values of $x$ and $y$ were determined at $1 < y > .1$, $y = 1$, $y > .1 \geq .1$, hence heat flow median was developed using Equation 6.12:

$$y = \sin(x - 2.15) \frac{hl}{(2t-1)}$$ \hspace{1cm} (6.12)

The area under the graph was equal to total heat load over the braking period. In the case of each rear wheel, where the heat load was 128,576 J, maximum $y$ value of the load was 30,982 J (half of the load for each wheel disc brake). The $y$ values of 15491W as a heat load on each disc brake were modelled on front and back friction faces. A time
stepped heat load modeled in the FE environment is illustrated in Figure 6.14. Convection of 17W/m²°C was modeled to the entire body.

Figure 6.14: Heat load input for each friction face on a disc brake

6.4.2.4 Finite element results

The results of the transient thermal analysis were used as the input load coupled to a static structural analysis to examine heat induced displacements and temperature rise on all the six discs. Key results are tabulated in Table 6.8 and maximum temperature representations of brake disc 5 are shown in Figure 6.17 and Figure 6.18. As shown in Table 6.8, results of the analysis of the initial design, disc brake 1, showed maximum displacement of 0.19mm at the point midway between fixings and reached a maximum temperature of 155°C.

Table 6.8: Disc brake 1 to 6 FE results

<table>
<thead>
<tr>
<th>Disc brake Design iteration</th>
<th>Volume (m³)</th>
<th>Mass (kg)</th>
<th>Maximum temperature (°C)</th>
<th>Maximum temperature at t=30s (°C)</th>
<th>Maximum displacements (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disc brake 1</td>
<td>5.69E-04</td>
<td>4.4133</td>
<td>155.3</td>
<td>106.93</td>
<td>0.1879</td>
</tr>
<tr>
<td>Disc brake 2</td>
<td>5.77E-04</td>
<td>4.477</td>
<td>152.45</td>
<td>106.5</td>
<td>0.162</td>
</tr>
<tr>
<td>Disc brake 3</td>
<td>5.29E-04</td>
<td>4.0963</td>
<td>166.38</td>
<td>113.36</td>
<td>0.1504</td>
</tr>
<tr>
<td>Disc brake 4</td>
<td>5.29E-04</td>
<td>4.0959</td>
<td>161.72</td>
<td>113.56</td>
<td>0.1815</td>
</tr>
<tr>
<td>Disc brake 5</td>
<td>6.48E-04</td>
<td>5.2107</td>
<td>104.12</td>
<td>103.47</td>
<td>0.1118</td>
</tr>
<tr>
<td>Disc brake 6</td>
<td>5.33E-04</td>
<td>4.1311</td>
<td>218.6</td>
<td>112.48</td>
<td>0.2371</td>
</tr>
</tbody>
</table>

A maximum displacement of 0.16mm that developed in disc brake 2 had stress reducing radii on the mounting tabs. It had a 152°C, which was evident across most of the friction surfaces.
Disc brake 3 featured additional cooling holes (54 holes). The cooling holes were ineffective, as evidenced by the increased maximum temperature of \(166.38^\circ\text{C}\). The radius to the outer perimeter was reduced between fixing points, resulting in reduced displacement at this point. An adding cooling hole, which was disc brake design 3, resulted in a maximum of \(0.2^\circ\text{C}\) better cooling than disc brake 4 (without cooling holes) after 30 seconds. From these results it was evident that the cooling strategy was not successful.

On disc brake design 4, the disc brake was relieved radially and displayed a higher maximum temperature of \(161.72^\circ\text{C}\). Corresponding areas of high deformation with a maximum deflection of \(0.18\text{mm}\) between mounting points were seen on this disc brake.

Figure 6.15: Disc brake design 5, maximum temperature

Disc brake design 5 displayed the lowest displacement of \(0.11\text{mm}\) compared to the other disc brakes, as shown in Figure 6.16. It exhibited the best thermal properties with a lowest maximum temperature of \(104.12^\circ\text{C}\) (Figure 6.15), and the lowest temperature of \(103.47^\circ\text{C}\) at 30 seconds (Figure 6.17). Disc brake 6 featured the same geometry as 5 with the addition of holes designed to cool the disc brake. Temperature results indicated that this strategy was ineffective, with the maximum temperature primarily affected by mass. Displacement was higher on this design, suggesting a greater susceptibility to frictionally excited thermo elastic instability and disc thickness variation, both of which conditions caused vibration in the brake system. The simulation results depicted a sharp rise in brake design temperature as the heat load was applied, and a sharp drop at 6
seconds to a point approaching a steady state, in the case of design 5 less than 3°C above the temperature after 30 seconds.

Figure 6.16: Disc brake 5, maximum displacement in mm

Figure 6.17: Disc brake 5, maximum temperature in °C at 30 seconds
Figure 6.18 is a summary of temperatures for all the disc brakes. Based on the results, disc brake 5 was chosen. Further studies were conducted on disc brake 5 to reduce thickness from 10mm and 5mm. The same rotor profile was used for both the thicknesses. The maximum temperature developed was 120.1°C with the 5mm thick rotor and 103.47°C with the 10mm thick rotor, as shown in Figure 6.19. From these studies, 10mm thick disc brake design was chosen for in-wheel SRM.

Figure 6.18: Disc brake design 1 to 6, temperatures at time intervals

Figure 6.19: Disc brake design 5 with 10mm and 5mm thickness
The transient thermal/static structural analysis results were used to optimise the thermal properties of the disc brake design. Designs returned maximum temperature results ranging from 218.6°C to 104°C. A temperature of 103.47°C on the final disc brake 5, was an acceptable result for the primary braking system for an emergency braking condition from 100-0km/h at 0.85g (8.35m/s²), unassisted by KERBS, with a safety factor of 1.2 applied to the vehicle mass. This represented the maximum expected load condition of the system, achieved under deceleration of 8.35m/s², 31% greater than the 5.8m/s² peak deceleration required by ADR 31/02. Analysis of the design of disc brake 6, which featured cooling slots, showed the peak temperature of 218.6°C. An ability to shed heat through increased surface area was evident with the temperature at 30 seconds down to 112°C. The greater peak temperature was a disadvantage given the proximity of the disc brake to the motor housing and the negative impact of rising temperature on the performance of the in-wheel SRM. Additionally, the cooling slots had a negative impact on the structural performance of disc brake 6, with a maximum displacement of 0.24 mm compared with 0.12mm for disc brake 5.

The high stress results returned were due to the mathematical singularities associated with the borders of fixed surfaces. The high stress result returned at these points was mitigated in a physical prototype by the incorporation of slotted mounting points that allowed the disc brake to expand radially. However, the simulation undertaken did not account for the effects of contact between the mounting fixing and the disc brake. Maximum deformation on disc brake 5 was 0.12mm outwards in a radial direction, located midway between the fixing points. When interpreted, this came about because the model was rigidly constrained at the fixing points in the analysis setup. In reality, 1.5mm of radial expansion was provided by the slotted mounting points so the expansion of the disc brake became uniform. The area experiencing the highest deformation was located at a point that did not touch the friction surfaces, which led to a high degree of confidence that the final design did not cause distortion resulting in frictionally excited thermo-elastic instability.
The final brake system design is shown in Figure 6.20 with an in-wheel SRM and suspension system. Disc brake design 5 displayed better thermal stability than the other designs, with an improvement of between 8% and 52% over the alternative designs. The design had the greatest mass of all those considered, at 5.2kg, 22% higher than any other. This figure was 42% lower than the existing small car disc brakes, which were in the order of 12kg.

### 6.5 Brake caliper selection

A study was undertaken of the caliper, and selection was based on the light weight and space accommodated by the caliper. As discussed in section 6.3.2.2 amongst three calipers available first was used in this research. The compactness of these calipers makes them ideal for front wheel drives. In this research due to limited packaging space of 0 mm, it’s an ideal candidate. Calipers are generally cast from spheroidal graphite cast iron or aluminium. The compact nature of the first caliper made it an ideal design candidate for use in this research, where packaging restrictions were severe (maximum ~70mm space).

The braking caliper is one of the fundamental systems of a motor vehicle, requiring relatively large amounts of development time. Disc brakes in this design used a flat disc brake made of 420 steel grade that is attached to the wheel hub. The process of a disc brake commences when the brake pedal is applied; it compresses hydraulic fluid through brake lines into the brake pistons. The pistons in turn compress the brake pads onto the disc brake resulting in the slowdown of the wheel. As the pads rub against the
disc brake, they cause an increase in friction: the kinetic energy produced by the friction is converted to heat via the brake pads (Kumar 2008). Using this design, the Barina Spark from Holden was implanted with two rear motors. This addition of a motor within the rear wheel rim required the relocation of the disc brake and caliper. The Holden Barina Spark model had stock rear brakes, as in the drum brake system. The positioning and mounting of the stock drum brakes clashed with the placing of the electric motor. The caliper selection was based on a ~70mm clearance between the MacPherson strut and the suspension system. Due to the limited space, an off the shelf smaller fist caliper was used. The in-wheel design selection of calipers was based on the following criteria:

- The space availability for the caliper was not significant: hence a small light weight caliper was selected. The caliper was chosen specifically for its efficiency in relation to functionality, so it was necessary to establish that the caliper could perform both normal braking and hand braking operations
- Braking performance was important, as it is a regulatory requirement; however braking effort was supplemented by regenerative braking inside the motor

The caliper chosen, manufactured by Gownloch Ducati, was a Brembo, 4 piston rear mount caliper, as shown in Figure 6.21. The caliper itself was a fist caliper with 4 opposed pistons. Like the majority of brake calipers on conventional cars, the brake pads were supported by one pin situated inside the caliper. The caliper was made from cast aluminium, which was light yet performed well under relatively elevated temperatures with high tensile strength. This caliper also met the constraint of fitting in the ~70mm gap between the wheel and suspension piston of the chosen vehicle.
6.6 Summary

The salient points of the research presented in this chapter are as follows:

- The brake design was evaluated to establish compliance to ADR 31/02 standard. The design required a peak deceleration (5.8m/s²) and mean deceleration (3.8m/s²) for ADR compliance. The literature review of the brake indicated that minimising frictionally excited thermo elastic instability in passenger vehicles is a key factor in brake comfort. The use of a composite was discounted due to the substantial costs of the braking system designs. Thermal expansion was found to be one of the key factors affecting the friction material contact ratio.

- Different mechanical brakes were compared, such as disc and drum. Choosing between these, a disc brake made from low cost steel was more suitable as it was able to fit inside the available space of ~70mm. The drum on the other hand has to be located inside the motor where there was no space available. A compact fist caliper design was required based on available space. Hence, the typical design of disc brake with 420 grade stainless steel was further developed.

- The six disc brake concept designs were developed with low rotating mass. The thermal stability was achieved by specific geometrical optimisation of the disc brake component. Experiments were undertaken using an accelerometer to examine the brake force generated for different car segments to establish the braking force for a passenger car with value of 0.85 to 0.9g. Similarly, temperature changes were
recorded for temperature differences before and after braking events based on ADR 31/02 standards. The calculations indicated a peak deceleration of 4.2m/s² mean deceleration of 3.8m/s², which ensured compliance with ADR requirements.

- Using these experimental data as inputs, thermal and displacement results were compared and analysed for the final brake design. FE methods were used to examine optimisation of different disc brakes. Among the different brake discs, the selected disc brake 5 exhibited the best thermal properties with the lowest maximum temperature of 104.12 °C and the lowest temperature at 30 seconds of 103.47°C. Over most of the disc brake 5, the lowest displacement observed was of value 0.12mm. It displayed thermal performance advantages compared to other designs, an improvement of between 8% and 52% more than alternative designs. The design had the greatest mass of all those considered, at 5.2kg, 22% higher than any other. This figure was 42% lower than existing small car disc brakes, which were in the order of 12kg. Compared to other disc brakes, the selected shape of the disc brake with 10mm thickness was used for the in-wheel SRM.

- Calipers fall into three categories, fixed, frame or fist. The brake caliper was finally chosen based on key selection criteria, including light weight, low cost and with sufficient clearance when fitted inside the wheel and MacPherson strut of the selected vehicle. The compact nature of the fist caliper made it an ideal design candidate for use in this research, where packaging restrictions were severe (maximum of ~70mm space). Hence an off the shelf Brembo 4 piston rear mount caliper manufactured by Gownloch Ducati, was selected.
Chapter 7

Evaluations of suspension system

7.1 Chapter overview

Suspension is an important part of the vehicle, as it determines the ride comfort and the vehicle stability. Consequently, it was closely analysed in this work, in an attempt to meet the following objectives:

- Developing a Holden Baina Spark suspension model for design evaluations. The in-wheel SRM developed has an increased mass when compared with an equivalent power output PMM. Consequently, the drawback of SRM is significant unsprung mass and hence could lead to undesirable vehicle driving characteristics. Thus, the aim is to recreate the Holden Barina Spark suspension design for evaluations.

- Evaluating suspension designs in context of in-wheel SRM (EV) and comparing it to the ICE using quarter-car models. Establishing key variations of two vehicle types. To develop the quarter car models of the selected vehicle suspension, the mathematical and computational simulations have been conducted. Analysing suspensions for dynamics, durability, and ride comfort. These analyses should include the comparison of the ICE vehicle performance with the in-wheel SRM for suspension of the Holden Barina Spark. Performance analysis based on the quarter-car models, focusing on the evaluation of the suspension for an in-wheel SRM, when subjected to road impacts. The MATLAB® and Simscape® analysis included using derived mathematical simulations. Simulation uses design scenario for driver seat, sprung mass and unsprung mass of the EV in context of the ICE car.

- Examining the effects of variable amplitude loads on suspension component by performing experiments with an increased mass (due to motor) in order to evaluate fatigue and conduct associated analyses for durability. This variable amplitude data used as an input for fatigue study of suspension design. The experimental data covered four load scenarios—the stationary vehicle, the vehicle moving at a constant velocity on a smooth road, the vehicle moving on a bumpy road, and the vehicle moving at a constant speed on a banked road. These scenarios simulated by applying a specific combination of the forces representing the load amplitude exerted on the suspension under the corresponding vehicle travel conditions. Thus,
in this work, the four scenarios simulate simplified version of the actual driving conditions, and thus variation in the load, using experimentation of the vehicle. Moreover, the loading cycles applied a corresponding incremental damage on the suspension parts, corresponding to the failure conditions. Therefore, the main aim of this analysis is to assess the suspensions life cycles for an in-wheel SRM.

7.2 Suspension modelling

Suspension is an important part of the vehicle design, as the aim is to ensure comfort during driving under a wide range of road and weather conditions. In this research, suspension analyses were carried out in order to assess the vehicle performance with an increased mass on each wheel. Both analytical and empirical methods were used to validate current suspension design of the Holden Barina Spark with an in-wheel SRM. Starting with quarter-car models, series of evaluations were performed, whereby the fatigue and other residual stresses were measured to validate performance of suspension elements. In modern vehicle designs, system validation and evaluation of its elements has become vital due to shorter life cycle approach, as the aim is to reduce the number of physical prototyping methods required (Firat and Kocabicak 2004).

Several studies were conducted on performance in vehicles, whereby the analysis was performed on a quarter-car model, which was compared to semi-active and passive system performance (Sohn, Hong et al. 2000, Georgiou, Verros et al. 2007). Quarter-car models were also used in studies of vibration response during ride conditions on roads (Gao, Zhang et al. 2008). Other authors applied Simulink-based models in order to obtain graphical output reflecting the design performance of the semi-active and passive suspension system (Chan and Sandu 2003). Half-car models were used to develop EV suspension control systems by other authors (Cao, Liu et al. 2007). The literature review summarised above was further expanded to include active and passive suspension adaptive control systems in order to assess their stability and dynamics (Cao, Liu et al. 2008, Cao, Song et al. 2011). The conclusion drawn from the results reported above is that, most of the cars utilised in the experimental studies reported in the literature used passive systems and the analysis was based on quarter-car model design to evaluate performance of the suspension elements with increased mass. Moreover, MATLAB® and Simscape® were used to effectively analyse suspension durability of an in-wheel SRM of EV in comparison with ICE car. This gave the starting point for the present
study, as the subsequent findings can be placed in the context of those previously reported by other authors working on similar issues.

Based on the studies identified during the above literature review, the same approach was adopted in the present study. Due to physical changes affected by the unsprung mass with the new in-wheel SRM, the vehicle response was also expected to change, as the vehicle suspension system remained unchanged. Thus, the aim of the analysis is to evaluate the changes due to the new in-wheel SRM. Firstly, the analytical method based on the MATLAB® software was employed to compare the two results. As a part of this process, MATLAB® program was written to solve the quarter-car equation for two different cases to study the changes brought in due to the new design. Subsequently, an empirical method using Simscape® was utilised, where a model was prepared for the quarter-car using same methods as those employed in the MATLAB® program.

### 7.2.1 Free body diagram

In this research, a quarter-car model was used for the optimisation, identification and control of the EV, when subjected to road impacts. A model with three degrees of freedom was used for the analysis. Here, the quarter-car model for the ICE vehicle (Holden Barina Spark) was compared to the quarter-car model of the proposed in-wheel SRM of the EV. The model had a sprung mass, referring to the mass of the car above the suspension system (the vehicle chassis), with the additional due to the driver-seat placed on top. These two components were insulated by the spring cushion that was, in the subsequent analysis, represented as a spring with a spring index $K_{ds}$ and damper with the coefficient $C_{ds}$. Underneath the sprung mass was the unsprung mass, which represented the mass of the single wheel, suspension, tyre, shock absorbers and suspension knuckles. In the case of the EV, the unsprung mass also included the mass of the in-wheel SRM, which was placed inside the wheel. In the subsequent analysis, the EV suspension was considered to have a spring stiffness of $K_s$ and the damping coefficient of the damper $C_s$. The tyre was also considered to act like a spring with a spring coefficient $K_t$. The model for passive suspension system of a quarter-car was thus ready for analysis. Figure 7.1 below provides the free body diagram of the quarter-car model.
Above, $X_r$, $X_u$, $X_s$ and $X_{ds}$ correspond to the vertical displacement of the road, unsprung mass, sprung mass and the seat, respectively. In Table 7.1, equation coefficients used in the subsequent text are given.

**Table 7.1: Descriptions of variables used in equations**

<table>
<thead>
<tr>
<th>Variables</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_{ds}$</td>
<td>Driver and seat mass</td>
</tr>
<tr>
<td>$M_s$</td>
<td>Vehicle sprung mass</td>
</tr>
<tr>
<td>$M_u$</td>
<td>Vehicle unsprung mass</td>
</tr>
<tr>
<td>$C_s$</td>
<td>Damping ratio of the vehicle suspension</td>
</tr>
<tr>
<td>$C_{ds}$</td>
<td>Damping ratio of the seat</td>
</tr>
<tr>
<td>$K_t$</td>
<td>Tyre stiffness</td>
</tr>
<tr>
<td>$K_s$</td>
<td>Vehicle suspension spring stiffness</td>
</tr>
<tr>
<td>$K_{ds}$</td>
<td>Seat suspension spring stiffness</td>
</tr>
<tr>
<td>$X_r$</td>
<td>Road input</td>
</tr>
</tbody>
</table>

The variables given above were assigned values for the corresponding car configuration. The driver-seat mass ($M_{ds}$) was considered to be a uniform 90kg for both car types, as it was identical in both experimental arrangements. The vehicle sprung mass ($M_s$) changed for the two configurations, as the vehicle kerb weight was modified. In the case of the ICE setup, the sprung mass was considered to be 110kg, which was the default load on each wheel for the Holden Barina Spark. This weight on each wheel was 240kg calculated based on kerb weight of the Barina Spark (960kg). Of the ~240kg total
weight above, 90kg was considered to be the weight of the driver and the seat. In addition, the measurements performed during the experiments indicated that each set of existing wheels, brakes and suspension weighed approximately 40kg, which was unsprung mass of ICE \((M_u)\). This lead to the estimate of 110kg for the ICE vehicle sprung mass.

The sprung mass for the EV \((M_{se})\) changed due to the ICE removed and weight of the motor placed inside the wheel. The conversion involved changing the full wheel and brake system with the new proposed in-wheel SRM and wheel. In the process, 1” wheels with SRM fitted inside were used as a replacement for the two rear wheels. These conversions increased the weight of the entire vehicle to 1200kg (including the battery weight as explained in chapter 8). Based on the overall weight; the sprung mass of the EV was calculated with aforementioned methods as \((M_{se})\) 130kg. Whereas the unsprung mass of an EV \((M_{ue})\) was 80kg in total calculated from: i) 40kg as weight of the in-wheel SRM (discussed in chapter 4), ii) 25kg as weight of the 1” wheel \((kg\ for\ rim\ discussed\ in\ chapter\ 3\ and\ 18kg\ for\ tyre\ nuts\ and\ bolts)\) and iii) 15kg as weight of a new brake system \((5kg\ for\ disc\ brake\ discussed\ in\ chapter\ 6\ and\ for\ caliper,\ hydraulics,\ nuts\ and\ bolts\ 10kg)\). While driver-seat mass of EV was 90kg.

### 7.2.2 Modelling

The parameters required for performing the analysis were the spring constants and the damping coefficients of the spring mass systems in the model. As previously noted, the model with two degrees of freedom was employed in the analysis, whereby the two spring masses and damper systems were the spring mass and damper system of the MacPherson Strut suspension in the car, and of the car seat and the driver. The parameters of the MacPherson Strut suspension were derived using the physical measurements taken from the digital vehicle (discussed in chapter 3) and Equations 7.1, 7.2 and 7.3:

\[
F = \frac{\pi \cdot d^3 \cdot r}{16 \cdot r} \quad (7.1)
\]

\[
f = \frac{64 \cdot \pi \cdot r^3 \cdot F}{d^4 \cdot G} \quad (7.2)
\]

\[
K = \frac{F}{f} \quad (7.3)
\]
Where, \(d\) is a spring coil diameter (14 mm), \(r\) is a spring radius (60 mm), \(n\) is number of coils (5) and \(G\) is a shear modulus of the spring - \(79.3 \times 10^9\) Pa. The above \textit{Equations 7.1 to 7.3} were used to calculate the spring constant of the MacPherson Strut suspension, which was 44070N/m. Next, the damping ratio of the suspension damper was calculated using \textit{Equation 7.4} given below, based on the value of the spring constant and the mass of the ICE vehicle.

\[
C_s = \frac{\sqrt{(K_s M_s)}}{\sqrt{2}}
\]

\textit{Equation 7.4}

Where, \(K_s\) is a spring constant for the MacPherson Strut and \(M_s\) is a sprung mass of the ICE vehicle. The damping coefficient was calculated as 2969N/m and the spring coefficient and damping coefficient for the seat were calculated as 176280N/m and 8372Ns/m. The road excitation for a passive suspension was considered using a road bump in the form of transient force. These conditions were simulated by non-periodic, deterministic excitation, or random excitation, with unpredictable time described using probability (Savaresi and Spelta 2009). A road bump in the form of a high-risk pothole was considered as a source of excitation for the quarter-car model (Cao, Song et al. 2011), using the depth of 100mm, leading to \(X_e\) equal to 1. The key parameters used to perform the quarter-car analysis are given in \textit{Table 7.2} below:

\textit{Table 7.2: ICE and EV specifications considered for quarter car modelling}

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values (ICE)</th>
<th>Values (EV)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driver-seat mass ((M_{ds}))</td>
<td>90kg</td>
<td>90kg</td>
</tr>
<tr>
<td>Sprung mass ((M_s) and (M_{se}))</td>
<td>110kg</td>
<td>130kg</td>
</tr>
<tr>
<td>Unsprung mass ((M_u) and (M_{ue}))</td>
<td>40kg</td>
<td>80kg</td>
</tr>
<tr>
<td>Damping coefficient suspension ((C_s))</td>
<td>2969Ns/m</td>
<td>2969Ns/m</td>
</tr>
<tr>
<td>Damping coefficient seat ((C_{ds}))</td>
<td>8372Ns/m</td>
<td>8372Ns/m</td>
</tr>
<tr>
<td>Spring constant tyre ((K_t))</td>
<td>125000N/m</td>
<td>125000N/m</td>
</tr>
<tr>
<td>Spring constant suspension ((K_s))</td>
<td>44070N/m</td>
<td>44071N/m</td>
</tr>
<tr>
<td>Spring constant seat ((K_{ds}))</td>
<td>176280N/m</td>
<td>176280N/m</td>
</tr>
</tbody>
</table>
The quarter-car model equations are based on the model depicted in Figure 7.1. The block diagram in Figure 7.2 shows the various forces acting on the different masses on the vehicle suspension. Equations of motion are established for the three different masses, namely sprung mass ($M_s$), unsprung mass ($M_u$) and the driver-seat mass ($M_{ds}$), whereby the latter includes the various forces acting on the driver-seat mass.

$$M_{ds} \ddot{x}_{ds} + K_s (X_{ds} - X_s) + C_{ds} (\dot{X}_{ds} - \dot{X}_s) = 0$$  \hspace{1cm} (7.5)

Equation 7.5 represents the various forces acting on the mass in an equilibrium position. This enables the calculation of driver-seat mass acceleration, as given by Equation 7.6.

$$\ddot{x}_{ds} = \frac{-K_s (X_{ds} - X_s)}{M_{ds}} - \frac{C_{ds} (\dot{X}_{ds} - \dot{X}_s)}{M_{ds}}$$  \hspace{1cm} (7.6)

The motion of the sprung mass of the system in the position of equilibrium is given by Equation 7.7:

$$M_s \ddot{x}_s - K_s (X_{ds} - X_s) - C_{ds} (\dot{X}_{ds} - \dot{X}_s) + K_s (X_s - X_u) + C_s (\dot{X}_s - \dot{X}_u) = 0$$  \hspace{1cm} (7.7)

As above, the acceleration of the sprung mass was obtained from Equation 7.7, and is given below as Equation 7.8:

$$\ddot{x}_s = \frac{K_s (X_{ds} - X_s)}{M_s} + \frac{C_{ds} (\dot{X}_{ds} - \dot{X}_s)}{M_s} - \frac{K_s (X_s - X_u)}{M_s} - \frac{C_s (\dot{X}_s - \dot{X}_u)}{M_s}$$  \hspace{1cm} (7.8)
Similarly, equation of motion in equilibrium position was written for unsprung mass as

\[ M_u \ddot{x}_u - K_s (x_s - x_u) - C_s (\dot{x}_s - \dot{x}_u) + K_t (x_u - x_r) = 0 \quad (7.9) \]

Once again, acceleration of unsprung mass was obtained from Equation 7.9, and is given below as Equation 7.10:

\[ \ddot{x}_u = K_s \frac{(x_s - x_u)}{M_u} + C_s \frac{(\dot{x}_s - \dot{x}_u)}{M_u} - K_t \frac{(x_u - x_r)}{M_u} \quad (7.10) \]

The three equations of second order (Equation 7.6, 7.8 and 7.10) were used as input into the system. Next, a state space equation was established using equations of motion, for which six different state space variables were defined, representing the displacement and velocity of the sprung, unsprung and driver-seat mass of the quarter-car suspension system, respectively. Hence, six state space variables were derived from Equation 7.6, 7.8 and 7.10. These were represented by Equation 7.11 to Equation 7.16 below:

\[ x_1 = \dot{x}_{ds} \quad (7.11) \]
\[ x_2 = x_s - x_{ds}; \quad (7.12) \]
\[ x_3 = \dot{x}_s; \quad (7.13) \]
\[ x_4 = x_u - x_s; \quad (7.14) \]
\[ x_5 = \dot{x}_u; \quad (7.15) \]
\[ x_6 = x_r - x_u \quad (7.16) \]

The state space equation was established using the state space variables given by Equation 7.11 to 7.16 and was written as a column matrix \( \mathbf{x}(t) \), \( n \times 1 \), which represented the current state of the system, and matrix \( \mathbf{u}(t) \), which represented the system input vector. When the current state and the system input were available in the state space equation, the system output \( \mathbf{y} \) was obtained from the state space equation. Here, the state space variables were differentiated with respect to time and formed into matrices. Equation 7.17 shows the state space equation for the quarter-car suspension system. This state space equation, which formed a mathematical model from a set of inputs and
state space variable, was written on the basis of the state space variables, as mentioned above.

\[
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2 \\
\dot{x}_3 \\
\dot{x}_4 \\
\dot{x}_5 \\
\dot{x}_6 
\end{bmatrix} =
\begin{bmatrix}
-K_{ds}/M_s & -C_{ds}/M_s & K_{ds}/M_s & -C_{ds}/M_s & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
K_{ds}/M_s & -C_{ds}/M_s & \frac{K_{ds}}{M_s} & -\frac{K_{ds}}{M_s} & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 
\end{bmatrix}
\begin{bmatrix}
x_1 \\
x_2 \\
x_3 \\
x_4 \\
x_5 \\
x_6 
\end{bmatrix} + u 
\] (7.17)

The above state space \textit{Equation 7.17} was used to derive the output equation for displacement and velocity, given below in the matrix form using \textit{Equation 7.18} below:

\[
y = \begin{bmatrix}
X_{ds} \\
X_s \\
X_u \\
X_{ds} \\
X_s \\
X_u 
\end{bmatrix} = \begin{bmatrix}
1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 
\end{bmatrix}
\begin{bmatrix}
x_1 \\
x_2 \\
x_3 \\
x_4 \\
x_5 \\
x_6 
\end{bmatrix} + u 
\] (7.18)

Clearly, \(y\) on the left side of equality in \textit{Equation 7.18} represented the system output.

The state space equation given above was later solved using MATLAB® to obtain the system output in form of displacement and velocity for suspension analysis, which are discussed in the next section.

\textbf{7.3 Suspension ride comfort analysis}

Quarter-car model equations derived in the preceding section were used as inputs to develop the suspension analysis program. As noted above, \textit{Equation 7.18} was used to derive the output equation for displacement and velocity, whereby MATLAB® was employed for analytical experimentation on the suspension system. The results obtained from MATLAB® and Simscape® were analysed to compare the variation in the performance of the suspension systems of the ICE and EV vehicles. The analysis performed on both ICE and EV examined the changes in the vehicle performance during driving. The results obtained from Simscape® and MATLAB® analytical analyses were similar, indicating that the suspension systems of the two vehicles performed similarly. As the response of the sprung mass was critical for performance evaluation of the EV, it was assessed, and the result plotted. The graphs below compare the velocity and displacement changes of sprung mass when subjected to a step input.

Using MATLAB® simulations, two programs were run, as given in 

\textit{Appendix 14} and \textit{15}. 

187
7.3.1 Sprung mass

It can be observed from *Figures 7.3 and 7.4* that the velocity of the sprung mass changed by more than 16% for the EV (from 1.5 seconds in an ICE to 1.8 seconds in an EV), even though the velocity fluctuation was not significant, but rather persistent. The 21% more displacement was observed in the EV compared to ICE ((from 1.1 seconds in an ICE to 1.4 seconds in an EV). The velocity and displacement stabilised over a shorter period of time for the ICE vehicle, as compared to the EV, implying that any input to the vehicle suspension over a longer period of time would affect the driving comfort. In other words, the higher frequency of oscillations in the EV had a direct impact on the ride comfort and handling of the vehicle. The main reason for this behaviour was the increase in the sprung mass, which meant that vehicle experienced smaller change in velocity for the same step input due to increased overall weight affecting the spring and dampers. This resulted in longer oscillation cycles. Consequently, although the suspension damper was able to handle the increased load of the EV, the damper efficiency was still compromised. In the experiments described above, the damping coefficient of the suspension damper was 2969Ns/m for an EV. The vehicle handling was also slightly affected due to this observed behaviour, especially during cornering and other sharp manoeuvres.

*Figure 7.3: Velocity and displacement of sprung mass (ICE)*
7.3.2 Unsprung mass

*Figures* 7.5 and 7.6 depict the difference in performance of the unsprung mass in the ICE and EV models, respectively. The velocity fluctuation of the unsprung mass for the EV was greater than 20% (from 1.2 seconds in an ICE to 1.5 seconds in an EV). Therefore displacement variations were 25% for unsprung mass of the two compared vehicles (from 0.6 seconds in an ICE to 0.8 seconds in an EV). These results suggest that this behaviour affects the ride comfort for an EV, since instability was observed due to the increase in the unsprung mass. The ride quality was enhanced due to decreased velocity changes for the same amplitude of displacement. The time required for the EV to stabilise to its normal position was longer than that of the ICE vehicle. In the EV, the damper in the vehicle was able to handle the extra load due to unsprung mass, which led to slight velocity fluctuations over a longer time period. Finally, the impacts of these loads were tested for driver-seat mass. The driver-seat of the vehicle was the most critical part of the quarter-car model analysis, since it had direct effect on the driver and passengers travelling in the vehicle.
Figure 7.5: Velocity and displacement of unsprung mass (ICE)

Figure 7.6: Velocity and displacement of unsprung mass (EV)

7.3.3 Driver-seat mass

In Figures 7.7 and 7.8, it can be observed that the response of the seat and driver to the step input was similar to the other two simulation results, suggesting direct impact on both the driver and the passengers in the vehicle. The velocity fluctuation of an EV was more than 16% of an ICE (from 1.5 seconds in an ICE to 1.8 seconds in an EV). The
displacement fluctuation was 20% greater in an EV than to an ICE ((from 1.0 seconds in an ICE, to 1.25 seconds in an EV). This change in velocity and displacement affected the drivers comfort range, with extra oscillations as observed in an EV.

Figure 7.7: Velocity and displacement of driver-seat (ICE)

Figure 7.8: Velocity and displacement of driver-seat (EV)

This had a direct effect on the damper handling due to the increase in both unsprung and sprung mass. The increased oscillations led to an uncomfortable ride, thus reducing the
quality of the experience for the passengers. In other words, although the increased spring stiffness of the seat and driver mass minimised the problem, it led to reduced ride quality. It can be concluded that the increase in damping coefficient benefitted handling and drive conditions of the vehicle. In an EV with the in-wheel SRM, the ride quality was also slightly affected. However, the simulation results indicated absence of large displacements or shocks during driving, when using in-wheel SRMs. Hence, this arrangement was concluded safe based on further Bodeplot analysis explained in section 7.3.5.

### 7.3.4 Simscape® analysis

Empirical analysis of the quarter-car model was performed using Simscape® tool (detailed model and results are given in Appendix 16, Figures 16.1) based on a block diagram technique, with logical sequencing of inputs, sensors, actions and the outputs, as per analysis requirements. The quarter-car models for both EV and ICE were created using free body diagram. Each mass included in the system was considered as a separate sub-block with its own damping coefficient and spring constant, both of which were treated as a separate unit within the relevant sub-block. As a result, three sub-blocks were formed representing the three mass systems, as shown in the free body diagrams. They were connected to sensors that map the output from these three different units identified as blocks. These outputs were plotted in order to observe the change in velocity and displacements of suspension systems for both EV and ICE. The results from the Simscape® were plotted and examples of EV are shown in Appendix 16 Figures 16.2, 16.3 and 16.4.

The variations in displacement and velocity of sprung mass in Simscape® analysis were similar to those indicated by the results of the MATLAB® analysis. An example of EV Simscape® analysis for sprung mass is shown in Appendix 16 Figure 16.2. The change in velocity was drastically reduced for the EV due to the increased weight of the vehicle. Although the displacement was also reduced, the proportional change was smaller in magnitude. The frequency of vibration was increased in the EV model, making the ride slightly uncomfortable, when compared to the ICE model. Insufficient damping coefficient due to the increased mass was the main reason for the observed reduction in the EV ride quality.
A very similar behaviour was observed for the unsprung mass of the EV suspension system, as shown in Appendix 16 Figure 16.3. The increase in frequency was more prominent in the unsprung mass, due to the addition of motor inside the wheel, which resulted in slight deterioration in the ride quality. Appendix 16 Figure 16.4 provides graphical representation of the direct impact of this change in behaviour on the driver-seat mass in an EV. As can be seen, the ride quality was slightly affected for both the passengers and the driver of the EV, due to increased unsprung mass inside the wheel.

7.3.5 Bodeplot analysis

The slight depreciation in the ride quality was confirmed by the analysis using MATLAB® and Simscape®; however, as the changes were of the order of 20 to 25%, the recommended vehicle safety was in question. This analysis only identified the variations in the ride quality; however safety of the vehicle suspension in an EV was in question. The increase in unsprung mass due to motor affected the EV ride quality; since safe performance was determined using Bode plots. To further investigate these effects on the EV with an increased mass, additional analysis on unsprung mass was carried out using the Bode plots, based on the natural system frequency, as this is a good indicator of the actual ride conditions. The natural system frequency is defined as the frequency at which a driving force causes maximum oscillation amplitude or even leads to unbounded oscillations. Bode plot analyses of the EV and ICE models were carried out to observe the changes in frequency in different parts of the suspension system. The plots showed the variation in both magnitude and phase with frequency of a liner time invariant system. The suspension system analysis was based on a time invariant system, as all the physical parameters of the system remained constant over time. The Bode plot input was a transfer function, which was the ratio of the function input and output, converted in to Bode plot by performing MATLAB® analysis. The plot in Figure 7.9 was obtained from the MATLAB® analysis.
The vehicle was subjected to the least amount of fluctuation during a normal ride, as shown in Figure 7.9. However, the applied step input brought about the changes in magnitude and phase at varying system frequency levels. The most acute change was observed in the both the systems. As is evident in Figure 7.9, Bode plot analysis showed promising results with no negative effect observed for the EV with an in-wheel SRM. Generally, a system is considered unstable if it has a phase of -180 degrees at its crossover frequency, which was, in this case, less than 1.0Hz or above 2Hz. In addition, the system was potentially stable as the magnitude was smaller than -180 phase at 1-1.5Hz and showed less than 10dB. However, the graph depicted in Figure 7.9 indicates that the EV was stable for frequencies ranging from 1 to 1.5Hz. Overall, the frequency analysis indicated positive results for the EV, in comparison with the ICE. The phase in -180 at cross over frequency range has no crossover for both vehicles between 1-1.5Hz, as noted above and shown in Figure 7.9. The small change for the EV only occurred in the 1-1.5Hz range with its cross over degrees due to its substantial increase in unsprung mass. Nonetheless, the spring and the damper of the vehicle had sufficient capacity to absorb the shocks due to the increased mass, resulting in a comfortable ride. The ride

**Figure 7.9:** Frequency response of ICE and EV vehicles using Bode Plot
quality was within a comfortable range and it was not in the harsh ride range of 2-5Hz. Above 5Hz, the deterioration in ride quality is much greater, and 5-7Hz causes abdominal injuries. Anything above this range causes critical spinal injuries (7-18Hz), as well as damage to passenger’s head and neck (anything >18Hz). A very slow frequency response is also known to cause motion sickness, however this was not observed in the EV. However, these analyses confirmed that the ride in the EV was slightly harsher than in the ICE; however, it was still within the recommended comfortable range.

7.4 Fatigue analysis

Fatigue is permanent damage that occurs when a material is subjected to cyclic loads during driving. In reality, the fatigue testing of the actual vehicle is expensive and time consuming. As a result small test cycles are used in a laboratory environment to virtually simulate the performance of vehicle suspensions. With virtual fatigue study the numbers of physical tests are minimised. Hence, in this study, it was essential to validate the durability and performance of suspension components with an increased mass at wheels.

Passenger safety is the primary concern of automotive design evaluation, typically conducted using fatigue analysis techniques. Fatigue typically leads to cracks in the vehicle components, whereby crack propagation results in the failure of the entire vehicle system. In this study, as the suspension was subjected to large load cycles, it was important to analyse the effects of increased mass due to the in-wheel SRM on suspension design. The initiation of cracks within a part is typically the reason for structural failures within the components. In other words failure of parts subjected to fatigue loads is mainly result of rupture or cleavage. The Society of Automotive Engineers (SAE, 1997) lists seven fundamental causes of cracking, namely manufacturing defects, poor choice of material or heat treatment, poor choice of production technique, poor design, unanticipated service environment, poor material property data and material defects. In this research, the design change due to in-wheel SRM had substantial effects on suspension system; hence, it was essential to investigate their magnitude and mitigate any resulting issues.
In the past, significant improvements in safety were achieved by analysing the fatigue life during design stages (Buciumeanu, Miranda et al. 2007). Fatigue study of suspensions and related components provides a good estimate of the life cycles (2000). Also from the literature it is evident that FE methods (Ansys® test bench 13.1) were used to identify the maximum Von mises stress concentrations for the suspension components. The findings suggest that plastic deformation caused by a combination of bending and torsion stresses caused fatigue failures in the components (Bayrakceken, Tasgetiren et al. 2006). Most of the suspension parts in a passenger car (e.g., ball joints, lugs, torsions bars, springs) were previously analysed using a combination of multi-body system simulation, fatigue analysis and shape optimisation (Haussler and Albers 2007, He, Wang et al. 2010, Ossa, Palacio et al. 2011). In this section, FE methods were used on the life cycle performance of high stress concentration area of a suspension, with the aim to evaluate the long-term durability and the safety adoption of an in-wheel SRM. An empirical method using Ansys® test bench 13.1 was used in this research to perform the fatigue life analysis of the suspension components. The main objectives of fatigue analyses were:

- Developing high stress area component within suspension using FE methods by modelling an appropriate service loads to mimic the physical conditions
- Development of variable amplitude loading event curve by conducting experiments on the suspension from an EV (ICE converted)
- Using the load curve data, Gerber method and S-N curve data of the material for the fatigue life cycle analysis
- Using a rain flow counting cycle method to generate damage matrix and life matrix for the suspension component. Using Plamgren-miner rule, predicting the fatigue life cycles based identified cumulative damages to suspension component

**7.4.1 Static study**

The rain flow counting algorithm with the Plamgren-miner rule facilitates correct prediction of suspension life cycle. The static study was not required for a fatigue analysis using rain flow count method chosen. However, it was important to obtain high Von-mises stress concentration areas within FE models to conduct a fatigue study.
Hence an initial static study was devised which was further carried to analyse fatigue life of the high stress concentration area within the suspension model.

7.4.1.1 Finite element modelling

The FE models were developed for the suspension in the Ansys® test bench 13.1, commercial FE software. The models were simplified to mesh the assembly for in-wheel SRM suspensions. The suspension was further simplified to remove beam element meshing, since axial and bending stresses were not required for this specific analysis. A suspension of an in-wheel SRM is shown in Figure 7.10, which was simplified to Figure 7.11, where mixed modelling techniques were used to recreate the suspension FE model. 3D meshing techniques were used, whereby local meshing was achieved using required element sizes in the small areas, such as fillets and chamfers. The model of the part consisted of system nodes, linked to form a mesh of tetrahedral elements, whereby each was assigned the properties of the material and subsequently analysed individually. The individual elements were then combined in order to assess the performance of the entire body of the component when subjected to the loading conditions set in the previous step. The contacts and connections, such as pins, springs bolts, fixed hinge and joints, were defined to complete the model prior to the analysis.

![Figure 7.10: Suspension model of an in-wheel SRM](image)

Initially, suspension was modelled using the local and the global 3D tetrahedral solid mesh with 4 nodes; an example is shown in Figure 7.11. The suspension FE model consisted of 116,107 elements and 187,864 nodes. The local mesh was modelled at: i) fillet/spline areas/corners of MacPherson strut (enlarged view- A in Figure 7.11) and ii)
near eye bolt holes at both lugs (enlarged view- B in Figure 7.11), with a tetrahedral finer mesh of 1 to 4mm. The global tetrahedral mesh of 10mm or more was modelled into the rest of the suspension with a smooth transition ratio of 0.288. This transition ratio allowed a smooth flow from the local to the global mesh within the FE model.

**Figure 7.11: Suspension FE model with load and boundary conditions**

### 7.4.1.2 Material

The following material data were used for: i) High strength low alloy steel ASTM A1011 A36grade (ASTM International 2012), chassis plate, body base plate and lugs, and ii) High tensile chromium steel A519 grade (ASTM International 2012) for MacPherson strut plunger and sleeve. The engineering data of ASTM A1011 A36 material used are i) Young’s modulus (\(E\)) 200GPa, ii) Poisson’s ratio (\(\nu\)) 0.30, iii) density (\(\rho\)) 7.83g/cc and iv) ultimate tensile strength (\(S_{ul}\)) 620MPa. A material with linear fatigue characteristics, corresponding to those of A1011 A36grade steel, was defined for fatigue analysis of suspension design, as shown in *Figure 7.12*. 

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*Figure 7.12: Linear fatigue characteristic*
The engineering material data of A519 material used are i) Young’s modulus \((E)\) 210GPa, ii) Poisson’s ratio \((\nu)\) 0.33, iii) density \((\rho)\) 7.86g/cc and iv) ultimate tensile strength \((S_{ut})\) 621MPa.

### 7.4.1.3 Load and boundary conditions

As shown in Figure 7.11, all the contacts and connections for the FE model were developed. The no penetration contacts were made between lug and: i) body base plate, and ii) chassis plate. Pin connections were defined between: i) MacPherson strut eye hole and lugs on both sides, and ii) MacPherson strut pin and cylinder. Spring connection was modelled between MacPherson strut cylinder face and end cap face. Fixed hinge support was modelled between MacPherson strut eye hole and top lug hole. Finally chassis plate back face was fixed in the FE model. Vertical load of 5kN (as discussed in earlier chapter) and horizontal load of 2kN (based on actual load determined using strain gauge) was modelled on body base plate face as shown in Figure 7.11.

### 7.4.1.4 Finite element results

These analyses were nonlinear because of the large displacements due to loading conditions and multi-body parts. Static nonlinear analysis simulated the maximum Von-mises stress and displacements under these loading conditions. Nonlinear study, as shown in Figure 7.13, revealed increased levels of Von-mises stress concentrations on
the lug, which was subjected to 349Mpa, and considering yield strength, had a factor of safety greater than 2. Overall displacements of 0.16mm were observed as shown in Figure 7.14. The lug was moving in both directions as shown in Figure 7.14.

*Figure 7.13: High Von-mises stress concentration area in the EV suspension (Mpa)*

*Figure 7.14: Maximum displacement of lug in the EV suspension (scaled 100 times for clarity)*
7.4.2 Fatigue study

Further fatigue study was conducted in order to investigate the lug performance at high Von-mises stress concentration area using variable amplitude loading obtained from the field testing.

7.4.2.1 Variable amplitude loads

Generally, two types of loading conditions were used for fatigue life in FE methods—constant amplitude and variable amplitude loading to simulate the physical conditions. Constant amplitude loading is a crude method that provides sinusoidal loads, where alternating stresses are exaggerated to predict the corresponding stresses in the structure. Variable amplitude loading is a more accurate method of simulating fatigue, as it provides life cycle over an entire stress cycle exerted on the suspension structure. The output of this analysis is generally compared to three curves showing material properties—Gerber, Sodberg and Goodman curves. Gerber curve was used in this research as it is appropriate for ductile materials and therefore provided correct representation of the actual system using parabola curve fitting. Although different methods can be used to determine cycles required for fatigue techniques, peak counting method was used in this research, which identified the occurrence of relative peak loads. Next, damage and rain flow matrices were used to explain the results and predict the system life cycle using fatigue techniques.

After a realistic load was applied, the analysis was conducted using realistic fatigue design procedures that modelled the real world scenarios. The stress-based approach was implemented using FE methods to predict the fatigue life of a part that was subjected to the fatigue loads obtained in the field tests. In most scenarios, the amount of force applied on the suspension was limited to the amount of force the entire vehicle body was subjected to. Since suspension is a complex part, the loading scenario was not simple. The strains of the assembly used in the analysis generated torque rather than stress, which was translated into a torsional shear stress using torsion Equation 7.19 below:

\[ \tau_{max} = \frac{\tau \cdot r}{T} \]  

(7.19)
Where $T$ is the torque applied, $r$ is the radius, $D$ is the diameter and $J$ is polar moment of inertia given by Equation 7.20:

$$J = \frac{\pi D^4}{32} \quad (7.20)$$

A classical stress-based fatigue design method was an accurate representation of the cyclic loads to which the suspension was subjected during the experimental drive cycles. The above scenarios were simulated by applying specific combinations of loads, and recording the data of the strain gauge to represent the changing load magnitude applied on the suspension under the corresponding vehicle travel conditions. A stress versus time curve (S–N curve or $\sigma$–N curve), shown in Figure 7.15, was plotted based on the results of the field tests performed using strain gauges over a period of time.

![Load history curve](image)

**Figure 7.15: Load history curve**

The small car was tested under different road riding conditions, with increased mass added to the wheel, reproducing the effects of in-wheel SRM unsprung mass. The curve was used as input for fatigue analysis to examine the safety of suspension components in the context of the EV. Each cycle was extracted and characterized by its $\sigma_{\text{max}}$, $\sigma_{\text{min}}$, $\sigma_{\text{alt}}$ and $\sigma_{\text{mean}}$, the graph was generated. When put on the log scale, the relationship of stress $\sigma$ and cycles $N$ curve are expressed using a Basquin slope $b$ by Equation 7.21:

$$N = N_0 \left(\frac{\sigma}{\sigma_0}\right)^b \quad (7.21)$$
Where \(N_0\) and \(\sigma_0\) were coordinates at any point for Basquin slope \(b\). In equation \(\sigma_0\) is \(\sigma_{\text{end}}\) then the \(N_0\) is considered as \(10^6\) cycles. Subsequently, among various mean stress correction algorithms, the Gerber correction factor was used in this study. The Gerber mean stress correction factor \((G_m)\) is given by Equation 7.22:

\[
G_m = \left(\frac{\sigma_{\text{alt}}}{\sigma_{\text{end}}}\right) + \left(\frac{\sigma_{\text{alt}}}{\sigma_{\text{ult}}}\right)^2
\]  

(7.22)

Therefore, the total damage was calculated as the sum of all the partial damage factors. The start time for every single variable amplitude event was identified and the variable loading curve was entered. Firstly, the input was amplitude only, where the \(X\) column represented time and the \(Y\) column was loading amplitude. The specified recorded data of the sampling rate corresponded to the intervals. Here the start time for each event was specified. The time and amplitude were input, thus \(X\) and \(Y\) columns represented the time and the loading amplitude in the relevant study.

### 7.4.2.2 Rain flow counting method

The fatigue study was based on the rain flow counting, whereby the algorithm divided the stress amplitude \(Y\) axis so that it was equally spaced, with the constant amplitude magnitude within each bin. In the rain flow method “quick counting” was available for non-constant amplitude loading substantially reducing runtime and memory. In quick counting, alternating \((\sigma_{\text{alt}})\) and mean stresses \((\sigma_{\text{mean}})\) were sorted into bins before partial damage was calculated. When quick counting is not applied, the data is not sorted into bins until after partial damages are identified. The accuracy of quick counting was usually very good if a proper number of bins were used when counting. When using the rain flow counting bins, the accuracy of the fatigue results is dependent on their number. In this study, 32 bins were used, as this was deemed sufficient for obtaining accurate loading results. If a higher number of bins was chosen, although the result would be more accurate, the solving time would also increase, which was undesirable. In the results obtained, stress peaks smaller than the endurance limits were filtered out, since their contribution to the overall damage results in the random loading history was negligible. Under the controlled test conditions and in the specified environment, using the generated S-N curves, the material resistance to fatigue was characterised. However, the environment in which the analysed product typically operated greatly differed from the conditions of the test undertaken. Consequently, a strength reduction factor was
introduced to account for the environment and other important phenomena that influenced fatigue life. The compound effect of all of the influences was described by the fatigue strength reduction factor, as per Equation 7.23:

\[ K_f = K_c \times K_m \times K_{freq} \times K_l \times K_r \times K_n \times K_{fret} \]  

(7.23)

The different effects considered in the fatigue design were generally compensated by the fatigue reduction factor, which comprised effects of corrosion \( K_c \), loading mode reduction factor \( K_m \), frequency reduction factor \( K_{freq} \), size reduction factor \( K_l \), temperature reduction factor \( K_t \), reliability reduction factor \( K_r \), notch effects reduction factor \( K_n \) and fretting reduction factor \( K_{fret} \). The study used a fatigue reduction factor \( K_f \) of 0.5, calculated using Equation 7.23 for the material.

### 7.4.2.3 Finite element results

The analysis was performed for the in-wheel SRM suspension assembly of an EV model in order to determine its durability when subjected to a variable amplitude fatigue. The analysis was performed on the lug, where high Von-mises stress area was observed in the earlier nonlinear study, as shown in Figure 7.13. The results were used to evaluate the failures in suspension, when the load factors were changed due to the additional unsprung mass of the in-wheel SRM.

Finally, after the analysis was complete, a 3D rain flow chart was plotted, as shown in Figure 7.16, with the \( X \) (horizontal) and \( Y \) (traverse) axes respectively representing the alternating stress (stress range) and mean stress, and the number of counts for a given alternating and mean stress bins given on the \( Z \) (vertical) axis. The results obtained thus provide the measure of composition of the loading history and worst case scenario. It was observed from Figure 7.16, that most of the alternating stresses had a positive mean value. It was also observed that number of cycles decreased with increasing alternating stress values. This implied that a small number of cycles were responsible for higher damage within the load history.
A rain flow cycle counting method was used to identify stress reversals and the damage summation was performed by applying i ner’s rule. In Palmgren-Miner rule, $N_i$ referred to fatigue lifetimes for all occurring cycles of type $i$ (for all elements of the rain flow matrix), where $n_i$ was the number of cycles of type $i$. Thus, failure or percentage damage $D$ is described by Equation 7.24:

$$D = \sum_i \frac{n_i}{N_i} \leq 1$$

(7.24)

**Figure 7.16:** 3D rain flow matrix for cycles of an in-wheel EV suspension

**Figure 7.17:** Damage plot for an in-wheel EV suspension
In Figure 7.17 the maximum percentage damage was 0.00024 for 244 cycles, based on the load curve, which was analysed using rain flow method with 32 bins. From Figures 7.16 and 7.17 maximum alternative stresses were observed as 150MPa and endurance limit of the material is 250MPa. Hence using Equations 7.12, 7.22, and 7.24, applying Palmgren-Miner rule the total life cycle of the suspension system is $9.9 \times 10^7$ cycles were calculated. Each cycle was 2500 seconds based on S-N curve modelled initially. Assuming average driving of 8 hours a day, the life of suspension was calculated as nearly 12 years. Hence this analysis provided the fatigue life of the suspension system in fully modified EV, corresponding to the changes in load scenarios. Thus, the suspension design was proven safe when the unsprung mass was increased for the in-wheel SRM.

### 7.5 Summary

The increased unsprung mass due to in-wheel SRM for the EV affected its ride conditions and comfort. In this chapter, quarter-car model, and fatigue studies were conducted, whereby the performance and safety of the suspension in an EV using in-wheel SRM were evaluated. The key conclusions that can be derived from the reported results are summarised below:

- Based on the block diagram, a free body diagram was constructed for modelling suspension system subsequently used for mathematical analysis. Three scenarios were compared, namely (1) sprung, (2) unsprung, and (3) driver-seat step loads derived from the free body diagram, and were verified using mathematical models.

- Both analytical and empirical analyses were conducted using MATLAB® and Simscape® for suspension of an in-wheel SRM and compared with the ICE car. Using these mathematical models, analytical study was reconstructed for suspensions in EV and ICE models using MATLAB®. Similarly, using block diagram methods of Simscape®, suspension analysis was performed for EV and ICE models and the findings compared and discussed. The results for sprung, unsprung and driver-seat locations for EV and ICE were similar in both methods.

- A key observation from the above analyses was slight variation in ride comfort of the EV model, when compared to that exhibited by the ICE. This was due to the variation in the velocity plots, whereby those for the EV exceeded those of the ICE by: i) 16% in displacements and 21% in velocity amplitude at sprung mass, ii) 20%
in displacements and 25% in velocity amplitude at unsprung mass, and iii) 16% in displacements and 20% in velocity amplitude at driver-seat mass. This large variation at unsprung mass was due to increased mass of the vehicle due to in-wheel SRM at the wheels.

- Following the mathematical analysis, further study was performed in order to determine the safety of the EV suspension in comparison to that of the ICE. Bode plot analysis techniques were used to determine the ride comfort range for the developed EV. Generally, a system is considered unstable if it has a phase of -180 degrees at its crossover frequency, which should be below 1-1.5Hz. A system is also potentially unstable if the magnitude exceeds 10dB when the phase is at its crossover frequency of 1-1.5Hz. In the experiments performed here, it was seen that both EV and ICE performed in a frequency range of 1 to 1.5Hz. Thus, suspension design was safe for the developed design.

- The fatigue analysis was conducted next, as it was essential to evaluate the safety of the suspension component, subject to maximum Von-mises stress and maximum displacements due to the increased unsprung mass. The FE model was developed and a nonlinear analysis was conducted. This study indicated that the lug at the bottom of the suspension underwent maximum Von-mises stress concentration of 349Mpa and maximum displacement of 0.16mm. Hence the lug high Von-mises stress area was identified as worst case scenarios for further fatigue study.

- The variable amplitude load curve was constructed using field tests and strain gauge at specific time intervals. The number of cycles obtained for each stress range was used in the cumulative damage theory to achieve an estimate of the structural fatigue life. The technique was based on variable amplitude loading on the vehicle, as it enabled safety evaluation at a location identified as a hot spot. The fatigue study result was located on a lug at higher Von-mises stress concentration area from an earlier static study. The damage plot at the location indicated that 0.00024% was the maximum, achieved for life cycle using 244 blocks. The number of cycles obtained for each stress ranges was further used in the cumulative damage theory to achieve an estimate of the structural fatigue life.

- Finally, a 3D rain flow matrix was developed. The study was performed in order to obtain damage and life plots for the EV suspension with an in-wheel SRM. Palmgren-Miner rule application was performed in time domains to estimate the
structural fatigue life. The life plot indicated that most of the alternating stresses had a positive mean value. It was also observed that number of cycles decreased with increases in alternating stress values. This implied that a small number of cycles was responsible for higher damage.

- Applying the Palmgren-Miner rule the total life cycle of the suspension system was calculated as $9.9 \times 10^7$ cycles. Each cycle was 2500 seconds based on the S-N curve modelled initially. Assuming average driving of 8 hours a day, the life of suspension was calculated as nearly 12 years. Based on these results, it was concluded that suspension system of the Holden Barina Spark with the motor fitted inside the wheel was completely safe.
Chapter 8
Vehicle performance modelling

8.1 Chapter overview
The vehicle fitment study was conducted using VR tools for the adoption of the novel drivetrain conceived in this research. This chapter demonstrates the following objectives:

- A vehicle fitment strategy was investigated using VR tools to fit the drivetrain and EV ancillaries, such as the battery package, management system, super controller and motor controller. Parts such as the engine, radiator, gear box, air box and battery were removed. These studies examined the clearance and space for fitment of the EV ancillaries and the drivetrain.

- The weight distribution of the vehicle was developed for the EV. The weight distribution was examined for lateral, longitudinal and height variations in the CG of the EV than the ICE. The vehicle rolling condition was determined by a longitudinal slip variation with the force in the tyre. This was established by modelling variations of longitudinal slips for an EV and an ICE using Pacejka’s magic formula.

- In this developed drivetrain wheel is an integral part of motor. As consequence wheel size is bigger and weight is more. As a result while servicing wheel, the operator safety and easy detachments were critical aspects. Hence, using VR/AR models, easy and safer wheel detachment was established.

- Based on EPA standards to develop a drive range for the developed the EV. The range of the vehicle was demonstrated for urban and rural drive conditions using 36 and 72 battery cells within the vehicle. Finally an EV specification sheet was developed.

8.2 Vehicle fitment

8.2.1 Front bay fitment
For this section, a small car was conceptually evaluated using VR tools for the deployment of an in-wheel SRM. The addition of a battery package, management system, super controller, motor controllers and in-wheel SRMs were required, based on functionality and the space available. Using a small car, the fitments of motor and other
EV ancillaries within the vehicle were examined for clearances and space availability. The initial digitised Holden Barina Spark model (discussed in chapter 3) was used and the items to be added and removed for electrifying were examined for their dimensional fitment. The engine, radiator, gearbox, air box and spare battery were removed from the existing engine bay of the ICE model as shown in Figure 8.1. The coolant system was left as it is in the front bay. This provided usable dimensional space in the front engine bay of 0.8 metre in a lateral direction, 0.6 metre in a longitudinal direction and 0.65 metre in a vertical direction. This provided an overall space of 0.31 m$^3$ from the total space of 0.47 m$^3$ in the front engine bay.

![Figure 8.1: Front bay of the Holden Barina Spark ICE](image)

In 0.31 m$^3$ space, 18 main battery packs, 1 auxiliary battery pack, 3 power supply boxes for the auxiliary pack and a battery management system (BMS) were placed. Battery packaging included the main high voltage battery packs, an auxiliary battery pack, a power box, a management system, and a high voltage battery pack box, all of which were fitted using an enclosed light weight sub frame with a space of 0.23 m$^3$ designed specifically for this drivetrain design inside the front bay of a Holden Barina Spark (Figure 8.2). The light weight frame was made up of a sheet metal casing with sandwich panel core and cooling pipes that were run to facilitate the cooling for battery packs. The bottom of the sub frame was relieved to facilitate power supply and coolant flow. The rest of the space was utilised for BMS (0.003 m$^3$) and a box connecting the chassis to battery pack in the front bay of the Holden Barina Spark.
8.2.2 Rear bay fitment

Apart from the rear wheel storage, the rear bay of the Holden Barina Spark was used to store the motor controllers, capacitors and battery charger (Figure 8.3). The arrangement of the rear bay of the Holden Barina Spark consisted of two motor controllers in the spare wheel well. The space available after removing the spare wheel was 0.005m$^3$, which is a rectangular space of 0.55metre in a lateral direction, 0.58metre in a longitudinal direction and 0.17metre in a vertical direction. The space acquired by each motor controller was 0.014m$^3$. However the motor controllers were extending by 0.05metre outside the spare wheel well into the luggage space. The battery controller and contractors were other additional items fitted within the rear bay. The battery charger (0.016m$^3$) and a battery contactor (0.0004m$^3$) were also placed in the spare wheel space. The in-wheel SRM with the novel brake replaced the existing wheels and brake systems. The wheel motor and the brake system from the shaft were connected to the vehicle chassis using M12 bolts. The reconnection of the brake caliper module was done to ensure that the rear brakes were mechanically functional. A series of capacitors were connected and fitted inside the rear bay of the Holden Barina Spark.
The silencer pipe and fuel tank were not usable in the converted EV; hence the decision was taken to remove the fuel tank from inside the Holden Barina Spark. A wire harness from the motor was passed through the inner wheel shafts connecting the front and rear end through the silencer pipe with slight modifications. The modification to the existing silencer pipe of the Holden Barina Spark included removing the muffler (Figure 8.4) and welding a silencer pipe in location for passing the wire harness.

**Figure 8.4: Muffler and silencer pipe in an existing Holden Barina Spark ICE**

### 8.3 Vehicle weight distribution

The vehicle drive characteristics are dependent on the weight distribution of the vehicle. The CG of the vehicle was used to proportion the weight distribution in the vehicle. The
shift in the CG of the vehicle caused a change in the vehicle performance and the durability. Hence, volumetric analysis and weight distribution models were created to evaluate the vehicle drive characteristics. The Holden Barina Spark was a front wheel drive small car with a weight distribution for the car was 59:41, which meant that 59% of the weight was concentrated in front of the longitudinal CG of the car, and the other 41% was at the rear of the car, as discussed in chapter 6. The lateral CG was in the centre of the vehicle as shown in Figure 8.5.

Following are the data sets obtained from the CG of the Holden Barina Spark in different directions (from chapter 6 and Figure 8.5):  i) longitudinal direction from the front axle \( (l_f) \) is 975mm and longitudinal direction from the rear axle \( (l_r) \) is 1400, ii) vertical direction \( (h_{CG}) \) is 545mm above the ground and iii) lateral direction from both wheels toward the centre of the car \( (la_l \text{ and } la_r) \) is 703.5mm. The standard Holden Barina Spark was an EV when fitted with an in-wheel SRM and other ancillaries. This involved major changes in the design of the vehicle, including the complete removal of many different parts of the ICE and redesign of parts to develop a new EV. This led to variations in the weight redistribution of the vehicle, and changed the CG of the EV. The change in the CG affected the dynamics of the EV performance, and it therefore became important to verify the effects of these changes. Hence, a new CG was established for the developed EV and effects of this new CG were examined in the context of determining the vehicle performance. First the weight distribution variations in longitudinal and lateral were studied. The worst-case scenario of the vehicle position at an inclined angle was examined for vertical forces, and the relevant effects on the EV were examined.

*Figure 8.5: Lateral CG of the Holden Barina Spark ICE (dimensions in mm)*
8.3.1 Longitudinal weight distribution

The CG of the EV varies in the longitudinal direction, due to weight redistribution as a result of replacement of many parts from the ICE. *Table 8.1* shows the weight distribution of the car in the longitudinal direction. A total of 104kg was added to rear and 54kg to the front of the EV. The weight added on the front side was more than the weight added on the rear side.

*Table 8.1: Longitudinal weight variations of an EV*

<table>
<thead>
<tr>
<th></th>
<th>Front</th>
<th></th>
<th>Rear</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Subtraction Items</strong></td>
<td></td>
<td><strong>Weight (kg)</strong></td>
<td><strong>Weight (kg)</strong></td>
<td><strong>Subtraction Items</strong></td>
<td><strong>Weight (kg)</strong></td>
</tr>
<tr>
<td>Engine</td>
<td>140</td>
<td>Main Battery</td>
<td>202</td>
<td>Spare wheel</td>
<td>12</td>
</tr>
<tr>
<td>Radiator</td>
<td>8</td>
<td>Aux. Battery</td>
<td>12</td>
<td>Existing wheels and brakes</td>
<td>24</td>
</tr>
<tr>
<td>Gear Box</td>
<td>10</td>
<td>Power Pack</td>
<td>1</td>
<td>Motor controller</td>
<td>20</td>
</tr>
<tr>
<td>Air filter</td>
<td>2</td>
<td>BMS</td>
<td>3</td>
<td>Motor</td>
<td>80</td>
</tr>
<tr>
<td>Battery</td>
<td>12</td>
<td>Box to connect HV pack to chassis</td>
<td>1</td>
<td>Motor</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Super-controller</td>
<td>2</td>
<td>Main battery charger</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Sub frames</td>
<td>5</td>
<td>Battery contractor</td>
<td>1</td>
</tr>
<tr>
<td>Total weight removed</td>
<td>172</td>
<td>Total weight added</td>
<td>226</td>
<td>Total weight removed</td>
<td>36</td>
</tr>
<tr>
<td></td>
<td></td>
<td><strong>Total weight added</strong></td>
<td></td>
<td><strong>Total weight added</strong></td>
<td>140</td>
</tr>
</tbody>
</table>

Extra weight added in front bay $(A_f)$ 54 Extra weight added in rear bay $(A_r)$ 104

The car weight distribution on rear axle $F_{wr}$ in longitudinal direction is given by *Equation 8.1*:

$$F_{wr} = F_{wt} \frac{l_f}{l}$$  \hspace{1cm} (8.1)

Where, $F_{wt}$ was the total weight of the ICE (960kg), $l_f$ was longitudinal distance from the front axle (975mm) and $l$ was the wheel base distance (2375mm). Using *Equation 8.1* weight on the rear axle of an ICE was calculated as 394kg. Also from below *Equation 8.2* weight on the front axle of an ICE $F_{wf}$ was calculated as 566kg.

$$F_{wf} = F_{wt} - F_{wr}$$  \hspace{1cm} (8.2)
Equations 8.3 and 8.4 were used to calculate the weight on the front axle of an EV $F'_{wf}$ and the weight on rear axle of an EV $F'_{wr}$:

\begin{align*}
F'_{wf} & = F_{wf} + A_f \quad (8.3) \\
F'_{wr} & = F_{wr} + A_r \quad (8.4)
\end{align*}

Where, $A_f$ and $A_r$ were difference of the weight added from Table 8.1 in front and rear bays respectively. Weight distribution of the car at front was 620kg and rear was 498kg making new weight of an EV $F'_{we}$ as 1118kg. Using Equation 8.1 substituting the weight of the EV ($F'_{we}$), rear axle weight of the EV ($F'_{wr}$) and wheel base length ($l$), the longitudinal distance from the front axle $l_f$ was calculated as 1058mm. Thereby longitudinal distance from the front axle $l_f$ was calculated as 1317mm. The new CG and the distance of front and rear axle in an EV are illustrated in Figure 8.6. The forces acting on the front wheel and the rear wheel were calculated using these dimensions from Equation 8.5:

\begin{equation}
F = m'g \times \frac{x}{l} \quad (8.5)
\end{equation}

Where $m'$ was weight of the EV (1118kg) and $x$ was the longitudinal distance from front ($l_f$) or rear ($l_r$) axle. The force on the front wheel $F_f$ was calculated as 6.08kN and the force on the rear wheel $F_r$ was calculated as 4.88kN. The weight distribution ratio from front to rear of the EV was 56:44. The longitudinal CG of the EV moved towards
the rear by 6% compared to the position in the ICE version. The major reason for this was that the more weight was added to the rear bay than the front bay.

### 8.3.2 Lateral weight distribution

The lateral CG of the ICE Holden Barina Spark version was at the centre. For an EV, the CG variation along its lateral axis was developed to examine the effects of the weight distribution.

**Table 8.2: Lateral weight variations of an EV**

<table>
<thead>
<tr>
<th>Subtraction Items</th>
<th>Weight (kg)</th>
<th>Addition Items</th>
<th>Weight (kg)</th>
<th>Subtraction Items</th>
<th>Weight (kg)</th>
<th>Addition Items</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine (56%)</td>
<td>78.4</td>
<td>Main Battery</td>
<td>123.2 (56%)</td>
<td>Engine (56%)</td>
<td>61.6</td>
<td>Motor battery</td>
<td>96.8 (44%)</td>
</tr>
<tr>
<td>Radiator (50%)</td>
<td>4</td>
<td>Aux. Battery Pack</td>
<td>12</td>
<td>Radiator (50%)</td>
<td>4</td>
<td>Aux. Battery Pack</td>
<td>96.8 (44%)</td>
</tr>
<tr>
<td>Gear Box (100%)</td>
<td>10</td>
<td>Power supply for aux. pack</td>
<td>12</td>
<td>Existing wheels and brakes (50%)</td>
<td>12</td>
<td>New wheels and brakes</td>
<td>96.8 (44%)</td>
</tr>
<tr>
<td>Air filter (100%)</td>
<td>2</td>
<td>Sub frames</td>
<td>6</td>
<td>Spare Wheel (50%)</td>
<td>6</td>
<td>Sub frames</td>
<td>96.8 (44%)</td>
</tr>
<tr>
<td>Car Battery (100%)</td>
<td>12</td>
<td>HV connector (100%)</td>
<td>1</td>
<td>Main battery charger</td>
<td>4</td>
<td>Motor (50% at left side)</td>
<td>40</td>
</tr>
<tr>
<td>Current wheel and brakes (50%)</td>
<td>12</td>
<td>Super-controller (100%)</td>
<td>2</td>
<td>Motor (50% at left side)</td>
<td>40</td>
<td>Motor Controller (45%)</td>
<td>9</td>
</tr>
<tr>
<td>Spare wheel</td>
<td>6</td>
<td>Battery Contactor (100%)</td>
<td>0.5</td>
<td>Motor (50%)</td>
<td>40</td>
<td>Sub frames (50%)</td>
<td>2.5</td>
</tr>
<tr>
<td>Motor (50%)</td>
<td>40</td>
<td>Sub frames (50%)</td>
<td>2.5</td>
<td>Motor Controller (55%)</td>
<td>11</td>
<td>Changed wheel and brake (50%)</td>
<td>16</td>
</tr>
<tr>
<td>Sub frame (50%)</td>
<td>2.5</td>
<td>Changed wheel and brake (50%)</td>
<td>16</td>
<td>Motor Controller (55%)</td>
<td>11</td>
<td>Changed wheel and brake (50%)</td>
<td>16</td>
</tr>
<tr>
<td>BMS (100%)</td>
<td>3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total weight removed</td>
<td>172</td>
<td>Total weight added</td>
<td>199.2</td>
<td>Total weight removed</td>
<td>83.6</td>
<td>Total weight added</td>
<td>168.3</td>
</tr>
</tbody>
</table>

**Extra weight added in left bay \(A_{l_0}\)** | 74.8 | **Extra weight added in right bay \(A_{r_0}\)** | 84.7 |
Figure 8.7: Lateral CG of the Holden Barina Spark EV (dimensions in mm)

The change in the CG was calculated by evaluating the weight redistribution along the lateral axis. Table 8.2 represents the weight redistribution of the EV. As the CG of ICE is equally distributed the ratio of weight difference in left to right is 50:50 in lateral axis. From Figure 8.7 the CG is equally divided in lateral axis; as a result the weight on each side was half the weight of the ICE. The weight on the left side of the ICE car $F_{wle}$ and the weight on the right side of the ICE car $F_{wri}$ were calculated as 480kg (50% of 960kg). These values substituted $F_{wfr}$ and $F_{wrr}$ in Equations 8.3 and 8.4. Using Equations 8.3 and 8.4, the weight on the left side of an EV $F'_{wle}$ (554.8kg) and the weight on the right side of an EV $F'_{wri}$ (564.7kg) were calculated by substituting the $A_f$ and $A_r$ with $A_{le}$ and $A_{ri}$ from Table 8.2. Total weight in lateral axis $F_{wlt}$ was calculated as 1119.5kg. Using Equation 8.1 substituting the weight of the EV $F'_{wlt}$, the right side weight of the EV ($F'_{wri}$) and the vehicle width ($la'$) from Figure 8.7, the lateral distance from the left side $la_l'$ was calculated as 708.72mm. Thereby lateral distance from the right side $la_r'$ was calculated as 696.28mm. The lateral CG of the EV moved towards the right of the car by 0.5% from its original location with reference to the left wheel of the car. The movement of CG was attributed to more weight being added to the right of the EV during the conversion.

8.3.3 Vertical weight distribution

The vertical CG of the car also moved after the conversion, as the extra weight was added to the vehicle. The CG of an ICE was $h_{CG}$ was 545mm above the ground as discussed in chapter 6. To calculate the height of the CG in the EV fitted with in-wheel SRMs, the force acting on the rear wheel of the ICE at an inclination of 45° was used. The force on the rear wheel of an ICE was calculated using Equation 8.6:
Where, $F_{zx}$ was force acting on the rear wheel of the EV on an 45° angle incline $\alpha$, $m$ was weight of the ICE (960kg), $l_f$ was distance between the front axle and the longitudinal CG of the ICE (965mm), $h_{CG}$ was CG height from ground plane in an ICE (545mm), and $l$ was wheel base of the ICE (2375mm).

![Figure 8.8: Forces acting on the vehicle at an inclined position](image)

So, the force of the rear wheel $F_{zx}$, as shown in Figure 8.8 was off centre due to deformation in the tyres at the tyre-road contact. This force was calculated to be 2.73kN.

Similarly, force of the rear wheel $F'_{zx}$ for the EV on a 45° angle incline $\alpha$ was calculated using Equation 8.6. In equation by substituting, $m$ with $m'$ (weight of the EV 1120kg), $l_f$ with $l'_f$ (distance between the front axle and the longitudinal CG of the EV 1058mm), $h_{CG}$ was CG height from ground plane in an ICE (545mm), and $l$ was wheel base of the EV (2375mm). This force was calculated to be 3.46kN. Using these forces the new CG height $CG_{he}$ for an EV was calculated using Equation 8.7:

$$CG_{he} = h_a + \{(F'_{zx} - F_{zx}) \times l' \times l_i\}/\{m'g \times (h_{ai} - h_a)\}$$  \hspace{1cm} (8.7)

Digital model was used to obtain the values for $h_a$ height of the front axle from the ground (216mm), $h_{ai}$ height of the front axle of the elevated EV (2375.10) and $l_i$ length of the reduced wheel base when the EV was on the incline (1679.10 mm).
These values were used to calculate the height of the CG of the EV, which was 381mm above the ground. These results showed that the height of the CG moved down in the EV due to the addition of extra weight. The height of the CG was 381mm above the ground height as compared to 545mm in the ICE. These studies indicated that the new CG had moved 43% toward the wheelbase. This resulted in better stability and car control for the EV as the height of CG was brought closer to the wheelbase. The CG variations were not substantial with 6% variations in the longitudinal and 0.5% in lateral directions. The vehicle performance was not majorly affected due to these changes for the developed EV with an in-wheel SRM.

8.4 Vehicle rolling using tyre longitudinal slip

In this section, the tyre longitudinal slip is modelled to determine the changes in the vehicle rolling as a consequence of the increased motor mass. When braking, a moment $M_b$ is applied to the tyres, causing the contact zone to stretch, which leads to an increase in the radius at the point of contact to $R'_e$. This causes a decrease in the tyre angular velocity ($\Omega$) at that point. During the rolling, the effective tyre rolling radius is $R_e$ and the contact area between the ground and the tyre is compressed rather than stretched, thus shifting $R_e$ more towards $R$ in the range between $R$ and $R_l$ and leading to an increased angular velocity ($\Omega_0$). In such conditions, it is possible to define a longitudinal slip, given by Equation 8.8:

$$\sigma = \frac{\Omega}{\Omega_0} \tag{8.8}$$

The longitudinal force $F_x$ acting on the tyre is a function of the longitudinal slip $\sigma$. The longitudinal force becomes zero in free rolling conditions (when longitudinal slip is zero) and changes almost linearly, when longitudinal slip is between -0.15 to -0.3 and 0.15 to 0.3 during driving. The longitudinal force varies proportionally to the vertical force $F_z$. The longitudinal slip $\sigma_x$ defines a relationship between longitudinal and vertical forces as shown in Equation 8.9:

$$\sigma_x = \frac{F_x}{F_z} \tag{8.9}$$

As the in-wheel SRM leads to the increased vertical load ($F_z$), the longitudinal force also increases based on the above relationship. The higher the value of $F_x$ the greater the longitudinal slip, whereby the slip ranges from the positive longitudinal force during
driving traction, to zero during free rolling to negative longitudinal force during braking traction. The tyre motion and the braking are described by Equation 8.10:

\[
J \frac{d\alpha}{dt} = F_x \times R_l - M_b \tag{8.10}
\]

The longitudinal slip is also affected by environmental conditions (e.g., wet conditions such as hydroplaning) and side lateral force \( F_y \). The assumption was made to implicitly define the longitudinal force coefficient. The magnitude of longitudinal force \( F_x \) was calculated using the magic formula also called a Pacejka tyre model. This tyre model provided a relationship between the longitudinal force \( F_x \) and the longitudinal slip \( \sigma \), as per Equation 8.11:

\[
F_x = D \cdot \sin(C \tan^{-1}\{B \cdot (1 - E)(\sigma + S_h) + E \cdot tan^{-1}[B \cdot (\sigma + S_h)]\}) + S_v \tag{8.11}
\]

In this research the longitudinal force is modelled based on the car travelling on the dry road using the low rolling resistance 205/50/R17 tyre. The effects of the side force \( F_y \) and wet conditions were not considered in this study. In Equation 8.11, \( B, C, D, E, S_v, \) and \( S_h \) are six rolling coefficients dependent on the vertical load \( F_z \) and the angle \( \alpha \). The values of the rolling coefficients are expressed as a function of coefficients \( b_i \). The coefficient \( b_i \) values are based on data obtained by supplier for a low rolling resistance 205/50/R17 tyre. The relationships for \( C \) and \( D \) coefficients are given by Equations 8.12 and 8.13, as follows:

\[
C = b_0 \tag{8.12}
\]

\[
D = \mu_p F_z \tag{8.13}
\]

Where \( \mu_p \) is the peak value of the longitudinal force coefficient given by Equation 8.14:

\[
\mu_p = b_1 F_z + b_2 \tag{8.14}
\]

The \( C \) and \( D \) coefficients are used to calculate coefficient \( B \) given by Equation 8.15:

\[
B = \frac{(b_2 F_z^2 + b_4 F_z) e^{-b_5 F_z}}{CD} \tag{8.15}
\]

The coefficient \( E \) is given by Equation 8.16:
\[ E = b_6 F_z^2 + b_7 F_z + b_8 \] (8.16)

\( S_v \) is 0 and coefficient \( S_h \) is calculated by \textit{Equation 8.17}:

\[ S_h = b_9 F_z + b_{10} \] (8.17)

The values of coefficients \( b_i \) obtained are listed in \textit{Table 8.3} below:

\begin{table}[h]
\centering
\begin{tabular}{|c|c|}
\hline
\textbf{Coefficient of } \( b_i \) & \textbf{Value} \\
\hline
\( b_0 \) & 1.65 \\
\hline
\( b_1 \) & -7.63 \\
\hline
\( b_2 \) & 1120.6 \\
\hline
\( b_3 \) & -0.008 \\
\hline
\( b_4 \) & 144.7 \\
\hline
\( b_5 \) & -0.07 \\
\hline
\( b_6 \) & -0.004 \\
\hline
\( b_7 \) & -0.09 \\
\hline
\( b_8 \) & -0.077 \\
\hline
\( b_9 \) & -0.025 \\
\hline
\( b_{10} \) & -0.025 \\
\hline
\end{tabular}
\end{table}

The values of \( B, C, D, E, S_v, \) and \( S_h \) for the present study were obtained from \textit{Equations 8.12 to 8.17} using appropriate \( b_i \) values. Total load of the ICE Holden Barina Spark (weight of car 10kN) was taken as 16kN with five passengers and luggage on board (6kN). Total load of an in-wheel SRM Holden Barina Spark (extra 2kN compensating motors and batteries) was taken as 18kN with five passengers and luggage on board. Hence vertical force \( F_z \) was derived as 4kN for the ICE Holden Barina Spark tyre (referred as ICE tyre) and the in-wheel SRM Holden Barina Spark tyre (referred as EV tyre) as 4.5kN. The comparisons with the ICE tyre were performed in order to determine the change in longitudinal force and slip for an EV tyre.

The relationship between the longitudinal force \( F_x \) and the longitudinal slip \( \sigma \) was established, based on \textit{Equation 8.11}. Figure 8.9 shows variation of the longitudinal force on a car tyre for both ICE and EV versions, with varying values of the longitudinal slip. The values \( \sigma \) ranged from the positive slip (\( \sigma > 0 \)) during driving traction, zero during the free rolling (\( \sigma = 0 \)) and negative slip during braking traction (\( \sigma < 0 \)). The comparison between the two cars established that there was slight variance in the drive characteristics of the EV when compared to the ICE car. Although these
variations were negligible, it was concluded that an increase in the EV longitudinal slip tyre was due to increased mass on the tyre. Hence the effect of the motor mass has not affected the tyre ride conditions for the proposed in-wheel SRM.

![Variation of longitudinal force with slip (ICE & EV)](image)

**Figure 8.9:** Variation of longitudinal force with slip (ICE & EV)

### 8.5 Tyre service evaluation using augmented reality

Easy installation of the in-wheel SRM was one of the key criteria in this study, since it was an essential element due to the need to periodically change tyres for servicing in long runs. In the proposed EV, when work is carried out due to maintenance issues, a single operator (mechanic) generally performs tyre changing or wheel/brake servicing. Two characteristics are thus important:

- Operator safety, whereby it is essential to prevent lower back pains associated with prolonged crouching position during servicing, as well as finger entrapment in the openings (less than 10mm gap from mudguards as discussed in chapter 3).
- Easy detachment of tyre rims from the motor during servicing.

#### 8.5.1 Virtual reality modelling

As a part of this research, VR-based tyre service evaluation studies were conducted for the in-wheel SRM. The computational models and virtual modelling simulations using motion capture, Arena® and EON® reality, mimicked live system environments, allowing the assessment of the effectiveness of the tyre rim assembly and disassembly
functionality using human as an interface. Initial stage consisted of schematic representations of models used to evaluate conceptualisation for different designs. This process was sequenced in three main stages: i) digitised model of the vehicle (using scanner discussed in chapter 3), ii) developed design (suspension, brake, and SRM) and iii) adding human interface by capturing actual human motion capture. These three data were modelled using VR tools. The required environment was added using available libraries or by modelling the ones required (i.e., garage and tyre service centre). Then the motion capture facilitated the real time simulation of tyre change sequencing. The avatar within the virtual environment performed the predefined tasks to analyse the sequence and physical entrapment dangers (i.e. fingers of the operator).

The in-wheel SRM, rim-tyres and vehicle digital models were recreated in the VR environment. Then digitised vehicle was modelled with the required material properties and textures. Lastly the environment in this case a garage was modelled and the scenario was created. Then the highlighter tools were predefined in the model to auto caution during the real time simulation (e.g., interference detect). The control triggers were also developed within the model to effectively perform required tasks using peripherals (e.g., mouse or hand held controller). Now this scenario was bought onto the VR screen with the triggers for real time simulations modelled.

8.5.2 Augmented reality modelling

In the second stage, motion capture was used, whereby 20 camera systems video-recorded the typical range of human movements and rigid body, such as a physical tyre, to develop a simulation of the live environment. Finally, all these stages were interfaced together in a VR environment to evaluate assembly and disassembly functions of motor rim design. Based on the evaluation of these processes, the initial designs were fine-tuned for effective assembly functionality. The AR-based real time safety and ergonomic evaluation procedures were used for demonstration of wheel assembly and disassembly functions by a single operator. These evaluations were novel for automotive design, as they were performed without substantial prototype costs typically associated with safety and ergonomic studies.

Assembly sequencing was determined in the design using motion capture system, which used 20 cameras using Arena® to generate mock up. The motion capture process consisted of camera calibration, whereby easy skeleton creation based on physique of
the personnel allowed recording multiple actors and real time motion capture. The calibration processes used the wand to wave and identify the camera settings. Experimentation for motion capture is illustrated in Figure 8.10, where reflective clothing and one of the cameras is shown. Figure 8.11 shows the Arena® digital model (avatar) with camera 11 (only one camera shown as an example) for motion capture using real time capture of the human motions. Motion capture clothing (marker suit) had reflective markers, allowing the infrared camera to capture reflection of the glow marks that represented real time human motion as spatial positions (3D points). The reflective markers placed on actors and props were used, as this created a cloud of 3D points. These 3D points were subsequently labelled and mapped to a skeleton solver for tracking full body motion. Captured actor and prop data were exported to VR models in real time using standard 6 DOF digital file format for further 3D animation and analysis. After the process described above was completed, rigid objects, such as tyres and spanner, were added into the virtual model. Then physical components were also attached with glow marks. With the glow marks attached the infrared camera detects and identifies the motion of these rigid bodies.

\[\text{Figure 8.10: Motion capture experimentation for tyre servicing}\]

The designed motion data was integrated as a 3D animation within the VR environment. The motion capture data was then developed into a BVH file using Arena® software, where the motions were edited in order to eliminate unwanted movements from the captured data, mostly due to inconsistencies of the flow in the motion path.
Generally, the affected data sets were broken into small segments in edit section to improve the flow of motion. These discrete areas were identified and processed to remove unwanted movements, as shown in Figure 8.12.

The data purging process consisted of highlighting the area of motion and editing the curvature in order to produce smooth curves that represented consistent motion. The BVH files subsequently were imported into VR models and attached to Bipeds, which automatically engaged the motion data and served as actors.

**Figure 8.11:** Digital model mimicking the actual tyre change scenario

**Figure 8.12:** Discrete motions within BVH file
8.5.3 Biomechanical modelling

Lower back pain analysis is of particular importance, as a quarter of the Australian population is currently affected by this issue, costing the health care system nearly A$9 billion, including direct and indirect costs (Walker, Muller et al. 2004). When moving from standing up to bending down with a weight, the lumbar spine goes from lordotic (straight) to kyphotic (curved) and this creates the risk of lower back pain. The human spine has 33 vertebrae bones to balance the centre of mass in a body. The weight of load and body mass are transmitted as compressive forces and this creates moments on these bones. This may result in lower back pain at the L5/S1 disc (lumbar 5 and fused sacral vertebrae 1). There is an action limit \( A_l \) imposed on the weight load by the National institute of Occupational safety and Health (NIOSH) for repetitive loads, which is also a revised edition of original 1981 NIOSH equation (Thomas, Vern et al. 1994). Safety analysis was conducted to analyse lower back pain risks for an operator during typical tyre servicing by embedding the load cases into C-motion® within AR environment.

![Biomechanical model front view for tyre servicing (dimension in mm and weight in Newton): a) during servicing, b) start of servicing](image)

**Figure 8.13:** Biomechanical model front view for tyre servicing (dimension in mm and weight in Newton): a) during servicing, b) start of servicing

Figure 8.13a depicts the loads and the distances used to model the tyre change scenario. The present analysis was based on modelling the upper body torso weight of \( W_t \) and...
lifting a load $W_l$, the total compression force $F_c$ required by internal spine at L5/S1 to overcome the moments $M_{lt}$ and internal muscle $F_{im}$ force by *Equation 8.18*:

$$F_c = M_{lt} + F_{im} \quad (8.18)$$

Whereby the relevant moment $M_{lt}$ and force $F_{im}$ were given by *Equation 8.19* and 8.20:

$$M_{lt} = \left( W_l h + W_l b \right) / c \quad (8.19)$$

$$F_{im} = \left( W_l + W_t \right) \cos \alpha \quad (8.20)$$

Whereby, $h$ was the distance from the load to the L5/S1 disc (Lumbar 5 and fused sacral vertebrae 1), $b$ was the distance from the centre of the body to the L5/S1 disc and $c$ was the distance from internal muscle force to the L5/S1 disc. Considering a person weighing 350N lifting a 200N load, at an angle $\alpha$ of $\circ$, the total force required at L5/S1 was calculated as 3.92kN. This figure at L5/S1 was much larger when compared to the lifting load of 200N and was required to keep the body in equilibrium. This figure was within safe limits of less than 5.5kN as proposed by Farfan H (Farfan 1973). The revised NIOSH lifting equation was used for tyre lifting and transfer by operators as per *Equation 8.21* given below (Thomas, Vern et al. 1994):

$$A_t = K \cdot H_f \cdot V_f \cdot D_f \cdot F_f \cdot C_f \cdot A_f \quad (8.21)$$

Where $K$ was the load constant (23kg), $H_f$ was the horizontal reduction factor (1) and $V_f$ was the vertical reduction factor (0.88), $D_f$ was the distance reduction factor (1.05), $F_f$ was the frequency reduction factor (0.99), $C_f$ was the coupling reduction factor (1) and $A_f$ was the asymmetric factor (0.97). The reduction factor parameters used for these were based on *Figure 8.13b*, where values of horizontal origin $H_o$ (250mm) and vertical origin $V_o$ (340mm) were taken from inner ankle bones mid plane and inner hand bones mid plane. The $D_t$ (200mm) was the distance travelled in vertical plane by the load. The following *Table 8.4* represents values of $K, H_f, V_f, D_f, F_f, C_f,$ and $A_f$, which were calculated from the corresponding equations. The $F_{avg}$ and $F_{max}$ were the average and maximum frequencies and ratio was calculated based on the frequency multiplier table from revised NIOSH equations (Thomas, Vern et al. 1994). Coupler reduction factor $C$ was 1 as the tyre was classified as fair with more than 75mm gripping faces. The
asymmetric reduction factor $A_f$ was calculated at 60° angle rotation between the half section of torso in XY plane in Figure 8.13a.

**Table 8.4: Reduction factors for NIOSH equation**

<table>
<thead>
<tr>
<th>Description</th>
<th>Equations</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load constant ($K$)</td>
<td>Constant</td>
<td>23kg</td>
</tr>
<tr>
<td>Horizontal reduction factor ($H_f$)</td>
<td>$25/H_o$</td>
<td>1</td>
</tr>
<tr>
<td>Vertical reduction factor ($V_f$)</td>
<td>$1-(0.003(V_o-75))$</td>
<td>0.88</td>
</tr>
<tr>
<td>Distance reduction factor ($D_f$)</td>
<td>$0.82+(4.5/D_t)$</td>
<td>1.05</td>
</tr>
<tr>
<td>Frequency reduction factor ($F_f$)</td>
<td>$1-(F_{avg}/F_{max})$</td>
<td>0.99</td>
</tr>
<tr>
<td>Coupler reduction factor ($C$)</td>
<td>Good</td>
<td>1</td>
</tr>
<tr>
<td>Asymmetric reduction factor ($A_f$)</td>
<td>$1 - \frac{0.0032\pi\theta}{180}$</td>
<td>0.97</td>
</tr>
</tbody>
</table>

NIOSH lifting *Equation 8.21* was used to calculate the action limit as 20.2kg. The result of this analysis indicated that the operators were not at risk of back injury whilst performing the tyre changing lifting task due to the fact that the lifting index ($L_I$) of 0.98 associated with this job was slightly below the recommended limit of 1.0. The lifting index was the ratio of load weight and the calculated action limit. Nonetheless, necessary precautions were still required to minimise the risk of injuries. These results concluded with a favourable outcome for the tyre change scenario.

**8.5.4 Augmented reality real time simulations**

Finally, the overall assembly was simulated in order to create the working environment. The motion capture data was integrated into the design for assembly feasibility studies, as shown in *Figure 8.14*. Digital validations of wheel design using mock-up of motion build were performed in order to establish the optimal assembly task sequencing for tyre disassembly servicing or tyre repair by a single person for an in-wheel SRM. For an in-wheel SRM, it was essential to ensure that unsprung mass due to motor mass was not an issue when a tyre was changed or serviced in the long run. Other potential associated risks were examined, such as operator finger entrapment due to low clearance between mudguards and tyres.
In an AR environment, rigid objects, such as tyres and spanner, were used and the motion capture and complex simulations were generated. The physical motion was being transformed to a virtual environment whereby a digital human model worked on the virtual prototype for a tyre disassembly. The following were fine-tuned motions through these methods and led to subsequent outcomes:

1) The operator bends and performs the unscrew operation using spanner for all the 8 bolts. Similarly the VR digital model performs the same operation on the digital vehicle in a real time.

2) Easy dismantling of the assembly was possible in the virtual model. As a consequence when the tyre was disassembled the motor and brakes were retained with the vehicle, while the central shaft inside the hub held the entire motor assembly in place. Easy and error-free assembly sequencing for enclosing electric motor into a wheel for an EV was established. This resulted in an inclusive design approach, facilitated by creating an interface between human factor and design.

3) The operator slides the tyre outside. In the VR digital model the interference triggers and warns the danger of interference. As the tyre was protruding outside the mudguard the operator should only use that area at the start. This indication was a clear identification of a possible occupational hazard mitigated with an appropriate standard operating procedure (SOP) or user manual (UM, whereby SOP or UM was informing the operator or the driver about possible hazard. This ensured operator safety and reduced the likelihood of injuries, such as finger
entrapment. Establishment of easy tyre dismantling protocol through UM can be used in emergencies when only a single mechanic or operator is performing all the functions. Safety and performance of a single operator are improved when operator was required to perform tyre change or maintenance activities for an in-wheel SRM. These safety tests included the evaluation of the appropriate clearance between the wheel mudguard and finger, as well as the required space for tool entry for tyre dismantling by an operator.

4) Also using a biomechanical model with AR environment the weight of the wheel was indicated as an issue if performed repetitively. As a consequence in a regular service centre (tyre changing suppliers) it is essential to have an automated pelleting system to transfer the wheel loads from the operator.

### 8.6 Vehicle range

This section discusses the battery pack selection process for the proposed Holden Barina Spark EV. In addition, a detailed analysis of power consumption was carried out to give an estimation of the range of the EV. The selection of the battery pack used was mainly based on the power requirement of the motor. The proposed EV was powered by two in-wheel SRMs of 15kW per motor. The ideal operating voltage for the motor was above 200V. The battery chosen for this research was a HED100series LiFePO$_4$ Prismatic type. The key specifications of these batteries include:

- Capacity rating, nominal 1C: 100Ah
- Cell voltage nominal: 3.25V
- Cell discharge current, maximum consistent discharge current: 300A
- Cell weight nominal 2.8kg

The battery pack included 72 battery cells in a series that used a nominal voltage of 234V (72 × 3.25) and a maximum consistent discharge current of 300A (3 × 100A). Hence, the selected battery pack satisfied the power requirement required by the system. Detailed calculation of total energy that was consumed by the motors was made to estimate the range of the proposed EV. The battery energy $E_b$ was calculated as shown below in Equation 8.22:

$$E_b = B_c \cdot M_e \cdot MC_e \cdot N \cdot V$$  \hspace{1cm} (8.22)
Where by $B_c$ was actual battery consumption (80% of rated capacity, 0.8 of 100Ah), $M_e$ and $MC_e$ were motor and controller efficiencies (considered as 90% efficient, 0.9), $N$ was number of cells and $V$ was average cell voltage (3.25V). Using Equation 8.22, battery energy was found to be 147kWh and 52908kJ.

The estimation of the battery pack was carried out based on the U.S Environmental Protection Agency (EPA) range-testing model. The estimation was conducted for two driving conditions: i) urban and ii) highway. The estimation of range was based on the car rolling on a plane surface with 1200kg total weight at an ambient temperature of 22°C, and in an air density of 1.225kg/m³. The regenerative and windy conditions were neglected as per the EPA standard. The test conditions for the two scenarios were considered as shown in Table 8.5:

**Table 8.5: Test conditions for the vehicle**

<table>
<thead>
<tr>
<th>Test condition</th>
<th>Urban</th>
<th>Highway</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average speed (km/h)</td>
<td>31.53</td>
<td>77.73</td>
</tr>
<tr>
<td>Average speed (m/s)</td>
<td>8.8</td>
<td>21.6</td>
</tr>
<tr>
<td>Test time(s) ($t_t$)</td>
<td>1352</td>
<td>754</td>
</tr>
<tr>
<td>Number of acceleration ($N_a$)</td>
<td>19</td>
<td>3</td>
</tr>
</tbody>
</table>

The power consumption of the car under test was considered as including a drag force and resistance force component and an acceleration component. The acceleration power required $P_a$ was calculated as per Equation 8.23:

$$P_a = \frac{(N_a \times 0.5m.v^2)}{t_t}$$  \hspace{1cm} (8.23)

Total force required was based on the following Equation 8.24 (discussed in chapter 5) of motion when gradient $\alpha$ was zero (when the car was moving on a plane surface):

$$F_t = 0.5\rho.C_d.A. v^2 + C_r.m.g$$  \hspace{1cm} (8.24)

Where, $F_t$ is the total force to overcome aerodynamic resistance offered by a vehicle at speed $v$ through air of density $\rho$ with coefficient of drag resistance $C_d$ offered by the vehicle frontal area $A$, with the vehicle weight $m$ and a rolling resistance coefficient $C_r$ at acceleration due to gravity $g$. The power required $P_t$ for the vehicle was then calculated as Equation 8.25:
\[ P_r = v \times F_t \]  \hspace{1cm} (8.25)

Hence, the total average power \( P_t \) required for the car to roll at an average test speed was calculated by *Equation 8.26:*

\[ P_t = P_a + P_r \]  \hspace{1cm} (8.26)

And the average range \( A_r \) was calculated by *Equation 8.27:*

\[ A_r = v \times \left( \frac{E_b}{P_t} \right) \]  \hspace{1cm} (8.27)

**Table 8.6: Test results with 72 battery packs EV design**

<table>
<thead>
<tr>
<th>Condition</th>
<th>Velocity v(m/s)</th>
<th>Number of acceleration ( N_a )</th>
<th>Test time ( t ) (s)</th>
<th>Acceleration Power required ( P_a ) (W)</th>
<th>Power require ( F_t ) (W)</th>
<th>Average power ( P_t ) (W)</th>
<th>Range (km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Urban</td>
<td>8.8</td>
<td>19</td>
<td>1352</td>
<td>647</td>
<td>1415</td>
<td>2062</td>
<td>225</td>
</tr>
<tr>
<td>Highway</td>
<td>21.6</td>
<td>3</td>
<td>754</td>
<td>1113</td>
<td>5729</td>
<td>6842</td>
<td>167</td>
</tr>
</tbody>
</table>

*Table 8.6* represents a summary of the calculations for the range of the vehicle. Considering the battery packs, this reduced the mass of batteries to approximately 100kg (2.8kg X 36 battery packs) bringing the total weight to 1100kg. *Table 8.7* is a summary of calculations for 36 battery packs:

**Table 8.7: Test results with 36 battery packs EV design**

<table>
<thead>
<tr>
<th>Condition</th>
<th>Velocity v(m/s)</th>
<th>Number of acceleration ( N_a )</th>
<th>Test time ( t ) (s)</th>
<th>Acceleration Power required ( P_a ) (W)</th>
<th>Power require ( F_t ) (W)</th>
<th>Average power ( P_t ) (W)</th>
<th>Range (km)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Urban</td>
<td>8.8</td>
<td>19</td>
<td>1352</td>
<td>593</td>
<td>1312</td>
<td>1905</td>
<td>122</td>
</tr>
<tr>
<td>Highway</td>
<td>21.6</td>
<td>3</td>
<td>754</td>
<td>1020</td>
<td>5475</td>
<td>6495</td>
<td>88</td>
</tr>
</tbody>
</table>

The estimated range of the proposed Holden Barina Spark EV was found to be 225km for urban driving conditions and 167km for highway driving conditions. The actual range was expected to be 20% lower than the estimation to compensate for windy conditions and uneven surfaces. For the worst case, the range was proposed as 180km and 134km respectively. This range is acceptable for the majority of car drivers since their daily trip is usually below 100km. Hence the selected battery pack was appropriate
in for the selected vehicle. The above calculation also indicated that if the mass of the car was reduced by 100kg by decreasing the number of packs in the battery to half, the range of the car was almost halved as well. This range was quite low and probably not appropriate for normal driving conditions. Hence a 72 battery pack was recommended for the Holden Barina Spark as a result of these studies.

8.7 Specification sheet of an electric vehicle

Following performance studies, a final specification sheet was developed for the EV. Table 8.8 is a summary of these vehicle specifications for the developed EV. The vehicle specification sheet covers all the details of the developed vehicle including exterior dimensions, suspension types, type of brakes, motor specifications and battery details. The vehicle had 1.6 X 1.5 X 3.6m as the exterior dimensions with a tare weight of 1200kg after modifications. It had a track of 1407mm in the front and 1454 in the rear wheels. The tyres were 205/50 R17 specification of a low rolling resistance. The suspension at the rear involved a MacPherson strut with torsion beams. Two SRMs were used at the back inside wheels with a power output of 15kW each and dimensions of 412mm and 194mm width. They weighed 40kg on each wheel with RPM of 5000. The batteries used were the HED100 series LiFeMnPO4 Prismatic with a nominal capacity rating of 1C: 100Ah. The 72 battery packs of 200kg produced an urban range of 225km and a highway range of 167km as per EPA standards.

Table 8.8: Vehicle specification sheet

<table>
<thead>
<tr>
<th>Vehicle</th>
<th>Holden Barina Spark EV</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Exterior</strong></td>
<td></td>
</tr>
<tr>
<td>Length (mm)</td>
<td>3595</td>
</tr>
<tr>
<td>Width (mm)</td>
<td>1597 (excluding mirrors)</td>
</tr>
<tr>
<td>Height (mm)</td>
<td>1522</td>
</tr>
<tr>
<td>Tare (kg)</td>
<td>1200</td>
</tr>
<tr>
<td>Wheel base (mm)</td>
<td>2375</td>
</tr>
<tr>
<td>Track (mm)</td>
<td>1407/1454</td>
</tr>
<tr>
<td>Tyre</td>
<td>205/50/R17</td>
</tr>
<tr>
<td>Suspensions</td>
<td>Rear MAC Pherson Strut and Torsion Beam</td>
</tr>
<tr>
<td>Brake</td>
<td>Front drum and Rear disc</td>
</tr>
<tr>
<td><strong>Motor</strong></td>
<td></td>
</tr>
<tr>
<td>Engine Motor</td>
<td>2 motors Rear wheel drive</td>
</tr>
<tr>
<td>Drive</td>
<td>1 ” X 8” in-wheel drive</td>
</tr>
<tr>
<td>Motor type</td>
<td>SRM</td>
</tr>
<tr>
<td>Dimensions (mm)</td>
<td>412 X 194</td>
</tr>
</tbody>
</table>
The key points covered in this chapter were:

- The fitment of the motor inside using a digitised Holden Barina Spark was envisaged with VR tools. Parts such as the radiator, engine, gearbox, air box, and battery became redundant in the existing ICE. This provided usable space in the front engine bay of 0.31m$^3$. In this space, 18 main battery packs, 1 auxiliary battery pack, 3 power supply boxes for the auxiliary pack, and the BMS were placed. However, 18 main battery packs, 1 auxiliary battery pack, and the charger for the auxiliary pack were packaged in one sub-frame designed for fitment in a vehicle using an in-wheel SRM.

- The spare tyre space in the rear bay of the Holden Barina Spark was utilised for the fitment of controllers in the vehicle. This created a space of 0.005m$^3$, into which two motor controllers of dimension 0.014m$^3$ were fitted. However, the motor controllers were extending by 0.05 metre outside the spare wheel well into the luggage space. The battery charger (0.016m$^3$) and a battery contactor (0.0004m$^3$) were other additional items fitted within the rear bay.

- The wheel motor and brake design were fitted through the shaft to the vehicle chassis using M12 bolts. The reconnection of the brakes was established for its functionality. A wire harness from the motor was passed through the inner wheel shafts connecting the front and rear end of the vehicle through the silencer pipe by removing the muffler and welding an extra pipe.

- The CG variation of the vehicle was examined to understand the vehicle performance. The CG in the longitudinal varied by 6% and lateral directions varied by 0.5% and 43% on the height of the vehicle. These studies indicated that the CG shift was not substantial in the lateral and longitudinal directions. The variations in

<table>
<thead>
<tr>
<th>Weight (Kg)</th>
<th>40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power (Kw)</td>
<td>15</td>
</tr>
<tr>
<td>Torque (RPM)</td>
<td>30 @ 5000</td>
</tr>
</tbody>
</table>

**Battery**

<table>
<thead>
<tr>
<th>Type</th>
<th>HED100 series LiFePO$_4$ Prismatic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity rating</td>
<td>Nominal 1C: 100Ah</td>
</tr>
<tr>
<td>No of battery packs</td>
<td>72</td>
</tr>
<tr>
<td>Weight (Kg)</td>
<td>200</td>
</tr>
<tr>
<td>Range (km)</td>
<td>Urban 225 and highway 167</td>
</tr>
</tbody>
</table>

8.8 Summary

The key points covered in this chapter were:

- The fitment of the motor inside using a digitised Holden Barina Spark was envisaged with VR tools. Parts such as the radiator, engine, gearbox, air box, and battery became redundant in the existing ICE. This provided usable space in the front engine bay of 0.31m$^3$. In this space, 18 main battery packs, 1 auxiliary battery pack, 3 power supply boxes for the auxiliary pack, and the BMS were placed. However, 18 main battery packs, 1 auxiliary battery pack, and the charger for the auxiliary pack were packaged in one sub-frame designed for fitment in a vehicle using an in-wheel SRM.

- The spare tyre space in the rear bay of the Holden Barina Spark was utilised for the fitment of controllers in the vehicle. This created a space of 0.005m$^3$, into which two motor controllers of dimension 0.014m$^3$ were fitted. However, the motor controllers were extending by 0.05 metre outside the spare wheel well into the luggage space. The battery charger (0.016m$^3$) and a battery contactor (0.0004m$^3$) were other additional items fitted within the rear bay.

- The wheel motor and brake design were fitted through the shaft to the vehicle chassis using M12 bolts. The reconnection of the brakes was established for its functionality. A wire harness from the motor was passed through the inner wheel shafts connecting the front and rear end of the vehicle through the silencer pipe by removing the muffler and welding an extra pipe.

- The CG variation of the vehicle was examined to understand the vehicle performance. The CG in the longitudinal varied by 6% and lateral directions varied by 0.5% and 43% on the height of the vehicle. These studies indicated that the CG shift was not substantial in the lateral and longitudinal directions. The variations in
height favoured the EV design with a CG shift of 43% towards the ground, thus making the vehicle more stable than to the ICE.

- Using Pacejka’s magic formula, the relationship between the longitudinal slip and the longitudinal force was modelled for the EV and the ICE tyres. The relationship graph indicated slight variations in the EV tyre compared to the ICE tyre. The longitudinal force exerted by the EV was found to exceed the force exerted by the ICE car for the same value of the longitudinal slip, due to the additional motor mass which was marginal.

- The effects of wheel removal on the operator were examined using NIOSH equation and digital validations employing VR tools. The operator safety levels indicated by load index were rather high, but still within the acceptable range of less than 1. The VR-based digital mock-up was used for running various experiments, based on which easy wheel detachment protocol and operator safety standards for in-wheel SRM were established. The user manual was recommended to update the gripping for minimising finger entrapments.

- The battery pack included 72 battery cells. The HED100 series LiFePO$_4$ prismatic were used in series with a nominal voltage of the battery pack at 234V and the maximum consistent discharge current at 300A. A range for the EV was developed using 72 battery packs for urban and rural driving conditions. These studies resulted in an urban drive range of 225km and a highway drive range of 167km as per EPA standards.

- Finally, a specification sheet was drawn up for the developed EV. It detailed the exterior dimensions, suspension, and brake for the conceived EV. It also covered motor specifications, e.g., 15kW SRM with 5000RPM. The specification sheet also detailed the battery type and range specifications conceived in this research.
Chapter 9

Conclusions and recommendations

9.1 Chapter overview

In Chapter 9, the following aspects of this research are covered:

- Initially, Section 9.2 discusses the research approach and key conclusions from this research including details on benefits of research.
- Then, Section 9.3 discusses significant outcomes and recommendations.
- Then, Section 9.4 briefly describes on the novelties of research and outcomes.
- Then, Section 9.5 outlines the detailed evaluations, which were based on the feedback from different sources such as the industry stakeholders.
- Finally, Section 9.6 of this chapter cites research dissemination outputs such as conference papers and journal publications arising from this research.

9.2 Research approach and key conclusions

The EV industry has been growing in recent years and four main factors have influenced wide spread adoption of EV: i) environmental issues, ii) government policies, iii) commercial viability and iv) new opportunities. The key conclusions from this research are summarised below:

- In Chapter 2, two main problems were targeted i.e.: i) choice of motor type and ii) fitment within the vehicle. In general, PMMs and IMs are widely used in most of the current EV drivetrains. However, PMMs have specific disadvantage of rare earth elements usage and IMs have comparative poor inefficiencies. Hence, SRMs have been identified as a potential alternative and more sustainable option. Amongst three fitment options regarding the motor inside a vehicle, the in-wheel option was chosen due to distinct advantages such as: i) minimal transmission losses, ii) improved CG potentials, iii) cost effectiveness, and iv) enhanced ground clearance as a consequence of removing gear boxes.
- The Chapter 3 mainly focused on: i) vehicle selection for in-wheel drivetrain by comparing small and medium cars and ii) selecting an appropriate rim-tyre for in-wheel design. Accordingly, the Holden Barina Spark from the small cars was
the chosen envelope for the motor development. Designing a customised rim for
the motor envelope was done as the off the shelf rims were not suitable. In the
rim selection, five models were compared for i) stiffness, ii) thermal stability
and iii) weight considerations. The selection of tyre was based on low rolling
resistance for suitability of the in-wheel EV drivetrain. Thus, a hollow end cap
based aluminium rim and a low resistance tyre were designed for the in-wheel
motor.

- In Chapter 4, motor design was done in two stages: i) conceptual design and ii)
detailed motor design. In the first stage, two motor concepts were proposed to
achieve maximum power density inside the wheel. Based on space utilisation
consideration, the vertical mounting of magnetic path was used, which yielded
maximum power density of motor. Detailed design was done with a key
challenge to stabilise the crucial air gap of 1mm required for the magnetic path
so as to yield maximum power density. Using FE methods structural and thermal
deflections were minimised and thereby air gap variations were reduced. All the
motor parts were evaluated and the SRM design was finalised.

- The Chapter 5 focussed on mechanical optimisation of developed SRM. This
was achieved by: i) weight reduction of motor covers, ii) space utilisation and
ease of assembly and iii) an appropriate cooling design for thermal stability.
Finally in this chapter SRM key characteristics were derived.

- In Chapter 6, the main focus was on brake design and caliper selection for which
key challenges were: i) fitting the brake caliper within available space in the
vehicle, ii) optimising the weight of disc brake, and iii) complying with requisite
ADR regulations. Accordingly, conforming designs were developed and
optimised for static and structural stability of brake designs.

- The Chapter 7 mainly covered two key issues of suspension systems due to the
increase in weight at the rear of the vehicle by the addition of motors: i) impact
on the vehicle ride, and ii) safety. The impact on the vehicle ride was examined
using MATLAB® and Simscape® programs and safety study was conducted
using fatigue analysis. The vehicle ride studies showed: i) a stiff ride with
maximum 25% fluctuations when compared to the ICE, however they were
within the comfortable range of 1 to 1.5Hz and ii) a safe performance with the
safety analysis, i.e. the damage matrix of 0.00024 for life cycle of 12 years.

238
In Chapter 8, the following key issues were covered: i) the vehicle fitment, ii) the weight distribution, iii) the vehicle longitudinal slip, iv) the vehicle tyre service requirements, v) the vehicle range, and vi) the vehicle specifications. The new EV fitment was demonstrated using VR tools and range was established based on batteries used. It was concluded that the EV range values of the vehicle designed in this research (e.g., urban 225km and rural 167km) were more than the current EPA requirements. The vehicle has good stability as the variation of CG and longitudinal slips were not substantial. The VR/AR based tyre service established safe and easy detachment of wheel during servicing. Also, a vehicle specifications sheet has been established for the new EV.

9.3 Significant outcomes and recommendations

This research has outlined specific development of drivetrain and associated components for the EV. Specific recommendations from this research are:

- Because of range issues the small car is recommended as more suitable for EVs, especially with the developed motor from this research. The medium and large cars can accommodate developed EV motor technology with a compromise on range of the vehicle, hence the HEV is recommended for these segments.
- Among three motor mounting arrangements, an in-wheel motor provided advantages such as: (i) no gear train and related components, (ii) increased dynamics due to individual control of wheels, and (iii) increased ground clearance. Hence, this research recommends widespread use of the in-wheel motor for an EV application.
- Among four types of motor compared SRM and IM offer non rare earth material for magnetic path. These two motors are recommended as potentially sustainable drivetrain technologies for EV industry.
- Research on rim-tyre integration recommends use of low resistance tyres with open hub design rim. This can be set as a standalone recommendation for adaptation into new EV manufacturers for the in-wheel motor application.
- An unique light weight brake was designed to meet performance standards set by ADR. These designs were recommended to be adopted for the in-wheel motor EVs using the vehicle mass of less than 1.2tons.
A specific recommendation for EV manufacturers is to adopt higher value springs and dampers for improving ride and performance characteristics for an in-wheel motor application.

The FE methods were extensively used in this research for: i) stiffness, ii) deformation, iii) thermal and iv) fatigue studies. The rain flow count methods and life prediction techniques were specific to EV application. Hence these applications can be used in similar related automotive industry applications.

Based on experience from this research, VR and AR tools are recommended for evaluating vehicle systems, for example: (i) ergonomics evaluations for safety of an operator, (ii) assembly sequence improvisation, and (iii) design intent visualisations, by 3D digitisation and design walk through provides intermittent feedbacks among multidisciplinary teams.

MATLAB®, Simscape® programs developed were unique to the in-wheel SRM; at this point in time there are not any programs to evaluate the in-wheel motor applications. These can be recommended for adoption of an in-wheel technology for similar related automotive systems such as in-wheel buses, bikes etc.

9.4 Novelty of research

In this section, basic perspectives on the novelty aspects of this research are discussed. These include specific innovative arrangements/developments in: (i) EV design, (ii) in-wheel SRM design, (iii) structured design methodology, and (iv) VR based simulation for evaluations.

9.4.1 EV Design

With respect to EV design, this research makes an innovative contribution regarding the development of a novel SRM drivetrain and fitment of this to a specific vehicle, in this case, the Holden Barina Spark. Therefore, a range of common cars were evaluated for motor envelope and corresponding details, including: (i) small cars, and (ii) medium cars. The following are notable points regarding the novelty of the vehicle fitment arrangements in this research:

- The literature review revealed that currently there are no commercial in-wheel vehicles that have successfully adopted the SRM. Only recently (i.e. in 2012), the Mitureshi iMiev has introduced in the market its first commercial in-wheel
EV with a PMM. The mass of the in-wheel SRM is a typical issue, which is not common in conventional ICEs or EV designs with PMM drivetrains. Hence, a novel design is required to tackle this main issue of the increased mass. State of the art design concepts, methods and tools have been used for this research, which include VR, FEM, Simscape®, and MATLAB®. Reliability and performance studies were conducted for all the related mechanical components within the EV, affected due to the increased unsprung mass of an in-wheel SRM. Such examples are, the rim-tyres, brakes, and suspensions evaluated for adoption of an in-wheel SRM technology – as detailed below:

- The rim selection was based on structural rigidity for given mass of in-wheel motor. Additionally low rolling resistance tyres were examined with favourable outcome for in-wheel motor EVs. The design also used VR tools to evaluate fitment, assembly and safety issues of wheel to motor.
- The brake designs were developed and evaluated for performance consolidated from design regulations such as ADR standards.
- The suspension systems were compared with EV and ICE vehicles for analysing performance of the EV affected due to increase in mass at rear wheel of the vehicle. Also, the fatigue analysis was conducted to study the suspension designs for safety issues.

- In general, this research enabled design and fitment of an in-wheel SRM for a selected small car. Study demonstrated range based on batteries that were generic and meet or exceeds current EVs available in the markets. The specification sheet for the new EV design has been established, based on the performance of mechanical components such as brakes, suspensions, and wheel of the vehicle.
- This research outlined a holistic approach to the whole vehicle design rather than just parts of the vehicle, which can be deemed as novel in some perspectives. The designs demonstrated use of both empirical and analytical methods to evaluate different vehicle subsystems (e.g., suspensions, tyre slips).
9.4.2 In-wheel SRM

Developed novel EV drivetrain using SRM targeted maximum power density by space utilisation and minimising deflections. The novelties of motor design conceived in this research were:

- Conventional PMM designs normally use rare earth elements which are costly. Hence, the motor designed in this research used SRM technology. Ideally, the non-usage of scarce elements for the motor design will potentially provide a sustainable solution for the EV industry.

- In general, the power density of the SRMs is inferior to the PMMs of corresponding mass. However, this research has successfully developed a novel SRM design which is reasonably effective in this regard, specifically, maximum power density motor for a given mass. This was achieved in the following ways:
  - The optimisation of assembly sequencing for the motor was done by means of VR tools, which facilitated maximum space utilisation for motor within the wheel assembly.
  - The wheel was an integral part of the motor assembly rather than add on.
  - All of the mechanical components within the motor were suitably evaluated for stabilising the crucial air gap required for magnetic path, mainly the rotors and stators. This included: (i) stabilising bearing design, (ii) reducing deflections within shaft and (iii) using effective stiffness for light weight motor covers.
  - Temperature elevation during operation is a challenge. Hence, thermal deflections were minimised by the use of forced cooling and evaluating parts, which were found to be within vicinity of stators.
  - Also the increased weight is undesirable for in-wheel designs. So, all the parts were light weighted by suitable optimisation arrangements, which were evaluated for stiffness so as to reduce the effect of unsprung mass.

- The demonstrated motor used three rotors and two stators with shaft and motor covers. The rotor and stators have been packaged to allow increase in the numbers depending on the power requirement of the vehicle. This design allowed dissemination to the small vehicle and was designed as a standalone solution.
9.4.3 Structured design methodology

To date there is no established design methods available for the in-wheel motor EV. Hence, a structured design methodology has been systematically developed in this research.

- The literature and relevant standards were determined for the relevant part designs.
- The design methodology used an array of advanced tools/ techniques of CAD and FE applications. These tools also support collaborative and concurrent engineering.
- A set of special programs were developed for the design using mathematical tools such as Simscape® and MATLAB®, which enabled the analysis and evaluation of the vehicle suspension design. These programs are potentially expandable to other related vehicles.

9.4.4 VR based simulations for evaluations

Generally, the automotive design uses rapid prototypes for evaluations of the designs, which is often expensive and time consuming as well. In this research, the evaluation of designs adopted contemporary VR/AR techniques with the use of virtual prototypes. The application of VR/AR or design evaluations was cost effective and saved substantial time in the development of motor design and the vehicle fitment studies.

Some notable aspects in this regard are:

- VR based tools to evaluate the motor assembly was a novel evaluation arrangement successfully demonstrated in this research. This was achieved in the following ways:
  - The conceptual analysis based on initial schematics which used space as constrain for mounting of stators and rotors within wheel. An optimal balance between space utilisation and safe design was achieved by visualising at early stages of motor design.
  - The VR based study was done for optimising motor design sequencing to utilise maximum space within-wheel.
- The extension of the VR, AR methods used for tolerance, interference, design walkthroughs and motion capture enabled virtual validations. This process enabled faster design and substantial cost savings.
• These tools supported bi-directional associativity with top down applications such CAD, FEM, PDM and collaborative engineering technologies. Such developed methods can be adapted to other automotive design applications.

9.5 Endorsements and publications

The evaluation details through assessment of value of research results mainly include: i) positive feedback from conference/seminar audience, ii) positive feedback from demonstrations, iii) acceptance from the industry stakeholders, and iv) publications.

9.5.1 Positive feedbacks from conference and seminars

So far, the research has been disseminated in a number of scholarly journals, peer-reviewed conferences and seminars (details are in the section 9.6). Also, the research dissemination included several invited speeches (e.g., in Adelaide, Brisbane, Melbourne and Bangalore (India)), which also received encouraging feedback. These feedbacks pertained to: i) the novel approach of developing an alternative drivetrain, ii) holistic approach of the whole vehicle system design and iii) utilisation of VR/AR tools for evaluations.

9.5.2 Positive feedbacks from demonstrations

In-wheel design and optimisation was demonstrated in many seminars and conferences in Australia and overseas. Use of stereoscopic displays and prototype presentations were effective in communicating the developed technology. Pilot prototypes of an in-wheel SRM were demonstrated at Motor show 2011 Melbourne Australia, attended by more than 10,000 participants. Feedbacks from these showcases were positive and interests were shown by group of industries. The demonstrations were also held at Auto CRC 2011, Motor show 2011, Plug in for Power 2010, 2011 and Grand Prix 2012, Vic roads Beyond 2012, Melbourne, attracted many participants to take interest in this technology. The feedbacks and reports from these were encouraging and there was lot of interest from potential car manufacturers. In most of the cases recognitions were awarded for identifying possible alternative fuel technology. Some factors attributed in recognition of the in-wheel EV were:

• Innovative and novel design approach adopted for developing it as proof of concept.
• Increased space and operational activities achieved through VR tools for designs including motor, rim-tyre integrations.
• Improvements in automations, reduced weight were some highlights. Novel motor design was a potential new benchmark for EVs reducing use of rare earth elements.
• Optimisation of the vehicle system parts during fitment was recognised by many attendees when displayed using alioscopic screens.

9.5.3 Acceptance from industry stake holders

The motor that was designed is an alternative drivetrain for EV applications. The cost effectiveness arising from use of non-rare earth elements makes it more advanced drivetrain of its kind and sets a benchmark to competitors. The in-wheel technology is widely accepted by auto manufacturers. The new motor design enables it to achieve the following advantages compared to available EVs in the market:

• High power density viable motor design which can be adopted to different vehicle platforms. The vehicle architecture to suite both conversion and OEM cars
• Optimised design with analysis of all related components for performance and reliability of vehicle
• High degree of optimisation for assembly, disassembly of motor
• Improved safety by eliminating hazard by design, such an example is rim-tyre assembly
• Full detailed analysis for conversion and design of new EV including batteries, capacitors, controllers and other electronic sub systems. Complete optimised range estimation to address range anxiety issues normally faced by the vehicle manufacturers for EVs.

9.5.4 Publications

9.5.4.1 Published articles


9.5.4.2 Articles in press


• Kulkarni, A. and A. Kapoor (December 2013). Rim and tyre investigation for an in-wheel motor of the electric vehicle using simulations. 20th International Congress on Modelling and Simulation – Adapting to change: Multiple roles of modelling, Adealide, South Australia.

9.5.4.3 Submitted articles


Chapter 10

Future scope

10.1 Chapter overview

This chapter discusses future scope of this research.

10.2 Scope for further research

This research has developed an in-wheel drivetrain for a small car. In spite of optimisation and evaluation techniques demonstrated throughout the research, few areas were identified which need to be investigated for improvisation.

10.2.1 Switch reluctance motor

Typical issues with SRMs include: i) noise, ii) vibrations and iii) unsprung mass. In this research though mechanical vibrations were minimised to a large extent by using appropriate bearings, structurally robust shaft and motor covers. Further noise reduction is achieved by electromagnetic simulations and minimising the amplitude by controller design. The design and development focussed in this research is purely mechanical and the noise needs to be investigated for improvisation.

One of the major drawbacks of in-wheel motors is weight to power ratio especially using SRM. Weights associated with SRM when compared with PMM were more for the same power output. Hence further research is required to investigate use of composites if cost is not an issue. However at this point in time costs are a prime concern, hence the low cost SRM needs further research to scope out light weighting.

10.2.2 Vehicle performance

The rim design used Aluminium material, however further mass improvisation can be achieved with composite wheel with a compromise on the cost. Tyres were analysed for longitudinal slip for dry conditions. Further study on hydroplaning of the tyres is essential.

The new developed brake needs to be incorporated with existing regeneration within Holden Barina Spark. Also, through optimised controller design the motor regeneration can be effectively used. These two areas need to be investigated further.
The suspension analysis established variation 20-25% in ride comfort. Further analysis can be simulated by incorporating new MacPherson Strut and simulating these studies for the improved dampening.

10.2.3 Adoption of SRM in other vehicle sectors
The aim of future research is to implement this motor into the other sectors mainly, heavy vehicles, bikes, rickshaws (three wheelers) and golf cart. The adoption of these vehicle types with minimal modification using modular SRM technology needs to be investigated. It would be useful to investigate other vehicles for similar design re-evaluations for electrification initially and the full vehicle development in later stages.

10.2.4 Use of styling creations for improving vehicle range
The styling of an EV plays vital role in improving the range, as it results in reduced drag force. Three dimensional digitisation and surfacing techniques needs to be investigated for the improved styling on EVs. This styling of surfaces also advances the vehicle range and performance by reducing the air drag. These techniques can effectively increase the range of an EV.

10.2.5 Enhancement of VR applications for auto industry
VR tools are effective in communicating and visualising the vehicle design concepts. It would be useful to investigate further usage of these tools for automotive design validation and re-evaluations. Automation of assemblies and concept proposals can be further investigated for developing training for electrification of vehicles. These tools could be used for scoping future research agendas. Such an example is demonstration of E-city or mobility solutions concept using VR tools for EVs and sustainable transport modes.

10.2.6 Improvement to design methodology
The research focused on design and development of a novel motor design for an EV. The methodology used different CAD, FE, MATLAB®, Simscape®, VR and AR tools for effective design and analysis of automotive subsystems. This advanced tool with advancement in high performance computing techniques allows faster design cycles with an improved quality. All these tools can be implemented in collaborative environment for an effective communications with design teams. This will enhance
large data sharing resulting in faster deployment of results. This will further advance the collaborative product development. All these tools could offer future scope to develop an innovative automated design methodology.
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Appendices

This chapter lists appendices used for all chapters in this research. The appendices include drawings, MATLAB® programs, Simscape® models and script files for auto stereoscopic displays. The soft copy for MATLAB® program, Simscape® models and script files are attached in CD. Appropriate software is required to open these files.

Appendix 1 EV conventional and by-wheel drivetrains

In the middle of 2007, GM announced the manufacture of the GM Volt. The GM Volt has been classed as a compact five-door hatchback with a 1.4litre, 4 cylinder engine and has two electric motors/controllers of 111kW (Appendix Figure 1.1) and 55kW (Lyle 2010, Khwaja, Mohammad et al. 2011). The Volt led a new era for electric automobiles by creating this new class of vehicles known as the Extended-Range EV (E-REV). The 2011 GM Volt E-Rev is a front-wheel-drive (conventional configuration) and four-passenger HEV that uses electricity as primary power source through batteries. The GM-Volt is extending range by utilising the smaller motor to function as a generator, driven by 1.4 liter petrol engine to charge the battery. The EPA range for GM Volt E-REV is 56km (city)/ 65km (highway) in EV mode with an extension of ~611km using combined mode (EV and petrol). GM estimates that, compared to petrol powered vehicles that have a fuel consumption of 7.8L/100km (calculated used ADR81/01 fuel consumption figures), the Volt has a proclaimed saving of about 1,892litres of petrol based on 64km of daily driving and 24,000km annually (GM 2012). The battery cells are T-shaped to achieve appropriate packaging inside the vehicle. The battery constitutes 288 cells, weighs 190kg and is able to supply a total power of 16kWh. It has a wheelbase of 2.68M with exterior dimensions of 4.5M length, 1.44M height and 1.8M width, and weighs 1715kg.
GM in 2012 announced the launch of EV called as GM Spark electric. This car uses 96kW motor and houses 20kWh batteries to produce a torque of 540Nm and 132km range (based on EPA results as claimed by GM).

Mitsubishi launched an i-MiEV (Mitsubishi innovative electric vehicle) concept EV in 2005 Tokyo Motor Show. It used Mitsubishi Colt platform with two 20kW rear in-wheel motors. Since then company has developed a series of concept cars, including an 8 seater EV. The current i-MiEV is a 5-door hatchback which uses a conventional drivetrain configuration with a 47kW 330V AC synchronous PMM using single speed reduction gear (Appendix Figure 1.2), producing a torque of 180Nm (Mitsubishi Motors 2008). A high capacity battery module (16kWh with 22 cells) is located under the unmodified floor panel, and individual battery cells are mounted both vertically and transversely. It has a wheelbase of 2.55M, with a 3.4M length, 1.6M height and 1.48M width, and weights 1080kg.

Appendix Figure 1.2: Mitsubishi i-MiEV motor, Source: (Mitsubishi Motors 2008)
The Nissan LEAF is a five door hatchback with a five passenger Japanese manufactured car. It is a front-wheel 280Nm torque, 80kW motor, with a conventional drivetrain (Nissan 2013). It has 48 LiFePO$_4$ battery cells weighing 218kg rated 24kWh, equivalent to energy density of 140Wh/kg. With a wheel base of 2.7M and exterior dimensions of 4.45M length, 1.77M width, 1.55M height and weights 1520kg. Newly launched 2013 Nissan LEAF is 32kg lighter than the original.

Toyota has been a world leader, with nearly 25% of the automotive market share (BBC News 2007, Mira Oberman 2013). With the launch of Prius in late 1995, Toyota entered the HEV market at early stage. In, Prius was made available in 2004, initially as a hatch back. It is now available in two additional variations: Prius-C (offered as a small compact size car seating 5), and Prius-V (offered as a people mover with 7 seats). Prius hatchback 2010, uses 5.3kWh batteries and a four-cylinder 1.8L petrol engine with 73kW/60kW PMMs (Toyota Motors 2011). It has a wheel base of 2.8M and exterior dimensions of 4.46M length, 1.75M width, 1.48M height and weighs 1376kg. Amongst several concepts Toyota is working, Toyota Fine is a FCEV concept uses four in-wheel motors with a fuel cell (FC). In addition to this, the second generation of Rav4 electric was launched in 2012, which uses Tesla 115kW motor in a Toyota body. Lexus electric is another HEV offered from Toyota.

Ford has recently launched Focus Electric, a five seater family sedan. It uses a 106kW electric motor with single-speed transmission (Appendix Figure 1.3). The Focus Electric is powered by a 23kWh high-voltage, light weight LiFePO$_4$ batteries (Ford Motor Company 2010, Ford Motor Company 2012). It has a wheel base of 2.65M and exterior dimensions of 4.40M length, 1.82M width, and 1.48M height and weighs 1674Kg. It also uses an advanced active liquid cooling and heating process to regulate the battery temperature (to maximise battery life). Another feature added is the use of a 100% recyclable seating systems. Prior to this, there was a Ford F-150, Hi-pa concept car using 112kW motor in each wheel (AOL Autos Inc. October, 2008). The key features include, regenerative braking (with a brake resistor) down to very low speed and full holding torque at zero speed.
Appendix Figure 1.3: Ford Focus electric motor, Source: (Ford Motor Company 2010)

Appendix Figure 1.4: Honda FCEV motor, Source: (American Honda Motor Company Inc. 2012)

The Honda FCX concept car uses an EV motor with a FC, as shown in Appendix Figure 1.4. The FC combines hydrogen with oxygen to make electricity. The electricity powers an electric motor, which in turn propels the vehicle. As claimed by Honda, water is the only by-product the FCEV leaves behind. It uses an 100kW electric motor for the drivetrain, its cell stack generates electricity from hydrogen fuel, and LiFePO$_4$ batteries are used to store the energy (American Honda Motor Company Inc. 2012). This motor produces torque about 256Nm with 3056rpm and range of 386km (according EPA as claimed by Honda). It has a wheel base of 2.80M and exterior dimensions of 4.83M length, 1.85M of width, and 1.47M of height and weighs 1625kg.

The Audi R8-based e-tron is an all-electric concept car that was unveiled at the 2009 Frankfurt Motor Show. It has an 230kW electric motor with an in-wheel drivetrain configuration (Bary 2009). The current Audi A1e-tron uses 45kW (continuous) and 75kW (peak) motor at front axle and drives 50km (with 96 prismatic cells less than150kg) (Audi 2013). The Blade Electric Vehicle and the Eday Life are two
Australian owned companies. Blade Electron Mk VI used Hyundai car as a platform with a 55kW PMM, 320V, 21kWh capacity LiFePO$_4$ battery pack. As claimed by company, the Blade Electron Mk VI, has a top speed of up to 120kmh and 0-60km in 4.7 seconds (Blade Electron May 2011).

Current Asian EV manufacturing market is gradually growing. Such EV examples are Chinese owned Chery S18/ Zhejiang Jonway Automobile Co., Ltd (ZAP) A380 and Indian owned Mahindra Reva NXG. The Chery S18, uses a 336V, 40kW drivetrain and 40Ah LiFePO$_4$ batteries (Green Car Congress February 2009). ZAP has been a leading distributor of fuel-efficient alternative energy vehicles in the USA and, according to the Electric Auto Association, there are over 56,000 ZAP EVs on the road today. ZAP introduced A380, is a five seat SUV uses a 82kW motor and 36kWh batteries producing a range of 160km (as claimed by company) (ZAP Jonway 2009). The Mahindra Reva NXG (Called e2o) from India, features 72 volt LiFePO$_4$ batteries and uses a 25kW Ac IM (Blog at world press July, 2010).

SIM drive a Japanese company with Hiroshi Shimizu from Keio University, Tokyo showcased EV using by-wheel drivetrain. It typically has eight wheels using 60kW PMM with a high speed of 370km/h (SIM drive 2012). The motor is placed by the wheel in this car as shown in Appendix Figure 1.5.

*Appendix Figure 1.5: SIM drive, Source: (SIM drive 2012)*
Appendix 2 Scanning with Artec 3D scanner

The scanner had a flash bulb that projected light pattern onto the object, which was recorded by the camera. The distortion in the light pattern due to the specific curvature of the object was then translated into a 3D image by Artec software. As the scanner moved around the object, the light pattern changed and the software recognised these changes. The light pattern was projected onto the Holden Barina Spark surface up to 15 times per second. Maximum flash intensity during the peak of the flash was 165mW/cm\(^2\), when measured 640mm from the scanner. Average flash intensity over time did not exceed 0.5mW/cm\(^2\). The flash operated for 0.2ms in 66.4ms intervals.
Appendix 3 Bezier curves and non-uniform rational B-spline (NURBS) mathematical methods

During late 1960s Pierre Bezier in Renault and Paul de Casteljau in Citroen initially developed a Bezier curve representation and extended it to a surface patch methodology. Bezier curves and non-uniform rational B-spline (NURBS) mathematical methods were used to refine these surfaces, as shown in Equations 3.1 to 3.15, where N was order, \(x_3\) and \(y_3\) are end points on \(x_0\) and \(y_0\) origin, and \(x_1, y_1, x_2,\) and \(y_2\) are control points. An increasing value of time interval, \(t\) was supplied to equations to obtain the required fine-tuned surface. The \(a, b, c,\) are numerical constants of defined control points. The quadratic equation given by Equation 3.1, for three degrees of freedom was derived, where values of \(x_1, x_2, x_3\) were given by Equations 3.2 to 3.4:

\[
x(t) = a_x t^3 + b_x t^2 + c_x t + x_0
\]

(3.1)

\[
x_1 = x_0 + \frac{c_x}{3}
\]

(3.2)

\[
x_2 = x_1 + \frac{c_x + b_x}{3}
\]

(3.3)

\[
x_3 = x_0 + a_x + b_x + c_x
\]

(3.4)

Calculating \(y\) at \((t)\) interval by Equation 3.5:

\[
y(t) = a_y t^3 + b_y t^2 + c_y t + y_0
\]

(3.5)

Replacing value of \(x\) with \(y\) in Equations 3.2 to 3.4 we derive values of \(y_1, y_2,\) and \(y_3\) from Equations 3.6 to 3.8:

\[
y_1 = y_0 + \frac{c_y}{3}
\]

(3.6)

\[
y_2 = y_1 + \frac{c_y + b_y}{3}
\]

(3.7)

\[
y_3 = y_0 + a_y + b_y + c_y
\]

(3.8)

Numerical constants of defined control points at \(x\) and \(y\) locations are given by Equations 3.9 to 3.14:

\[
c_x = 3(x_1 - x_0)
\]

(3.9)
The final value of $N$ at positions $(x, y)$ was calculated using the expression given in Equation 3.15:

$$N(x, y) = \sum_{x_0=0}^{n} \binom{n}{x_0} (1-x)^{n-x_0} x_0 \sum_{y_0=0}^{n} \binom{n}{y_0} (1-y)^{n-y_0} y_0 p_{x_0,y_0}$$  (3.15)
Appendix 4 Finite Element results for rim optimisation

**Rim 1**- Maximum Von-mises stress, maximum deformation, maximum strain, and fatigue life cycles

*Appendix Figure 4.1: Rim 1 maximum Von-mises stress concentration*

*Appendix Figure 4.2: Rim 1 maximum deformation*
Appendix Figure 4.3: Rim 1 maximum strain

Appendix Figure 4.4: Rim 1 lifecycles
**Rim 2**- Maximum Von-mises stress, maximum deformation, maximum strain, and fatigue life cycles

**Appendix Figure 4.5:** Rim2 maximum Von-mises stress concentration

**Appendix Figure 4.6:** Rim2 maximum deformation
Appendix Figure 4.7: Rim2 maximum strain

Appendix Figure 4.8: Rim2 lifecycles
**Rim 3-** Maximum strain

*Appendix Figure 4.9: Rim 3 maximum strain*

**Rim 4-** Maximum Von-mises stress, maximum deformation, maximum strain, and fatigue life cycles

*Appendix Figure 4.10: Rim 4 maximum Von-mises stress concentration*
Appendix Figure 4.11: Rim4 maximum deformation

Appendix Figure 4.12: Rim4 maximum strain
Appendix Figure 4.13: Rim4 life cycles (predicted failure area enlarged)

**Rim 5**- Maximum Von-mises stress, maximum deformation, maximum strain, and fatigue life cycles

Appendix Figure 4.14: Rim5 maximum Von-mises stress concentration
Appendix Figure 4.15: Rim5 maximum deformation

Appendix Figure 4.16: Rim5 maximum strain
Appendix Figure 4.17: Rim5 life cycles (predicted failure area enlarged)
### Appendix 5 Drawing documentation

In this section only main drawings, drawing master file are attached for reference.

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Appendix 6 Bearing SKF data sheet

### Appendix Figure 6.1: Bearing life factor for ball and roller bearings (Source: SKF data sheet)

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<th>fₚ</th>
<th>L₁₀⁻¹</th>
<th>fₚ</th>
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<th>fₚ</th>
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Ball bearings

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Roller bearings
Appendix 7 Shaft calculations

The following dimensions were taken from the car: i) shaft length, $L$ (235 mm), ii) load at the end of the shaft, $P$ (5kN), iii) EN 26 steel’s Young’s modulus, $E$ (210GPa), iv) distance from neutral axis to extreme ends, $c$ (22.5 mm). From the shaft design, moment of inertia was calculated as per Equation 7.1:

$$I = \pi \left( D^4 - d^4 \right) / 32 \quad (7.1)$$

$$\pi(0.04^4 - 0.030^4)/32 = 1.5 \times 10^{-7} \text{m}^4$$, hence displacement was given by Equation 7.2:

$$w(x) = \frac{Px^2(3L-x)}{6EI} \quad (7.2)$$

Thus, maximum deflection was calculated using Equation 7.3:

$$w_{\text{max}} = w(L) = -\frac{PL^3}{3EI} \quad (7.3)$$

Therefore, $w_{\text{max}}$ was -0.721 mm and shaft slope was calculated using Equation 7.4 below:

$$\theta(x) = -\frac{P(2L-x)x}{2EI} \quad (7.4)$$

and the maximum slope was calculated based on Equation 7.5:

$$\theta_{\text{max}} = \theta(L) = -\frac{PL^2}{2EI} \quad (7.5)$$

Therefore, $\theta_{\text{max}}$ was calculated as -0.00368 degrees and moment on the shaft was calculated using Equation 7.6:

$$M(x) = P(L-x) \quad (7.6)$$

Maximum moment was calculated when $x = 0$, therefore it was 1175Nm. Maximum bending stress in shaft was 176.25MPa calculated from Equation 7.7, as follows:

$$\sigma_{\text{max}} = \left| M_{\text{max}} \right| \frac{c}{I} \quad (7.7)$$

Therefore it was, and shear at the shaft was calculated using Equation 7.8 as $-5KN$

$$V(x) = -P \quad (7.8)$$
Appendix 8 Auto stereoscopic script file

To be used with i and play assistant and JPEG’s are not attached due to size restrictions.
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splash = no

[input]
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last = 100
loop = 1
fade = yes

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loop = 1
fade = yes

$output$
type = image/alm
mode = 8mixed
file = "C:\Users\ambarishkulkarni\Desktop\FinalRenders\alm\GP_<F4>.alm"
Appendix 9 Thermal expansion of Aluminium bushes

Values marked red in figures indicate the expanded dimensions after heat exposure.

*Appendix Figure 9.1: Thermal expansion of the bush inside motor*

*Appendix Figure 9.2: Thermal expansion of the bush inside motor*
Appendix 10 Motor characteristics

Values marked yellow in the appendix table indicate the maximum values considered

**Appendix Table 10.1: Motor characteristic derivations**

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<th>vehicle speed (km/h)</th>
<th>vehicle speed (m/s)</th>
<th>Angular velocity (rad/s)</th>
<th>Rolling resistance (N)</th>
<th>Gradient resistance (N)</th>
<th>Aerodynamic resistance (N)</th>
<th>Tractive effort without $F_G$ (N)</th>
<th>Normal Power Needed (kW)</th>
<th>Total tractive effort (N)</th>
<th>power needed with gradient (kW)</th>
<th>Torque without $F_G$ (Nm)</th>
<th>Torque with $F_G$ (Nm)</th>
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Appendix 11 Drum brakes

**Appendix Figure 11.1:** Drum brake mechanisms. a) Simplex, b) Duplex, c) Duo Duplex, d) Servo, e) Duo Servo (Breuer and Karlheinz 2008)

**Appendix Figure 11.2:** Drum brake materials. 1) Grey cast iron, 2) Aluminium casting, cast iron insert, and 3) Aluminium/Ceramic casting (Breuer and Karlheinz 2008)
Appendix 12 Disc brakes and calipers

Appendix Figure 12.1: Fixed caliper: 1) brake rotor, 2) hydraulic connection, 3) brake piston, and 4) bleed screws (Breuer and Karlheinz 2008)

Appendix Figure 12.2: Frame caliper: 1) brake rotor, 2) brake piston, 3) hydraulic connection, 4) bleed screw, 5) mounting, and 6) frame (Breuer and Karlheinz 2008)

Appendix Figure 12.3: Fist caliper: 1) brake rotor; 2) brake piston, 3) hydraulic connection, 4) bushings, 5) mounting, and 6) frame (Breuer and Karlheinz 2008)
Appendix 13 Brake Testa Millennium

Appendix Figure 13.1: Brake Testa Millennium, a) testing device (left) and b) load sensor on foot brake pedal (right)

Appendix Figure 13.2: Holden Barina Spark -Brake test data at 60-0km/h

Appendix Figure 13.3: Holden Barina Spark- Brake test data at 100-0km/h
Appendix 14 MATLAB® program- ICE

MATLAB® Program for quarter Car Analysis of the suspension system on an ICE vehicle.

Mds=90; %Seat/Driver mass
Ms=110; %Sprung mass
Mu=40; %Unsprung mass
Cs=2969; %Sprung mass damping coefficient
Cds=8372; %Seat/Driver mass damping coefficient
Kt=125000; %Spring coefficient Unsprung mass
Ks=44070; %Spring coefficient Sprung mass
Kds=176280; %Spring coefficient seat/driver mass
Xr=1.0; %Road input

% Coefficient of A
A=[0,1,0,0,0,0;-Kds/Mds,-Cds/Mds,Kds/Mds,Cds/Mds,0,0;0,0,0,0,0,1;Kds/Ms,Cds/Ms,-Kds/Ms-Ks/Ms,-Cds/Ms-Cs/Ms,Ks/Ms,Cs/Ms;0,0,0,0,0,1;0,0,Ks/Mu, Cs/Mu,-Ks/Mu-Kt/Mu,-Cs/Mu];

% Coefficient of B
B=[0;0;0;0;0;Kt/Mu];

% Displacement analysis
C1=[1,0,0,0,0,0];
C2=[0,0,1,0,0,0];
C3=[0,0,0,0,1,0];

% Velocity analysis
v1=[0,1,0,0,0,0];
v2=[0,0,0,1,0,0];
v3=[0,0,0,0,0,1];

% Coefficient of D
D=[0];

% Change in displacement
dds = ss (A, Xr*B, C1, D);
ds = ss (A, Xr*B, C2, D);
du = ss (A, Xr*B,C3,D);

% Change in velocity
vds = ss(A, Zr *B,v1,D);
vs = ss(A, Zr *B,v2,D);
vu = ss(A, Zr *B,v3,D);
% Driver/Seat response
figure;
step (dds,vds)
axis ([0 10 -1 1.5])
legend ('Displacement','velocity');
title ('Velocity and displacement response of seat of an ICE')

% Spung Mass response
figure;
step (ds,vs)
axis ([0 10 -1 2])
legend ('Displacement','velocity');
title ('Velocity and displacement response of Spung Mass of an ICE')

% Unspung Mass response
figure;
step (du,vu)
axis ([0 1 2 7])
legend ('Displacement','velocity');
title ('Velocity and displacement response of UnSpung Mass of an ICE')
Appendix 15 MATLAB® program-EV

MATLAB® program for quarter Car Analysis of the suspension system on an EV.

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Ms=130;                     %Sprung mass
Mu=80;                     %Unsprung mass
Cs=2969;                  %Sprung mass damping coefficient
Cds=8372;                           %Seat/Driver mass damping coefficient
Kt=125000;               %Spring coefficient Unsprung mass
Ks=44070;                           %Spring coefficient Sprung mass
Kds=176280;             %Spring coefficient seat/driver mass
Xr=1.0;                                %Road input

% Coefficient of A
A=[0,1,0,0,0,0;-Kds/Mds,-Cds/Mds,Kds/Mds,Cds/Mds,0,0;0,0,0,1,0,0;Kds/Ms,Cds/Ms,-Kds/Ms-Ks/Ms,-Cds/Ms-Cs/Ms,Ks/Ms,Cs/Ms;0,0,0,0,0,1;0,0,Ks/Mu, Cs/Mu,-Ks/Mu-Kt/Mu,-Cs/Mu];

%Coefficient of B
B=[0;0;0;0;0; Kt/Mu];

% Displacement analysis
C1=[1,0,0,0,0,0];
C2=[0,0,1,0,0,0];
C3=[0,0,0,0,1,0];

% Velocity analysis
v1=[0,1,0,0,0,0];
v2=[0,0,0,1,0,0];
v3=[0,0,0,0,0,1];

% Coefficient of D
D=[0];

% Change in displacement
dds= ss (A, Xr*B, C1, D);
ds= ss (A, Xr*B, C2, D);
du = ss (A, Xr*B,C3,D);

% Change in velocity
vds= ss(A, Zr *B,v1,D);
vs= ss(A, Zr *B,v2,D);
vu= ss(A, Zr *B,v3,D);
% Driver/Seat response
figure;
step (dds,vds)
axis ([0 10 -1 1.5])
legend ('Displacement','velocity');
title ('Velocity and displacement response of seat of an EV)

% Spung Mass response
figure;
step (ds,vs)
axis ([0 10 -1 2])
legend ('Displacement','velocity');
title ('Velocity and displacement response of Spung Mass of an EV')

% Unspung Mass response
figure;
step (du,vu)
axis ([0 1 -2 7])
legend ('Displacement','velocity');
title ('Velocity and displacement response of Unspung Mass of an EV')
Appendix 16 Simscape® model and results

Appendix Figure 16.1: Simscape® model used in this research

Appendix Figure 16.2: Velocity and displacement of sprung mass (EV)
Appendix Figure 16.3: Velocity and displacement of unsprung mass (EV)

Appendix Figure 16.4: Velocity and displacement of driver-seat mass (EV)