Electronic stability control for electric vehicle with 4 in-wheel electric motor

Muhammad Mehedi Al Emran, Hasan
Electronic stability control for electric vehicle with 4
in-wheel electric motor

Research Thesis

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Requirements for the
Degree of Doctor of Philosophy

Muhammad Mehedi Al Emran, Hasan

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The research thesis was done under the supervision of
Dr. Mehran Motamed Ektesabi and Prof. Ajay Kapoor
in the Faculty of Engineering and Industrial Sciences
DECLARATION

I hereby declare that I am the sole author of this thesis. To the best of my knowledge, it contains no material that has been published by others previously except where necessary references have been mentioned. No material of this thesis work has been submitted or accepted for any other degree of diploma at any university and this thesis discloses the relative contributions in case of joint research.

Name: Muhammad Mehedi Al Emran Hasan

Signature:                          Date:

..................................................  ..............................................
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# Table of contents

1 Introduction .................................................................................................................. 8
   1.1 Research background and motivation .............................................................. 8
   1.2 Objectives ....................................................................................................... 9
   1.3 Approaches ................................................................................................. 10
   1.4 Dissertation organisation ...................................................................... 10

2 Pollution and the advanced electric vehicle .......................................................... 13
   2.1 Introduction ................................................................................................. 13
   2.2 Transport systems and their effect .............................................................. 14
   2.3 Pollution control and viable transport by EV .......................................... 15
   2.4 Conventional concept of EV .................................................................. 16
   2.5 Different drive train in EV ....................................................................... 17
   2.6 In-wheel motor technology ..................................................................... 20
       2.6.1 Technical feasibility of using in-wheel motor .................................... 23
   2.7 Concept of four in-wheel EV .................................................................... 25
   2.8 Challenges of using four in-wheel motors EV ........................................ 26
   2.9 Four in-wheel motors EV development for research .............................. 27
   2.10 Summary .................................................................................................. 28

3 Electronic stability control .................................................................................... 31
   3.1 Introduction .................................................................................................. 31
   3.2 Concept of stabilizing using ESC ................................................................. 32
   3.3 Effectiveness of ESC ............................................................................... 35
       3.3.1 Effectiveness from crash data analysis .............................................. 36
       3.3.2 Human factors analysis using a simulator ......................................... 37
   3.4 Different methods of ESC ......................................................................... 37
       3.4.1 Differential braking type ESC ......................................................... 39
       3.4.2 Steer by wire control method ............................................................ 40
       3.4.3 Differential torque distribution type ESC ...................................... 42
   3.5 Comparison of different types of ESC focusing on four in-wheel EV ... 44
       3.5.1 Required sensor ............................................................................... 44
       3.5.2 Required actuators .......................................................................... 44
       3.5.3 Complexity and performance ............................................................ 45
       3.5.4 Proposed ESC type ........................................................................ 45
   3.6 ESC regulations ............................................................................................ 46
# Table of contents

3.7 Summary .................................................................................................................. 50

4 Investigation into differential torque based ESC systems and proposed method .......................................................... 52

4.1 Introduction ............................................................................................................. 52

4.2 Related parameters and estimations ........................................................................ 53

4.2.1 Vehicle dynamics ...................................................................................... 54

4.2.2 Tire force ...................................................................................................... 59

4.2.3 Control law parameters estimations .............................................................. 63

4.2.3.1 Yaw rate estimation .............................................................................. 64

4.2.3.2 Vehicle Side Slip angle ....................................................................... 66

4.2.3.3 Coefficient of friction estimation .......................................................... 69

4.3 Control law and Strategies .................................................................................. 73

4.3.1 Previously used strategies ............................................................................ 73

4.4 Wheel torque control technique .......................................................................... 77

4.4.1 Mechanical torque calculation of wheel......................................................... 80

4.4.2 Proposed wheel speed control technique ...................................................... 82

4.4.3 Wheel electrical torque calculation ............................................................... 83

4.5 Overall discussion and scope of current research ............................................. 83

4.6 Proposed vehicle stability control method ......................................................... 87

4.6.1 Overview of the method .............................................................................. 87

4.6.2 Controller structure ..................................................................................... 88

4.6.3 Top level controller ..................................................................................... 88

4.6.3.1 Top level input and output .................................................................... 89

4.6.3.2 Required longitudinal force calculation ................................................. 90

4.6.4 Lower level controller .................................................................................. 91

4.6.4.1 Lower level input and output ................................................................. 91

4.6.4.2 Wheel speed control interface ............................................................... 92

4.6.5 Proposed method summary ......................................................................... 94

4.7 Summary ............................................................................................................... 94

5 Modelling the components of four in-wheel EV for ESC .................................. 97

5.1 Introduction ......................................................................................................... 97
Table of contents

5.2 Requirement of Vehicle modelling ................................................................. 97
5.3 Overview on the proposed simulation .............................................................. 99
  5.3.1 Vehicle body modelling ............................................................................. 101
  5.3.2 Wheel modelling ....................................................................................... 105
  5.3.3 Desired values model block for yaw rate and slip Angle ......................... 108
5.4 Modelling Controllers for ESC in 4 in-wheel EV .......................................... 113
  5.4.1 Sliding mode controller law ................................................................. 113
  5.4.2 Modelling the controller blocks ............................................................. 116
  5.4.2.1 Modelling control law ....................................................................... 116
  5.4.2.2 Longitudinal force difference calculation: ........................................ 117
  5.4.2.3 Wheel torque calculation ................................................................. 120
5.5 Simulation ..................................................................................................... 124
  5.5.1 Vehicle simulation for longitudinal motion ....................................... 125
  5.5.2 Vehicle Simulation for Stability ............................................................. 130
    5.5.2.1 Open loop simulation ...................................................................... 133
    5.5.2.2 Closed loop simulation ................................................................... 137
5.6 Performance of the proposed system .............................................................. 141
5.7 Summary ....................................................................................................... 145

6 Hardware setup and experiments ..................................................................... 148
  6.1 Introduction .................................................................................................. 148
  6.2 Proposed hardware system for experiment .............................................. 148
  6.3 Measurement of experimental platform ...................................................... 150
    6.3.1 In-wheel test Rig ................................................................................ 151
    6.3.2 Observation of the throttle pedal signal to a desired wheel velocity ........ 153
  6.4 Digital Controller ........................................................................................ 154
    6.4.1 Embedded host with I/O and programming environment ................... 154
    6.4.2 Feed-back controller for wheel ........................................................... 156
    6.4.2.1 Requirement of a controller ........................................................... 156
    6.4.2.2 Challenges ...................................................................................... 156
    6.4.2.3 Control method ............................................................................. 157
Table of contents

6.4.2.4 In-wheel motor and driver ................................................... 158
6.4.2.5 Filtering motor feedback signal ........................................... 159
6.4.2.6 Programming the PID controller ......................................... 160
6.4.3 Results and discussion ............................................................ 170
6.5 Summary ............................................................................................... 178

7 Advanced features of ESC ............................................................................. 181
7.1 Centre of gravity and its effect on electronic stability control .............. 181
   7.1.1 Vehicle model and analysis .................................................... 184
   7.1.2 Simulation result of vehicle with centre of gravity shifting ... 185
   7.1.3 Summary ................................................................................ 188
7.2 Effect of CoG in vehicle mass distribution ........................................... 189
   7.2.1 Mass Distribution and the centre of gravity ...............190
   7.2.2 Calculation of CoG in case of converted EVs: ............... 190
   7.2.3 Simulation .............................................................................. 191
   7.2.4 Results and Analysis .............................................................. 192
   7.2.5 Summary: ............................................................................... 194

8 Conclusions and future work ........................................................................ 196
8.1 Objectives .............................................................................................. 196
8.2 Achievements ........................................................................................ 196
8.3 Contributions ......................................................................................... 199
8.4 Suggestions for further research ............................................................ 202

Reference ................................................................................................................. 204
**List of figures**

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>Passenger vehicles increase in Australia from 2008 to 2013</td>
<td>8</td>
</tr>
<tr>
<td>1.2</td>
<td>Greenhouse gas emissions in 2007 in Australia</td>
<td>14</td>
</tr>
<tr>
<td>1.3</td>
<td>Overall efficiency of different types of vehicle</td>
<td>15</td>
</tr>
<tr>
<td>1.4</td>
<td>Basic concept of a conventional Electric Vehicle</td>
<td>17</td>
</tr>
<tr>
<td>1.5</td>
<td>Indirect drive in-wheel system</td>
<td>22</td>
</tr>
<tr>
<td>1.6</td>
<td>Direct drive in-wheel system</td>
<td>22</td>
</tr>
<tr>
<td>1.7</td>
<td>Exploded view of used in-wheel motor</td>
<td>23</td>
</tr>
<tr>
<td>1.8</td>
<td>Conceptual chassis layout of conventional ICEV, EV and modern in-wheel EV</td>
<td>24</td>
</tr>
<tr>
<td>1.9</td>
<td>Four in-wheel motor EV architecture</td>
<td>26</td>
</tr>
<tr>
<td>1.10</td>
<td>Four in-wheels EV used in this study</td>
<td>28</td>
</tr>
<tr>
<td>1.11</td>
<td>Yaw control functionality</td>
<td>33</td>
</tr>
<tr>
<td>1.12</td>
<td>Yaw moments for different steering angle [12]</td>
<td>34</td>
</tr>
<tr>
<td>1.13</td>
<td>Friction circle concept</td>
<td>38</td>
</tr>
<tr>
<td>1.14</td>
<td>Differential braking ESC manoeuver (a) over-steering (b) under-steering</td>
<td>39</td>
</tr>
<tr>
<td>1.15</td>
<td>(a) Steer-by-wire steering system (b) Steer-by-wire ESC</td>
<td>41</td>
</tr>
<tr>
<td>1.16</td>
<td>Torque vectoring based ESC manoeuver (a) over-steering (b) under-steering</td>
<td>43</td>
</tr>
<tr>
<td>1.17</td>
<td>Sine dwell steering test (a) steering angle profile and lateral displacement (b) steering angle and yaw rate for lateral stability</td>
<td>48</td>
</tr>
<tr>
<td>1.18</td>
<td>Fishhook manoeuver description</td>
<td>49</td>
</tr>
<tr>
<td>1.19</td>
<td>Lane change track setup</td>
<td>50</td>
</tr>
<tr>
<td>1.20</td>
<td>Simplified vehicle model for analysis of transient motions</td>
<td>56</td>
</tr>
<tr>
<td>1.21</td>
<td>Tire forces and rotation</td>
<td>60</td>
</tr>
<tr>
<td>1.22</td>
<td>Longitudinal force versus tire slip ratio using Magic Formula tire model [13]</td>
<td>70</td>
</tr>
<tr>
<td>1.23</td>
<td>Control structure of case 1</td>
<td>74</td>
</tr>
<tr>
<td>1.24</td>
<td>Control structure of case 2</td>
<td>75</td>
</tr>
<tr>
<td>1.25</td>
<td>Control structure of case 3</td>
<td>76</td>
</tr>
<tr>
<td>1.26</td>
<td>Control structure of case 4</td>
<td>76</td>
</tr>
<tr>
<td>1.27</td>
<td>Wheel rotational dynamics under the influence of torque</td>
<td>81</td>
</tr>
<tr>
<td>1.28</td>
<td>An example setup of in-wheel motor with driver and throttle</td>
<td>82</td>
</tr>
</tbody>
</table>
List of figures

Figure 4.10 Overall coverage on differential torque based ESC done in previous works ......................................................................................................................................................................................... 85
Figure 4.11 Top level of the proposed controller .................................................................................................................................................................................................................................................. 88
Figure 4.12 Proposed control strategy (partial) using yaw rate and vehicle slip angle .................................................................................................................................................................................................................. 90
Figure 4.13 Lower level of the proposed controller .................................................................................................................................................................................................................................................. 91
Figure 4.14 Proposed wheel speed controller .................................................................................................................................................................................................................................................................................................. 92
Figure 4.15 Proposed method of stability control with the overall system .................................................................................................................................................................................................................... 93
Figure 5.1 All components of proposed simulation for ESC .................................................................................................................................................................................................................................................. 100
Figure 5.2 Forces acting on the vehicle ........................................................................................................................................................................................................................................................................................................ 102
Figure 5.3 Illustration of vehicle body subsystem block .......................................................................................................................................................................................................................................... 103
Figure 5.4 Front wheel subsystem model .................................................................................................................................................................................................................................................................................................. 106
Figure 5.5 Rear wheel subsystem model .......................................................................................................................................................................................................................................................................................... 106
Figure 5.6 Steady state steering angle calculation block .............................................................................................................................................................................................................................................. 109
Figure 5.7 Targeted yaw rate calculation ......................................................................................................................................................................................................................................................................................... 111
Figure 5.8 Targeted slip angle calculation ................................................................................................................................................................................................................................................................................... 113
Figure 5.9 Control law block ...................................................................................................................................................................................................................................................................................................................... 117
Figure 5.10 Longitudinal wheel force calculation block ........................................................................................................................................................................................................................................ 118
Figure 5.11 Differential wheel torque calculation block ......................................................................................................................................................................................................................................... 121
Figure 5.12 Differential braking torque Calculation block ....................................................................................................................................................................................................................................... 122
Figure 5.13 wheel rotation calculation block .............................................................................................................................................................................................................................................................................. 124
Figure 5.14 Steering angle input .............................................................................................................................................................................................................................................................................................. 126
Figure 5.15 Wheel driving torque for all wheels ........................................................................................................................................................................................................................................................................... 126
Figure 5.16 Vehicle longitudinal velocity .................................................................................................................................................................................................................................................................................. 127
Figure 5.17 Vehicle lateral velocity .......................................................................................................................................................................................................................................................................................... 128
Figure 5.18 (a) Longitudinal forces of the wheels (b) lateral wheel force .................................................................................................................................................................................................................................. 128
Figure 5.19 Wheels’ Angular Velocities .................................................................................................................................................................................................................................................................................. 129
Figure 5.20 (a) Vehicle body slip angle (b) yaw rate of the vehicle .................................................................................................................................................................................................................................. 130
Figure 5.21 Steering angle series ............................................................................................................................................................................................................................................................................................ 131
Figure 5.22 Wheel driving torque ............................................................................................................................................................................................................................................................................................ 131
Figure 5.23 Vehicle longitudinal velocity without ESC in FMVSS 126 .................................................................................................................................................................................................................................. 132
Figure 5.24 Comparison of yaw rates ....................................................................................................................................................................................................................................................................................... 132
Figure 5.25 Sine with Dwell Steering input for open loop test ................................................................................................................................................................................................................................... 133
Figure 5.26 Driving wheel torque in open loop .............................................................................................................................................................................................................................................................................. 134
List of figures

Figure 5.27 Vehicle longitudinal velocity in open loop simulation ..................... 134
Figure 5.28 (a) Longitudinal wheel forces and (b) Wheel angular velocity in open loop simulation .............................................................. 135
Figure 5.29 Expected longitudinal wheel force to minimize instability ............. 136
Figure 5.30 Desired, targeted and actual yaw rate in open loop simulation .......... 136
Figure 5.31 Desired, targeted and actual vehicle slip angle in open loop simulation ... 137
Figure 5.32 Longitudinal wheel force in closed loop simulation ...................... 138
Figure 5.33 Desired, targeted and actual yaw rate in closed loop simulation ........ 139
Figure 5.34 Desired, targeted and actual vehicle slip angle in closed loop simulation 139
Figure 5.35 Differential Torque in closed loop simulation ............................. 140
Figure 5.36 Vehicle velocity in closed loop simulation ................................. 141
Figure 5.37 Yaw rate response of a linear quadratic regulator based controller with sliding mode wheel slip controller .......................................................... 142
Figure 5.38 Yaw rate response of a torque based ESC with fuzzy controller controller .............................................................................................................. 142
Figure 5.39 Yaw rate response form the simulation of the proposed method ... 143
Figure 5.40 Vehicle longitudinal velocity of the proposed system .................... 144
Figure 5.41 Vehicle velocity of a motor torque based fuzzy controller for stability ... 145
Figure 6.1 Proposed experiment setup ......................................................... 149
Figure 6.2 Physical measurement of the four in-wheel platform ....................... 150
Figure 6.3 Current measurement of the in-wheel motor to determine the electrical torque .............................................................................................................. 152
Figure 6.4 In-wheel test rig for observation and control of the wheel ............... 152
Figure 6.5 Test data of motors shows similarity in responses against throttle input voltages varying from 0.8 V to 3.6 V......................................................... 153
Figure 6.6 In-wheel motor inner view .......................................................... 159
Figure 6.7 R-C filter to remove noise from hall sensor feedback signal .......... 159
Figure 6.8 Digital PID controller for wheel control with the subroutines .......... 160
Figure 6.9 function for In-coming pulse counting of frequency counter for hall sensor .............................................................................................................. 161
Figure 6.10 Proportional component of the PID controller ............................ 163
Figure 6.11 Integral component of PID ....................................................... 164
Figure 6.12 Derivative component of PID controller ..................................... 165
List of figures

Figure 6.13 Merging function for proportional, integral and derivative components ..166
Figure 6.14 Calculation and implementation of the sampling time .........................167
Figure 6.15 Individual motor control function..........................................................168
Figure 6.16 Accumulator function to provide throttle voltage to motor driver ..........168
Figure 6.17 Overall system for motor control using digital controller ....................169
Figure 6.18 Throttle voltage vs. wheel speed in terms of Hall Effect sensor frequency .................................................................171
Figure 6.19 Motor speed control test 1 shows rise time vs. settling time ...............173
Figure 6.20 Motor speed control test 1 shows overshoot (voltage) and steady state error (voltage) ..................................................................................................................173
Figure 6.21 Motor speed control test 2 shows rise time vs. settling time after tuning .174
Figure 6.22 Motor speed control test 2 shows overshoot vs. steady state error voltages after tuning ................................................................................................................175
Figure 6.23 Motor speed control load test shows rise time vs. settling time ..........176
Figure 6.24 Motor speed control load test shows overshoot (voltage) and steady state error (voltage)................................................................................................................176
Figure 7.1 Structure of Electronic Stability Control ..................................................183
Figure 7.2 Vehicle model for finding centre of gravity ..............................................184
Figure 7.3 Weight distribution by axial sensors.........................................................185
Figure 7.4 Vehicle model with in wheel electric motor.............................................186
Figure 7.5 Case-1: Centre of gravity in normal position and the incline angle is 0 degree ...............................................................................................................................187
Figure 7.6 Case-2: Centre of gravity shifted horizontally by 10 cm and the incline angle is 5 degree .................................................................187
Figure 7.7 Case 1 of ratio 60:40 with over steering behaviour .................................192
Figure 7.8 Case 2 with ratio 50:50 with neutral or slight over steering behaviour.....193
Figure 7.9 Case 3 of ratio 40:60 with under steering behaviour ...............................194
List of tables

Table 2-1 Conventional propulsion systems of electric vehicles.
Table 4-1 Average values of tire-road friction.
Table 4-2 Levels or phases of ESC system.
Table 5-1 Numbered labels in the figure 5.3 describe the attributes of input and output parameters in the vehicle body modelling.
Table 5-2 Numbered labels in the figure 5.4 and figure 5.5 describe the attributes of input and output parameters in the wheel modelling.
Table 5-3 Numbered labels in the figure 5.6 describe the attributes of input and output parameters in the steady state steering angle block.
Table 5-4 Numbered labels in the figure 5.7 describe the attributes of input and output parameters in modelling desired yaw rate.
Table 5-5 Numbered labels in the figure 5.8 describe the attributes of input and output parameters in modelling desired slip angle.
Table 5-6 Numbered labels in the figure 5.9 describe the attributes of input and output parameters in modelling control law block.
Table 5-7 Numbered labels in the figure 5.10 describe the attributes of input and output parameters in the modelling of force difference calculation block.
Table 5-8 Numbered labels in figure 5.11 describe the attributes of input and output parameters in Differential driving calculation block.
Table 5-9 Numbered labels in figure 5.12 describe the attributes of input and output parameters in differential braking torque calculation block.
Table 5-10 Numbered labels in figure 5.13 describe the attributes of input and output in parameters require wheel rotation calculation.
Table 5-11 Parameter used in simulation.
Table 6-1 Measurement of developed platform with four in-wheel motor.
Table 6-2 PID controller tuning effects.
Table 7-1 Vehicle parameters for center of gravity shifting simulation.
ABSTRACT

Recently, environmental pollution from fuel based vehicles has become a major concern due to the increasing number of vehicles on the road. Pure electric vehicles have become the better option in comparison with the other alternative fuel based vehicles. Electric vehicles are a viable solution for passenger cars, if improvements continue in EV technology and infrastructure developments. Technological improvement in EV has introduced in-wheel motors for a new and simplified drivetrain which consists of four in-wheel motors. Focus has been given in this research work to active safety controls system like Electronic Stability Control to assist the driver in maintaining control of EV in critical manoeuvers.

A simplified drivetrain with four in-wheel motors is an advanced idea which gives greater flexibility in controlling the vehicle. This also reduces weight by eliminating mechanical parts. Our research suggests that independent control of the in-wheel-motors is the preferred method for vehicle stability. To develop this type of controller requires a detailed analysis of vehicle dynamics to establish a comprehensive model. Physical development of the platform can then begin. Advanced control technique is used in the simulation to analyse the vehicle dynamics and calculate the required wheel torque to stabilize the vehicle in motion.

Electric motors in the wheels facilitate the instant torque generation which provides the option of changing longitudinal forces in the wheels. The performance and effectiveness of this control approach is evaluated based on predefined standard test manoeuvres. To verify the proposed wheel control technique, a hardware interface has been developed to create the required torque by controlling the rotation of the in-wheel motor using an intelligent digital controller. This digital controller accepts the generated demand of wheel rotation and performs the appropriate action. The results obtained from the stability controller simulation were satisfactory. The experiment showed that the wheel speed was controlled well.
List of publication


### List of symbols and abbreviations

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( F_{aero} )</td>
<td>Aerodynamic drag force</td>
</tr>
<tr>
<td>( M_Z )</td>
<td>Aligning moment in Pacejka's tire model</td>
</tr>
<tr>
<td>AWD</td>
<td>All-wheel-drive</td>
</tr>
<tr>
<td>( \dot{\omega} )</td>
<td>Angular acceleration of the wheel</td>
</tr>
<tr>
<td>( \omega_w )</td>
<td>Angular speed of a wheel</td>
</tr>
<tr>
<td>( K_a )</td>
<td>Armature winding constant</td>
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<tr>
<td>( I_{dc} )</td>
<td>Average dc bus current</td>
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<tr>
<td>( T_b )</td>
<td>Brake torque of the wheel</td>
</tr>
<tr>
<td>( F_{X_{brake}} )</td>
<td>Braking force</td>
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<td>CoG</td>
<td>Centre of gravity</td>
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<tr>
<td>( \eta )</td>
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<td>Control law variable in sliding surface</td>
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<td>( a )</td>
<td>Control variable</td>
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<td>( C_{af} )</td>
<td>Cornering stiffness of front tire</td>
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<tr>
<td>( C_{ar} )</td>
<td>Cornering stiffness of rear tire</td>
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<tr>
<td>( C_\alpha )</td>
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<td>( M_{\delta h} )</td>
<td>Corrective yaw moment</td>
</tr>
<tr>
<td>( N_c )</td>
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</tr>
<tr>
<td>E</td>
<td>Curvature factor in Pacejka's tire model</td>
</tr>
<tr>
<td>( \beta_{\text{desired}} )</td>
<td>Desired side slip angle</td>
</tr>
<tr>
<td>( \Psi_{\text{desired}} )</td>
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</tr>
<tr>
<td>( T_{dfl} )</td>
<td>Differential driving torque for front left wheel</td>
</tr>
<tr>
<td>( T_{dfr} )</td>
<td>Differential driving torque for front right wheel</td>
</tr>
<tr>
<td>DGPS</td>
<td>Differential Global Positioning System</td>
</tr>
<tr>
<td>( l_f )</td>
<td>Distance of front axle from CoG</td>
</tr>
<tr>
<td>( l_r )</td>
<td>Distance of rear axle from CoG</td>
</tr>
<tr>
<td>( T_d )</td>
<td>Drive torque of the wheel</td>
</tr>
<tr>
<td>( r_{eff} )</td>
<td>Effective tire radius</td>
</tr>
</tbody>
</table>
ESC  Electronic Stability Control
FMVSS  Federal Motor Vehicle Safety Standards
FPGA  Field Programmable Gate Array
4WD  Four-wheel-drive
FCV  Fuel cell vehicles
GPS  Global Positioning System
g  Gravitational force
$h_{CoG}$  Height of CoG
$h_{aero}$  Height of the point of application of the aerodynamic resistance
$S_H$  Horizontal shift in Pacejka's tire model
$\theta$  Inclined angle or angle of slope of the vehicle with ground
$T_{dfl,init}$  Initial driving torque for front left wheel
$T_{dfr,init}$  Initial driving torque for front right wheel
I/O  Input output
ICE  Internal Combustion Engine
OICA  International Organization of Motor Vehicle Manufacturers
$a_y$  Lateral acceleration
$a_{yp}$  Lateral acceleration at a point $P$ is chosen between CoG and front axle
$a_{y_{meter}}$  Lateral acceleration from sensor
$\ddot{y}$  Lateral acceleration of vehicle
$F_{Ylateral}$  Lateral force
$F_{yfl}$  Lateral force on front left wheel
$F_{yfr}$  Lateral force on front right wheel
$F_{yrl}$  Lateral force on rear left wheel
$F_{yrr}$  Lateral force on rear right wheel
$F_{yr}$  Lateral forces at rear axle
$v_y$  Lateral velocity
$\dot{y}$  Lateral velocity of vehicle
$F_y$  Lateral wheel forces
$\dot{x}$  Longitudinal acceleration of vehicle
\( F_{xft} \)  Longitudinal force on front left wheel
\( F_{xfr} \)  Longitudinal force on front right wheel
\( F_{xrl} \)  Longitudinal force on rear left wheel
\( F_{xrr} \)  Longitudinal force on rear right wheel
\( F \)  Longitudinal or lateral force
\( \sigma \)  Longitudinal slip ratio
\( C_\sigma \)  Longitudinal stiffness of tire
\( v_x \)  Longitudinal velocity
\( \dot{x} \)  Longitudinal velocity of vehicle
\( F_x \)  Longitudinal wheel forces
\( F_{friction} \)  Maximum friction force
\( \mu_{max} \)  Maximum value or the peak value of tire road friction coefficient
\( T_{es} \)  Motor steady torque and
\( T_e \)  Motor torque
NADS  National Advanced Driving Simulator
NHTSA  National Highway Traffic Safety Administration
NM  Newton Meter
N  Normal force of vehicle for load
OEM  Original equipment manufacturer
D  Peak value in Pacejka's tire model
PID  Proportional-Integral-Derivative controller
PWM  Pulse width Modulation
R  Radius of the circular road
\( \Delta F_x \)  Required differential longitudinal force difference
\( F_{result} \)  Resultant force of these
RPM  Rotation per Minute
C  Shape factor in Pacejka's tire model
S  Sliding surface
\( \alpha_f \)  Slip angles at front tires
\( \alpha_r \)  Slip angles at rear tires
\( \delta_{ss} \)  Steady state steering angle
\( \delta \)  
Steering angle

B  
Stiffness factor in Pacejka's tire model

\( T_{surf} \)  
Surface torque brushless DC Motor

\( \tau_m \)  
Time constant

\( \mu \)  
Tire road friction coefficient

\( \rho_t \)  
Torque fixed ratio

\( \Phi_t \)  
Total flux per pole in webers

\( \lambda \)  
Variable used in Dugoff tire model

\( X \)  
Variable used in Pacejka's tire model as input

\( Y \)  
Variable used in Pacejka's tire model as output

\( m \)  
Vehicle mass

\( \beta \)  
Vehicle slip angle

\( v \)  
Vehicle velocity

\( F_z \)  
Vertical force on tire

\( S_V \)  
Vertical shift in Pacejka's tire model

\( J_w \)  
Wheel inertia

\( l_w \)  
Width between the wheels

\( I_z \)  
Yaw moment of inertia of vehicle

\( \Psi \)  
Yaw rate of the vehicle
Chapter 1
Introduction
1 Introduction

1.1 Research background and motivation

The automotive industry is a major contributor to national economic development. Around the world the number of vehicle is increasing. According to the International Organization of Motor Vehicle Manufacturers (OICA), passenger cars make up approximately 74% of the total motor vehicles annual production in the world. If we consider the case of Australia, it can be seen from the Bureau of Statistics (ABS) survey in Figure 1.1, that the number of passenger cars increased by 10.1% from 2008 to 2013 [1].

![Number of passenger vehicle in Australia](image)

*Figure 1.1 Passenger vehicles increase in Australia from 2008 to 2013*

Transport activity is the one of the major sources of emissions which involves the combustion of fossil fuels. The increasing numbers of vehicles on the road causes environmental problems like atmospheric heating and air pollution. Electric Vehicles with low carbon emissions are a viable alternative.

Up until recently, electrical battery powered vehicles have been inefficient, expensive and unpopular. Due to the limited range, the long charging times and the short life cycle of the batteries in electric vehicles of the past, the public reception of electric vehicles
has been poor. As such, motor companies have not had the motivation to invest in these technologies, and the development of electric vehicles has been stifled by lack of industry demand. However, due to recent improvements in battery technology, electric vehicles are becoming a more feasible means of transportation. Modern electric vehicles have a comparable range to many petrol powered vehicles and the cost and charging times for the batteries are continually going down. One such example, the Tesla Roadster, has a single charge range of almost 350 km, and has a life expectancy of up to 7 years. Recent developments in battery technology have made batteries smaller, lighter and more powerful. Advances in electric vehicles have led to innovative ideas in drive train designs. One of the most innovative ideas is to employ the emerging technology of in-wheel motors. This promising idea also requires an advance control for normal vehicle operation and critical manoeuvres. The new era of drive train control will require analysis, design and development of both vehicle and controller.

### 1.2 Objectives

The increase in the number of passenger vehicles has led to more vehicle crashes. Electronic Stability Control (ESC) is an active safety control system to assist the driver to maintain directional control of the vehicle in critical manoeuvring conditions. ESC improves controllability and prevents accidents due to loss of control. Many studies have been done from different perspectives to analyse the effectiveness of ESC in reducing loss of control and in accident prevention.

An advanced EV with in-wheel motors has a drive train consisting of 4 in-wheel motors. To make a simple and effective ESC which is viable for the selected vehicle type requires fully independent control of drive torque distribution. The objective of this research is to find a simple method of overcoming the complexity of ESC, to use the advanced features of individual wheel control, to analyse the proposed method and to assess its potential by testing.
1.3 Approaches

To find a suitable method, analysis is done on the existing available ESC methods. Differential braking based, steer-by-wire and differential torque based ESC have been scrutinised. We focused on the required resources complexity and performance of the systems in a vehicle. Considering available resources, flexibility of individual wheel control and the improvement in expected longitudinal response, a driving torque based ESC will be chosen for the selected four in-wheel EV. Further investigation is done on existing driving torque based ESC to identify a less complex solution.

Simulation is the primary way to do deep analysis in a minimum time and with less cost hence our approach is to develop the entire simulation environment related to the selected vehicle and proposed method. Simulation provided satisfactory outcomes and supported the proposed method. In parallel, the development of such a conceptual vehicle will help to identify the problems in design. Another hardware testing setup will be developed if necessary to obtain an experimental outcome of the proposed method.

1.4 Dissertation organisation

In order to investigate an effective and simple vehicle stability control, a brief idea of the research is provided here by describing the motivation, objective and approach. Details of this research work are described in the following chapters. In chapter 2 we discuss background information on emissions, emission reduction and control of EV, the basic concept of EV, in-wheel motor technology and advanced architecture of EV.

In chapter 3, basic information on ESC is discussed. The concept of stabilizing using ESC, the effectiveness of ESC and different types of ESC are reviewed. Then, an ESC type is proposed and ESC regulations are described.

In chapter 4, investigation continued on the proposed type and analysis of differential torque based ESC for in-wheel EV. An overall description of a proposed system is included at the end of this chapter.
In chapter 5 the required simulation environment is modelled with the reference vehicle, wheels and controllers. Modelling of these subsystems is described in steps and at the end the simulation result is presented.

In chapter 6, the proposed hardware system is described. Experimental setup, digital controller development for wheel speed control and results are presented.

In chapter 7, advanced topics are discussed on the effect of centre of gravity shifting and the simulation results. Also, the effect of load distribution is analysed for different distributions to determine proper vehicle handling.

Finally, Chapter 8 summarizes the work, highlights the contributions, and discusses directions for future work in this area.
Chapter 2
Pollution and the advanced electric vehicle
2 Pollution and the advanced electric vehicle

2.1 Introduction

Automobiles as a means of transport play an essential role in the economy as they affect the mobility of populations and businesses. Most of automobiles or vehicles on the globe use fossil fuel or petroleum in their inefficient internal combustion engine (ICE). Fuels, dug out of the earth, are fast becoming a scarce commodity. It is inevitable that our fossil fuel sources will expire at some point in the future. Also weighing into the argument against fossil fuel sources is that even with modern advances in combustion engines, motor vehicles are still a major contributor to air pollution worldwide. While an exact figure is difficult to ascertain, it is estimated that motor vehicles contribute to anywhere between 50% to 90% of the world’s air pollution every year. With the number of cars on the road set to almost double in the next 20 years, unless something changes, the amount of air pollution from motor vehicles is likely to rise considerably in the years to come. Public awareness has increased around the world and this has created a demand for less pollutant and more fuel efficient transports systems. This change in consumer’s behaviour has been already noticed by the automotive industry. As an alternative to conventional fossil fuel based vehicles, electric vehicles are becoming widespread.

In this chapter, background information on emissions caused by ICE is discussed. Pollution control by using EV and making EV viable for urban transport is discussed here based on the paper “Pollution control and sustainable urban transport system - electric vehicle” [2].

There are potential issues in relation to the development of electric vehicles as a viable transport system. Advanced architecture and better handling are also important issues which are addressed in this study. Basic concepts of EVs, in-wheel motor technology in EVs and different drive trains of EV are also discussed here.
2.2 Transport systems and their effect

There are several issues related to the environmental impacts of vehicles. Carbon emission is the significant one for environment pollution. As road transport is mostly dependent on petroleum therefore vehicles are responsible for a significant and growing segment of worldwide emissions of carbon dioxide. Carbon emission is taken as the single key for environmental impact.

Production of greenhouse gas is generally expressed in terms of how much carbon dioxide (CO2) is produced. A motor vehicle produces 2.3 kilogram of carbon-dioxide by burning a litre of petrol. Millions of these vehicles on the planet are emitting huge amounts of greenhouse gas. For example, 1.5% of global greenhouse gas emissions are produced in Australia [3]. These emissions are classified into six areas by the Department of Climate Change in Australia. According to ABS, in 2007 transport activities contributed 78.8 Metric tons of CO2 equivalents which is 13% of Australia’s net emissions.

Figure 2.1 shows that transport activity is one of the largest sources of pollution, and emissions from this sector are increasing. The survey shows it was 26.9% higher in 2007 than in 1990.

![Figure 2.1 Greenhouse gas emissions in 2007 in Australia](image)
Global warming is a result of the greenhouse gas emissions. An increase in earth temperature caused by increased amount of greenhouse gases will result in a major ecological disaster.

Moreover, vehicle exhausts can create various health hazards which are dangerous for current and future generations. Besides carbon dioxide and water, the ICE combustion products contain a certain amount of sulphur dioxide, nitrogen oxides and carbon monoxides which are toxic to human health. Sulphur dioxide causes chronic respiratory illness and Carbon monoxide reduces the oxygen carrying capacity of the blood which results in less oxygen to the brain and the tissues.

2.3 Pollution control and viable transport by EV

EV technology is chosen to effectively resolve environmental issues and to provide more feasibility [4, 5]. If the efficiency of different types of vehicles is compared in terms of energy use, it can be seen that EV are more efficient as shown in Figure 2.2.

![Figure 2.2 Overall efficiency of different types of vehicle](image)
Companies are developing alternative fuel vehicles to address the issue of carbon emission. Administrative and government bodies are reviewing safety, design and maintenance related issues for alternative fuel technologies like EV. There are number of issues existing in different sectors in the development of EV. Some of them are as follows:

- Decrease in environmental pollutions
- Use of renewable and cleaner energy to avoid cumulative emission
- Improvement in vehicle technologies

In this study improvement in EV technology is discussed and improvements in stability control are discussed in Chapter 4.

### 2.4 Conventional concept of EV

An electric vehicle consists of an electric motor that drives the wheels, an energy (such as batteries) and a controller that controls motor and regulates the energy flow to the motor.

There are six basic types of electric vehicle considering energy source and architecture, which may be classed as follows.

- **Battery Electric Vehicle:** It consists of an electric motor, battery and controller.
- **Hybrid electric vehicle:** This type of Electric Vehicle has a combination of an electric motor and an Internal Combustion (IC) engine; this type of vehicle is very likely to become the most common type of Electric vehicle.
- **Replaceable fuel Electric Vehicle:** This is another type which can use different source of energy like fuel cells or metal air batteries.
- **Road-Powered Electric Vehicle:** There are vehicles supplied by power lines.
- **Solar Electric Vehicle:** These are electric vehicles which use solar energy directly from solar panels.
- **Alternative energy EV:** vehicles that store energy by alternative means such as flywheels or super capacitors. These vehicles are nearly always hybrids using some other source of power as well.
Besides the list above, EV can be categorized in terms of size and usability.

This research work focuses on developing a passenger EV because it is the largest group of vehicles in the world. Replacing conventional ICE vehicles with EVs will make a major contribution to pollution control.

The concept of the conventional electric vehicle is essentially simple as shown in Figure 2.3. The vehicle consists of a battery, an electric motor, and a controller. The battery is normally recharged from mains electricity via a plug and a battery charging unit. The controller controls the motor to go forward or reverse, and the vehicle speed. For the drive system of EV, the electric motor is connected through the transmission gear to the driving the wheels.

![Figure 2.3 Basic concept of a conventional Electric Vehicle](image)

A conventional EV has a single electric motor instead of an internal combustion engine. Electric motors are efficient in term of energy consumption over ICEs which is one of the important reasons to use them.

### 2.5 Different drive train in EV

An EV has a more flexible drive train compared to a conventional ICE vehicle. The increased flexibility is mainly due to the energy flow via electrical wires rather than
mechanical shafts. These flexible wires make it possible to distribute the subsystems of the drive train more freely. The flexibility is increased due to the many options in the two major subsystems of the drive train, the energy source system and the electric propulsion system. This increase in flexibility leads itself to multiple propulsion systems that can be used for an EV. Based on these electric propulsion configurations, EV can be classified in six typical groups [6]. Table 2-1 shows these vehicles with descriptions.

Table 2-1 Conventional propulsion systems of electric vehicles

<table>
<thead>
<tr>
<th>EV Propulsion Configuration</th>
<th>Propulsion Description</th>
<th>Remarks</th>
</tr>
</thead>
</table>
| 1                           | This configuration is a direct conversion of ICE, it consists an electric motor, a clutch, a gearbox and a differential. | • Easy to implement  
• Use of clutch and gear is a redundant component in an electric motor based drive train as the electric motor has the ability to generate torque over a wide range of speed.  
• While cornering outer wheel covers greater distance than inner wheel. |
| 2                           | In this configuration an electric motor, a differential and a fixed gear is used instead of a gearbox and clutch. | • Not similar to ICE.  
• Comparative reduction in both the size and weight of propulsion system.  
• Fixed gear and differential match electric motor speed with wheel speed.  
• Inefficient due to use of differential. |
<table>
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<tr>
<th></th>
<th>Description</th>
<th>Pros</th>
<th>Cons</th>
</tr>
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</table>
| 3 | This propulsion configuration has an electric motor, a fixed gear and a differential integrated and placed on the front axle, it works as a conventional front wheel drive vehicle. | - Compact drive train provides flexibility in designing the vehicle. 
- Higher efficiency achieved using fixed gear. 
- Less complex and reduced in size and weight. 
- Adopted by most EVs. 
- Inefficient due to use of differential. | |
| 4 | Two electric motors are connected with separate fixed gears in this configuration. This propulsion has the flexibility for driving the two motors at different speeds for cornering and other operations. | - Higher efficiency as differential is removed. 
- Two motors are electronically controlled for differential actions. | |
| 5 | High speed inner rotor type electric motor with fixed gear is placed here. This is called by-wheel motor. Two such in-wheel motors are used in this propulsion configuration. | - Mechanical transmission path is shortened which is helpful tactic. 
- Technical advantages achieved placing input and output shaft inline. 
- Motor speed is reduced to match desired wheel speed. | |
| 6 | Further modification is done in in-wheel systems by placing the lower speed outer rotor electric motor in the hub of a wheel without a fixed gear. This configuration has two in-wheel motors for propulsion. | - Lower weight and size is achieved by removing gear. 
- Each wheel is controlled electronically and can have different wheel speed for manoeuvres. 
- Offers better vehicle design. | |
An electric motor in an EV has several advantages including better control of the vehicle. In a conventional EV, the speed differentials are connected with the drive-motor and reducers, then to the driving wheels. Though the two wheels are connected with the single motor, their speed would be different [7]. To control the EV, hence to control the speed of two wheels, it is necessary to calculate the relationship between the motor and wheel speed considering the reduction ratios.

In some EVs two electric motors are connected to the wheels for propulsion. Synchronization of these motors is required for better handling. But the presence of mechanical gears can make this calculation very complex. In the case of ESC, this complex calculation will weaken the real-time characteristic of the electric motor and electronic control.

2.6 In-wheel motor technology

Despite using an electric motor and batteries instead of fossil fuels, most electric vehicles are based around the same basic design as the fossil fuel powered cars that came before them. Their drivetrain is still based around a single, centralised motor, with transmission to either two, or more rarely all four wheels. This design approach has its advantages and disadvantages. A centralised single motor design allows for a simpler method for controlling the vehicle. It also allows a single location for service and maintenance of the motor. However, it still suffers from some of the many problems that beset fossil fuel powered cars. With a single centralised electric motor, the transmission of power to the wheels will still result in wasted energy due to the friction in the transmission. Additionally, a differential will still be required to ensure the wheels spin at an appropriate speed during cornering. This adds weight to the vehicle, which in turn reduces the range and power of the vehicle.

An alternative to a single, centralised motor is to have in-wheel motors. In wheel motors reduce the need for a transmission system as the power from the motor is already at the wheel. As such, in-wheel motors can be highly efficient, with some motors transmitting as much as 98% of the electrical energy required to run them on the road [8].
In-wheel motors also allow the vehicle to be lighter as most of the weight of the engine, transmission and drivetrain can be removed.

With some additional high level control, in-wheel motors can be used to improve control, stability and performance of a vehicle in a range of situations. Having the ability to individually control the speed of four different motors allows for advanced scenarios like launch control, stability control, and torque vectoring. As such, the controller should be layered, so as to be as modular and flexible as possible, whilst facilitating development of both the higher level, and lower level control systems.

Advances in integration of the motor into the hub of the wheel have changed the in-wheel motor a lot since the first in-wheel motor was used. After disappearing for decades, in-wheel motors have gained popularity for their advanced features and performance. An in-Wheel motor system has an electric motor, sensor, brake and wheel. This integrated system is placed in the hub of the wheel. This modular electric drive system can be integrated into a wide range of vehicle applications starting from a light bicycle to a heavy bus. They can be integrated into many different platforms like electric vehicles (EVs), hybrids and fuel cell vehicles (FCV). Currently, in-wheel motor EVs are of great interest in electric vehicle technology for researchers and industry. Generally, in-wheel motors are of two types.

**Indirect drive in-wheel system**

This type of in-wheel motor has an inner rotor type high speed permanent magnet motor and includes fixed gearing. When the vehicle is at high speed, a low torque is required to operate. Considering this lower torque requirement, a smaller size electric motor with lower weight is used here. This small motor can operate at higher speeds and control the vehicle velocity. Figure 2.4 shows an indirect-drive in-wheel system.
Direct drive in-wheel system

Figure 2.5 shows a direct-drive In-wheel system. In this type of system, an outer rotor type electric motor is used. To make the wheel speed and motor speed the same a bigger motor is used in this system without any fixed gearing. This motor has more weight than the motor in an indirect drive system, but removal of the fixed gearing makes the wheel lighter, less complex and increases the efficiency as there are no transmission losses.
Different parts of an in-wheel motor which is used in this study are shown in Figure 2.6. In this in-wheel motor the stator is on the inner side. The stator also includes the coils and sensor. The rotor is on the outer side, attached to the wheel. For this in-wheel motor, a motor driver supplies power and is controlled by the signal from the intelligent controller. Built-in sensors in this in-wheel motor system give the speed and the location of the motor. The drum brake is also integrated into the hub of this wheel for braking.

Figure 2.6 Exploded view of used in-wheel motor

2.6.1 Technical feasibility of using in-wheel motor

Locating the electric motor within the wheel hub offers many advantages as discussed below.

- In a conventional Battery EV, the propulsion system consists of batteries, electric motors with drives, and transmission gears to wheels. But in-wheel motor based drive train system can eliminate the transmission and its many components. Transmission losses can be minimized, and the operation efficiency and reliability can be improved [9] with this in-wheel architecture.
• Compared to an ICE, the electric motor is more sensitive to torque. It is easier to generate a certain torque with an electric motor than it is with an ICE. Motor torque can be measured easily by knowing the motor current. So it is possible to disperse and arrange the torque quickly according to requirements. Braking torque is also easier to generate by an electrical motor, compared to that of an IC engine with hydraulic brake.

• In-wheel reduces the traditional clutch and mechanical gear, thus giving more space for layout design and more flexibility in body design. Extra batteries can be installed in place of a mechanical clutch and transmission to increase the driving range per charge. Figure 2.7 shows conceptual chassis layout of a conventional ICE vehicle, an EV and a modern In-wheel EV.

![Figure 2.7 Conceptual chassis layout of conventional ICEV, EV and modern in-wheel EV](image)

• Regeneration of power is maximized by using in wheel motors. Different algorithms can be used for regeneration by keeping the in-wheel motor in non-driving mode when it is not required. Thus an in-wheel system can save energy for more range coverage.
In-wheel motors allow regulation of the driving and braking torque with precision which means a greater controllability. Using the electrical characteristics of the motor, measuring torque and controlling torque is much easier. It is possible to create torque difference between the left and right wheels by actuating them individually without any help of gear or clutch system. These in-wheel electric motors have an integrated breaking system as well as an integrated wheel rotation sensor. Operating voltages level of the in-wheel motor can be shifted to a higher value by switching from normal operation to turbo operation for more speed and manoeuvrability.

There are disadvantages of using an in-wheel motor in EV such as:

- The in-wheel motor environment is hostile where components suffer up to 20g vs. less than 5g when inboard.
- In-wheel motors increase the vehicle unsprang mass. This results in a decrease in ride comfort, in a decrease of road holding capability and in an increase in suspension travel.

2.7 Concept of four in-wheel EV

The electric vehicle has now entered its third century; many advances have been seen in the EV over this time. Like other innovative ideas for an EV drive train, in-wheel motors have been implemented previously. However, it is noticed that researchers, manufacturers and governing bodies have their focus concentrated on electric vehicle with advanced technology for more efficiency and better handling. This attention on EV has resulted in simplified architecture and better drivetrain concepts for EV.

In contemporary architecture drive shafts, gear and differentials become unnecessary resulting in an overall decrease of size, weight and transmission losses. Vehicle layout facilitates an increased number batteries and battery swapping. Recently researchers and manufacturers are focusing on an advance EV configuration with four in-wheel motors. Four in-wheel motors work as the drive train of the vehicle. These motors are fully integrated and are connected as a modular electric drive system. Most of these wheels
have a built-in inverter, control electronics and software. This leads to a simplified drive train. Figure 2.8 shows the simple architecture of 4 in-wheel-motor electric vehicle.

![Figure 2.8 Four in-wheel motor EV architecture](image)

In this configuration, each wheel is controlled from a central controller. Its output torque is directly transmitted to the wheel for direct drive. This concept gained popularity with researchers because it improves control of the vehicle in an intelligent way and also gained the attention of several manufacturers.

### 2.8 Challenges of using four in-wheel motors EV

The simultaneous, real-time control of four individual motors in a motor vehicle is something that involves a degree of complexity as the speed of each motor needs to be controlled with a high degree of accuracy. It is therefore important that the low level control of the vehicle is highly accurate to allow a higher level controller the flexibility it needs for the safe control of a moving vehicle. Another important consideration in the design of this motor controller is its response time. The controller needs to respond quickly so that a higher level controller can work efficiently. If the lower level controller is slow to respond or is inaccurate in its response, then the higher level controller will not be able to function properly and effective control of the vehicle will not be possible.
The speed of four in-wheel motors could be different regardless of using identical motors and identical controllers. Synchronization of four motors has to be achieved for a four in-wheel power train. Normal operations like longitudinal movements require similar speed in all four wheels. In this case, the central controller controls these motor to achieve the synchronization. In other scenarios, like turning when the vehicle is over-steering or under-steering, the four in-wheel EV has to control its motors with differential rotation to achieve the proper heading. The controller should be responsible for the stability of the vehicle. To drive the vehicle with four in-wheel motor propulsion a central controller is required to synchronize the four motors. But most OEMs' prefer centralised motors over in-wheel motors. This poses a major challenge to the implementation of a central controller. It is essential to develop an intelligent centralized controller to control the four in-wheels motors. An increase in unsprung mass requires in-wheels with suspension for comfort driving. Steer-by-wire, brake-by-wire and suspension-by-wire would be challenging due to high voltage requirements and added complexity.

2.9 Four in-wheel motors EV development for research

A four in-wheel motor EV has very simple architecture and the potential to control the torque of each wheel. This creates options for advance control such as stability control of the vehicle. Figure 2.9 shows the development of a four in-Wheel EV which is used in this study.

This four in-wheel EV is used as a test bench for several areas of research. These examples include motor control for advanced drivetrains, better handling by distributing load correctly and stability control by differential wheel speed. In this study we focus on yaw stability control for an EV.
Some of the current features of the prototype car are given below:

- Four in-wheel motors of 800 watts
- Individual motor controller
- 60 volt battery pack
- Plug-in charger
- Built in Hall effect sensors for wheel rotation sensing
- Gyro sensor for yaw rate sensing
- Lateral acceleration sensing module
- Steering angle position sensor
- Central controller for motor drivers

Other components will be added in the future.

2.10 Summary

In this chapter, environmental pollution and the effects of transport activities are discussed. Electric vehicles are more environmentally friendly because of their efficiency and can be a viable transport system. Advances have been achieved in
electric vehicle technologies which provide flexibility in designing drive trains, which are compared here. The in-wheel motor is reviewed here and we discuss how to use them effectively to fulfil the current requirement in EV development.

The increased simplicity and efficiency achieved by using four in-wheel motors have been described in this chapter. A brief description of an EV with simple architecture and various drivetrain concepts has been studied and included in this chapter. It is challenging to use such a drive train as this advanced power train concept requires advanced controlling to achieve the desired operations. In-wheel motors provide torque to each wheel independently and can be used as a type of steering control input to make stability control possible.

In the next chapter, analysis on stability control, the different methods of stability control are evaluated to determine a suitable method for four in-wheels EV.
Chapter 3

Electronic stability control
3 Electronic stability control

3.1 Introduction

In this chapter, the basic idea ESC and its variants are discussed based in the paper “A suitable ESC system using sliding mode controller for an in-wheel electric vehicle” [10]. Different ESC techniques for electric vehicles with four in-wheel electric motors have been investigated to find the most suitable method. An analysis of differential braking, steer-by-wire and differential torque based ESC has been undertaken based on the required resources, complexity and performance of the systems in a vehicle. Regulation for ESC is also discussed at the end of this chapter.

A large number of road accidents occur due to human error or external circumstances. In both these scenarios a vehicle can reach its critical limits and become uncontrollable. ESC is an active safety control system to assist the driver to maintain directional control of the vehicle in critical manoeuvres.

ESC compares the driver’s steering intentions with the vehicle’s actual direction and operates a programmed intervention on individual wheels to correct for any variance. This system continuously monitors the dynamics of the vehicle. An ESC system uses some sensors to measure some dynamic parameters and then estimates more parameters from the measured values. From these measurements and estimations, the system detects the loss of control such as over steering or under steering. After detecting the instability the ESC system restores the direction of the vehicle automatically by using actuators. Actuation can be done in three different ways; differential torque based control, steer-by wire based control or differential braking based control.

A vehicle equipped with ESC will try to steer according to the driver’s intention and avoid skidding. Thus, ESC allows a driver to maintain the direction and stability of a vehicle beyond its normal capability up to a physical limit. Normal operations and performance of the vehicle are not affected by this intelligent system.
3.2 Concept of stabilizing using ESC

People normally drive within the physical limit of adhesion between the tires and the road. It is very difficult to drive at the physical limit of adhesion between the tires and road. At this physical limit of adhesion vehicles behave quite differently. To understand this phenomenon a special scenario, is shown in Figure 3.1. Here the vehicle is negotiating a turn on a curved road. In response to the correct steering input from the driver a vehicle may follow two different paths depending on the friction coefficient and vehicle speed. These two possibilities are given here in two different scenarios:

Scenario 1: when the road is dry and has a high tire road friction coefficient.
In this case high tire road friction coefficient will provide smaller lateral acceleration and vehicle will follow the nominal motion. It will follow the lower curved road ‘C’ shown in Figure 3.1 Yaw control functionality

Scenario 2: when the road is slippery with a small coefficient of friction or vehicle is at high speed.
In this case the vehicle would not follow the nominal motion and the radius of the turn will become larger as curve road ‘A’ shown in Figure 3.1.
In this situation driver may counter steer to control the vehicle but this may not be possible for the average driver. If it is not possible to control by counter steering then this will lead to loss of control and an accident can occur.

In the critical situation given in scenario 2 stabilizing a vehicle is very challenging due to the physical effects. Changing the steering wheel angle of a vehicle should change the direction of the vehicle but it depends on the actual side slip of the vehicle [11].
Figure 3.1 Yaw control functionality

If the side slip angle is large the direction can be changed very slightly by steering input. Steering input has no effect of changing the direction on a dry road where slip angle is more than 10 degrees and on icy road where side slip angle is 4 degree or less. If the situation becomes more extreme when the side slip angle of the vehicle increases due to a greater speed or a slippery road, then the counter steering by the driver will not create any corrective yaw moment and it will decrease the driver’s ability to steer the vehicle. Figure 3.2 shows the decrease of yaw moment due to the steering angle when the side slip angle increases [12].

When the side slip angle is in the range of 10 to 12 degree the steering induces a yaw moment close to zero. Then a different mechanism is needed to regain control of the vehicle.
When a vehicle is over-steered or under-steered and corrective yaw moment by counter steering fails to control the vehicle it can respond in an unpredictable way. This can cause loss of control and induced panic seizes the driver which makes it difficult to gain control of the vehicle. This creates a need for an intelligent control like ESC. ESC reduces the difference between the actual and nominal direction and keeps the vehicle slip angle smaller.

An ESC system maintains the yaw or direction of the vehicle according to the driver’s intention. While driving at the limit of adhesion, ESC can control the yaw or direction of the vehicle by determining the drivers intended yaw and vehicles actual yaw. If the vehicle response and driver’s intention do not match then ESC automatically turns the vehicle to the right direction. If the friction coefficient is very low then it would not be possible to entirely achieve the nominal yaw rate, but ESC may be able to make the yaw rate closer to the expected yaw rate [13] as shown in Figure 3.1 with the curve ‘B’.

![Figure 3.2 Yaw moments for different steering angle [12]](image)
ESC includes some sensors like wheel speed, yaw rate, lateral acceleration and steering wheel sensors to identify the dynamics of the vehicle. ESC has a different mode for yaw stability control and roll stability control. In this study yaw stability control mode is discussed. In yaw stability control mode ESC calculates:

- Direction of the vehicle using a steering angle sensor and vehicle speed sensor.
- Radius of the current path using lateral acceleration sensor and vehicle speed.
- The correct yaw rate of the vehicle travelling on the path by calculating radius and measuring the vehicle speed using on board yaw rate sensor, ESC system measures actual yaw rate.

In ESC a microcontroller based system usually does the required calculations mentioned above and estimates the intended direction and ideal motion compared with the actual motion. After comparison, if the difference between the measured yaw rate and calculated yaw rate exceeds the threshold value then the ESC system sends a control signal to the actuators to apply corrective force on the vehicle and restores yaw stability. Thus the ESC system intervenes early in an impending loss-of-control situation as an active safety system.

### 3.3 Effectiveness of ESC

Technological advancement in automotive control resulted in advanced active safety systems like ESC. It was first introduced in 1995 in European passenger cars. ESC is provided by the manufacturer as a standard or optional system. ESC was introduced worldwide under different names and is now included in less expensive passenger vehicles. Variants of the ESC system are based on the vehicle’s driving parameters and the different ways to reduce deviations. All these variants enhance the controllability of the vehicle in different manoeuver situations and prevent loss of control [14].

ESC must obey the laws of physics so it may not be always possible to avoid loss of control. As ESC reduces accidents by giving more control over vehicle, it is important to study the effectiveness of this system in practical ways. Many studies have been done
from different perspectives to improve the effectiveness of ESC in reducing loss of control and in accident prevention.

### 3.3.1 Effectiveness from crash data analysis

The real world effectiveness of ESC is analysed based on vehicle crashes reported by police in many countries. A review has been done of key published studies evaluating ESC for different countries around the world like Germany, Japan, Great Britain, France, Sweden and USA.

As ESC helps to prevent loss of control of the vehicle, it is expected that ESC would reduce single vehicle accidents. This study shows that ESC is highly effective in reducing single-vehicle crashes. For passenger cars, ESC reduces fatal crashes by 30-50%. For Sports Utility Vehicles (SUV) the reduction is 50% to 70% [15]. An updated study on the effectiveness of ESC [16] for USA shows that out of 2 million vehicle crashes 75% of vehicles are without ESC. It shows at least 600 thousand of those accidents could be avoided if these cars were fitted with ESC. The effect of ESC on multiple vehicle crash is not clear.

In [17] effects of ESC on multiple vehicle crash are mentioned as positive and negative both but only the positive effects are significant. This study says that fatal multiple vehicle crashes reduced by 35%.

Based on the crash data from Australia and New Zealand, the effectiveness of ESC was assessed in another study [18]. This study estimated that ESC reduced single vehicle crashes by 29%. This research also found the negative effect of having ESC in multiple vehicle crash. It is estimated that ESC increases the risk by 15% of multiple vehicle crashes. Several reasons have been suggested for the increase of multiple vehicle crashes. It was suspected that pre-crash intervention of ESC can cause a multiple vehicle crash. One reason would be that the presence of an ESC system can cause a change in driving style which can possibly increase the chances of a multiple crash. It is also possible that over-reporting of damage to expensive ESC fitted vehicles may show
an increased amount of multivehicle crashes. Research suggests further study may be necessary to confirm all these reasons.

It would be worth considering vehicle to vehicle communication for control when an ESC starts intervening in a multiple vehicle travel situation on road. Also an ESC system can be interfaced with the obstacle avoidance radar system to avoid multiple vehicle crashes.

### 3.3.2 Human factors analysis using a simulator

Research was conducted to find the effectiveness of ESC in real time by creating different situations in a simulator rather than analysing the crash data. Advanced simulation using the National Advanced Driving Simulator (NADS) facility was done to find the effectiveness of ESC in reducing loss of vehicle control [19]. In this research drivers from different age groups were presented with different scenarios such as obstacle avoidance departure, curve departure and wind-gust. The tests were done to find the loss of control in the presence and absence of ESC. The analysis found the benefit of having ESC present. It was clear that drivers maintained the control of vehicle in the presence of ESC. This research concluded that ESC system helped the drivers significantly to avoid loss of control in critical manoeuvres and was effective in 88% loss of control situations.

### 3.4 Different methods of ESC

Identification of unexpected behaviours of the vehicle and the generation of control signals for the actuators are done by the ESC system. These actuators apply corrective forces on the wheels to change the yaw to maintain yaw stability. In some cases the controller applies corrective steering wheel for yaw stability.
Based on the actuating behaviour of different ESC systems for yaw stability there are three types of popular ESC system [13] developed by industry and proposed by researchers, they are:

- Differential braking based ESC
- Differential torque based ESC
- Steer by wire based ESC

Differential braking and differential torque type ESC use the underlying principle of friction circle. The concept of friction circle of a tire is briefly explained here. The relationship between longitudinal and lateral forces of the tire can also be explained by this friction circle [20]. If $F_Z$ is the normal load on the tire and $\mu$ is the coefficient of friction for tire road surface, then the maximum friction force, $F_{friction} = \mu F_Z$. This friction force is used in the longitudinal or lateral direction or a combination of both. If $F_{brake}$ is the braking force and $F_{lateral}$ is the lateral force, then the resultant force $F_{result}$ of these will be equal to the friction force $F_{friction}$ or $\mu F_Z$ with a different direction as shown in the Figure 3.3.

![Figure 3.3 Friction circle concept](image)

Longitudinal and lateral forces of the wheel are coupled while vehicle is cornering. When longitudinal force is minimal then maximum lateral force can be achieved from these coupled forces. In the opposite way, when longitudinal force reaches maximum then the lateral force becomes minimal. If the braking force $F_{brake}$ is increased by applying brakes then the direction of the resultant force will change and that will generate less lateral force which can be used to change the yaw of the vehicle.
3.4.1 Differential braking type ESC

Yaw stability using differential braking is the most popular ESC. Yaw stability control using differential braking stabilizes the vehicle by controlling the slip of each wheel to correct the yaw moment of vehicle.

In this type of ESC, four individual brakes are used on the four wheels to distribute braking torque among them to correct yaw of the vehicle, so that the vehicle remains on the desired path. In a differential braking system, by increasing the brake pressure on the left wheels compared to the right wheels, a corrective counter clockwise yaw moment is generated and by increasing the brake pressure on the right wheels compared to the left wheels a corrective clockwise yaw moment is generated [13]. The key point of differential braking ESC is finding the difference between the actual and desired yaw and to correct the yaw using differential braking in a closed loop control system. This closed loop systems also includes vehicle slip angle. Both the yaw rate and vehicle slip angle are used here because the controlled variables in this closed loop system are limited by the coefficient of friction of road [11].

Figure 3.4 Differential braking ESC manoeuvre (a) over-steering (b) under-steering
To observe the behaviour of the vehicle dynamics, the yaw rate sensor, steering wheel sensor, lateral acceleration sensor and brake pressure sensor are used here. Differential brake pressure is determined for each wheel and applied to each wheel using a closed loop break pressure control system [21, 22]. This differential braking creates a net torque to track the determined yaw torque via the controller. Automatic turning of the vehicle to correct the direction is accomplished by uneven braking rather than the steering wheel. This breaking method is applied for both the over-steering and under-steering manoeuvre [23].

If the vehicle travels at high speed while entering a left curve, its rear wheels would slide on the sideways or spin out as shown in Figure 3.4 (a). In case of over-steering, the on board ESC system will detect the high yaw rate (quick change of direction) of the vehicle which will detract from the driver’s intended path. Then ESC will automatically apply the brake for a fraction of a moment to the front outer side wheel. It will reduce the potential lateral force of that tire and will generate corrective yaw moment.

In the other situation shown in Figure 3.4 (b) the vehicle is in the same situation as mentioned above but sliding occurs at the front. This means that the vehicle direction is not changing as quickly as expected by the driver and the ESC system will detect this incorrect direction. It will then apply brake force on the left inner side wheel. This reduces the lateral force and generates corrective yaw moment so that the vehicle turns towards the intended path. ESC cannot increase the available traction but it can help driver to keep the vehicle on the road in an emergency manoeuvre.

### 3.4.2 Steer by wire control method

Instead of a direct mechanical connection between the steering wheel and axle, a steer by wire system has two mechanically separate parts or sub systems for the steering as shown in Figure 3.5 (a). The first part involves a steering wheel that gives the driver the feel of steering as well as reading the steering angle. The rest of the system is for actuation according to the driver’s input angle given to first part.
The second part of the steer-by-wire system is responsible for vehicle stability in sudden manoeuvres while over-steering or under-steering.

A steer-by-wire based ESC system modifies the driver’s steering input as shown in Figure 3.5 (b) for yaw stability control by skid prevention. The primary task of the driver is path following using the steering angle. This automatic intervention of steering angle modification is done without interfering with the vehicle’s response to the driver’s desired path [13].

![Figure 3.5 (a) Steer-by-wire steering system (b) Steer-by-wire ESC](image)

By steering, the driver excites the yaw rate by lateral acceleration \(\alpha_y\) to the mass of the vehicle \(m\) to keep the vehicle tangential to the desired path [24]. The yaw rate of the vehicle is also excited by disturbance torque \(M_2D\) which is the result of road friction on the tire. The driver’s secondary task is to compensate for this disturbance torque by counter steering. The yaw motion is dependent on the driver’s desired lateral acceleration, and also depends on induced disturbance torque by the tire and road friction coefficients. In steer by wire ESC disturbance attenuation is done by decoupling in such a way that it does not affect the driver’s ability to follow the primary path.
The first key point is removing the influence of yaw rate from lateral acceleration \( \alpha_{yp} \). Instead of C.G a point P is chosen between C.G and front axle so that lateral acceleration \( \alpha_{yp} \) at P does not depend on lateral force at rear axle \( F_{yr} \).

The second key point is to design the controller in such a way so that yaw rate \( \Psi \) does not influence front tire slip angle \( \alpha_f \) and thus yaw rate \( \Psi \) does not influence lateral acceleration \( \alpha_{yp} \).

In designing a controller, the steering angle input of the driver, the steering angle input registered by the disturbance attenuation control system, vehicle slip angle and yaw rate are considered. Steer by wire offers unprecedented flexibility in vehicle handling. It requires an actuator for steering control. This method is simplified by assuming that the velocity is constant or slowly varying during steering and small yaw rate. This indicates a limitation for this method. Also this method is complicated and costly to implement.

### 3.4.3 Differential torque distribution type ESC

Differential torque distribution type ESC can be used in vehicles with an independent drive torque in each wheel. In the case of ICE four-wheel-drive (4WD) vehicles, all wheels can be permanently linked to the engine. This is called All-wheel-drive (AWD). Or one of two axles can be linked to the engine [25]. Engine torque can be transmitted by using inter-axle differentials or clutches.

In four wheel drive systems yaw moment in the opposite direction of the turn can be generated using side to side torque transfer. Applying differential clutch torque to the turning vehicle, torque can be transferred from the faster wheel to the slower wheel that generates a yaw moment in the opposite direction of the turn. Like the ICE vehicle, if a central electric motor is used for the drive train of the electric vehicle, it can be considered as a conventional four wheel drive or all-wheel drive. In an all-wheel drive system, twin clutch torque biasing differentials can be used to transfer torque to inner or outer wheels [26]. Using centre differential, this torque can be transferred between the front and rear wheels. Differential torque distribution from a centralized electric motor using clutch and gear is very complicated.
In the case of over-steering, the direction of the vehicle can be corrected by using left-to-right torque vectoring on front axle to transmit greater torque to the inner side wheel as shown in Figure 3.6(a). In case of under-steering at the limit of adhesion, direction can be corrected by using left-to-right torque vectoring on the rear axle to transmit greater torque to the outer side wheel. Figure 3.6(b) shows the under-steering manoeuvre using ESC.

In front-to-rear torque vectoring mode transmitted torque to the front axle can correct the over-steering, and transmitted torque to the rear axle can correct an under-steering manoeuvre [12]. This front-to-rear torque vectoring also controls the lateral potential forces in each case and helps the vehicle travel in the intended direction.
3.5 Comparison of different types of ESC focusing on four in-wheel EV

Identification of unexpected behaviours of the vehicle and the generation of control signals for the actuators are both done by the ESC system. These actuators apply corrective forces on the wheels to change the yaw to maintain yaw stability. In some cases the controller applies corrective steering wheel for yaw stability. Below we discuss how to select a most suitable type ESC for a 4 in-wheel EV from three basic types of ESC.

3.5.1 Required sensor

If required sensors are taken into account, any of the basic types of ESC can be used in the proposed vehicle. In the case of a steer by wire ESC system, multiple lateral acceleration sensors are required [24]. Differential torque based ESC requires a sensor to the wheel speed and electrical characteristics of a motors. The proposed vehicle has 4 in-wheel motors where each wheel can give feedback of the wheel speed via it's built in Hall Effect sensors. Electrical characteristics can be measured from the circuit. So, in a four in-wheel EV a differential torque based ESC system may not require more sensors than the other two types of ESC. Thus it is the preferred type.

3.5.2 Required actuators

If the required actuators are taken into account, then a steer by wire based ESC can be chosen because in this system actuation is done only via the steering wheel. To make an indirect connection between steering wheel and axle two electric motors are used and they work as actuators. These are additional components compared to the proposed EV. The proposed EV has 4 in wheel motors as the existing drive train and these can be used as actuators for differential torque based ESC.
3.5.3 Complexity and performance

Implementation of steer a by wire ESC system is very complex. To simplify we assume that the velocity is constant or slowly varying during steering. A small yaw rate shows the limits of this method.

A differential Braking type ESC requires measurement and control of brake pressure which requires extra components. This method may not be chosen as it can create energy loss. Also it slows down the vehicle while differential braking ESC is active [27]. On the other hand using a differential torque type ESC the driver can get the expected longitudinal response while it is active. Determination of parameters is easier from existing EV components.

It can be concluded that a differential torque type ESC would be more suitable for this in-wheel EV. A differential torque based ESC system may not be able to generate adequate yaw rate to respond the driver’s needs when the coefficient of road friction was small or if the vehicle speed was too high. As the yaw moment is created by generating different torque in different wheels, the possibility of achieving the expected yaw rate may not be the nominal, but it is possible to make the yaw rate closer to the nominal yaw rate and yaw can be partially controlled.

The challenge is how a differential torque based ESC will work for in-wheel EV so that it can overcome the limitation of torque generation for adequate yaw moment. The approximate yaw rate can be achieved by operating the electric motors to a higher voltage than normal.

3.5.4 Proposed ESC type

To select a certain type of ESC from the basic types or to improve a previous one, it is important to focus on available resources and feasibilities in the proposed EV as well as to focus on the improvement of the proposed ESC type. Considering the available resources and the technical feasibility of controlling individual motors of in-wheel EV,
and the requirement for keeping expected longitudinal response by the driver in controlling stability, differential torque based ESC may be chosen with some changes to the working principle or algorithm. More focus is given in this proposed ESC type to find possible ways for controlling the force of four wheels so that it can generate corrective yaw moment.

Using sensors and a dynamic vehicle model, the in-Wheel ESC finds the required longitudinal force in each wheel for a corrective yaw moment then generates the torque accordingly.

In-wheel ESC will determine the required differential longitudinal force difference $\Delta F_x$ between the wheels to generate corrective yaw moment. Then the controller will try to create that longitudinal force difference by generating torque in each individual wheel. This can be done by manipulating the supply voltage and current for the in-wheel motor automatically via the intelligent controller.

To create the corrective force difference $\Delta F_x$, the individually required torque should remain within the limits of possible torque generation by this electric motor. If $\Delta F_x$ is large enough and it requires torque beyond the capacity of the electric motor, then the torque distribution system may not provide adequate yaw moment.

In practice, besides the normal operating voltage and current there is a higher level voltage and current needed for running the in-wheel motors in turbo mode. In turbo mode, the limit of the operating voltage increases to a higher level and motors generate higher torque than they usually do in the normal operation mode. This voltage level shifting solves the problem of generating required differential torque by these in-wheel motors.

### 3.6 ESC regulations

It is well known that ESC can prevent a single vehicle crash by assisting drivers to avoid loss of control. Because the number of crashes remains high, transport authorities in different countries are using legislation to have ESC fitted to the new cars. All
vehicles have to have ESC fitted when sold in USA from 1st September 2012. The Australian Government also introduced the ESC as a safety standard and should be fitted to all passenger vehicles from 1st November 2011 [28].

The United Nations announced the regulations in Global technical regulation No. 8 for ESC systems in vehicles [29]. This is a proposal to develop a global technical regulation concerning ESC systems and recommendations on the development of a global technical regulation concerning ESC for light vehicles. These regulations included the vehicle performance requirements with ESC.

To speed up the ESC fittings to vehicles, national Highway Traffic Safety Administration (NHTSA) has introduced standards listed in FMVSS 126 for a vehicle equipped with ESC. It also includes the test procedure to ESC.

FMVSS 126 has listed the vehicle performance requirements to be measured in an open loop dynamic manoeuvre [30]. A test where the vehicle will have a steering and a counter steering sequence with progressively greater steering input angles. This manoeuvre is called “Sine with Dwell” which is used to evaluate the ESC performance. The test is conducted using 0.7 Hz sine wave steering input with a delay of 500 milliseconds between the 3rd and 4th quarter of cycle to make a dwell. Figure 3.7 shows the Sine dwell steering angle profile [31]. Before performing this test the vehicle should be travelling at 80 km/h and then the steering angle input increased slowly until the lateral acceleration reaches to 0.3 g.

If the lateral acceleration of the vehicle reaches 0.3 g for the steering angle $\delta$ then the steering angle input for Sine dwell will start from $1.5 \times \delta$ and increase by 0.5 in the next steps until the steering angle reaches 270 degrees. This test assesses the performance of the vehicle based on lateral stability.
The vehicle passes the test when it passes the three conditions given below:

- After the start of Sine dwell at time $T=1.07$ sec, if the steering angle $\delta$ gain is 0.5 or more then the lateral displacement of vehicle centre of gravity must be 1.83 m or greater for a vehicle of 3500 kg or less, and a lateral displacement of 1.52 m for vehicle weighting more than 3500 kg.

- At time $T=2.93$ (1 sec after steering input stops) the instant yaw rate is 35% of the peak yaw rate or less.

- At time $T=3.68$ (1.75 sec after the steering input stops) the instant yaw rate is 20% of the peak yaw rate or less.

Generally, when an ESC system detects a high yaw rate then it helps the driver to control the directional stability by using braking intervention in each wheel to correct the yaw moment.

This test involves a special test facility for the accurate input of steering data with respect to time.
Other dynamic tests for electronic stability controls are given below.

**NHTSA Fishhook test**

The Fishhook Manoeuver is a dynamic test developed NHTSA to understand how vehicles rollover. The Fishhook test involves the programmable steering controller input the hand wheel commands shown in Figure 3.8.

![Fishhook manoeuver description](image)

**Figure 3.8 Fishhook manoeuver description**

**ISO Lane Change Test**

This lane change manoeuvre is a test procedure which involves of driving a vehicle through a set track. This is also known as a double lane-change manoeuver. It is widely used to evaluate vehicle handling and safety. Figure 3.9 shows an example setup of the road for lane change procedure.
These test methods may not be the only possible methods to test the stability of the vehicle. Vehicle state or stability status can be identified by comparing the expected values of yaw rate and slip angle of the vehicle. In a field test actual vehicle dynamics can be compared with the nominal values of yaw rate, body slip angle and lateral displacement to observe the stability of the vehicle.

3.7 Summary

Stability control concept and its basic operations are included in the initial part of this chapter. The effectiveness of ESC was discussed in this chapter based on crash data analysis and human factors analysis. Then different types of stability control techniques are discussed here with a view to finding the required type of ESC suitable a four in-wheel vehicle. A comparison of different types of ESC, focusing on four in-wheels EV, is done here considering sensors, actuators, complexity and challenges. A proposed method based on differential torque is described here which can be used in a four in-wheel vehicle with required modifications. At the end of this chapter, we discuss testing and regulation of ESC. In the next chapter, we investigate a differential torque based stability controller by reviewing previously adopted techniques.
Chapter 4
Investigation into differential torque based ESC systems and proposed method
4 Investigation into differential torque based ESC systems and proposed method

4.1 Introduction

This chapter presents analysis of differential torque based ESC for in-wheel-motor electric vehicles (EV). Discussion on wheel torque based ESC is done here based on the paper “An investigation into differential torque based strategies for ESC in an in-wheel electric vehicle” [32]. There are several technical challenges [33] associated with the development of ESC. To find a suitable and effective differential torque based ESC, several torque based strategies have been investigated. We consider parameters used in control, control strategies, vehicle drive train and feasibility of implementation.

When we focus on the differential torque based ESC for electric vehicles, we find several modern techniques which use a centralized electric motor or multiple electric motors. A number of techniques for differential torque based ESC are being in EV research. In this paper we investigate different torque based ESC techniques for EV and then develops a suitable ESC for an in-wheel EV.

ESC based on differential braking is popular and available in most vehicles. In a differential braking type ESC, differential brake force in each wheel is applied for control vehicle stability. Similar stability enhancement can be achieved by using independent torque control in each wheel. Differential braking based ESC has the demerit of slowing down the vehicle, and drivers do not get the expected longitudinal response. To avoid this problem a differential torque based ESC can be used for the same purpose. The purpose of a differential torque ESC is to find the difference between the actual and desired yaw and correcting the yaw using differential torque in a closed loop control system. Differential torque distribution type ESC can be used for vehicles where independent drive torque in each wheels can be applied.
Differential torque distribution from a centralized electric motor or engine using clutch and gear is very complicated. In the case of four-wheel-drive (4WD) vehicle all wheels can be permanently linked to the engine (this is called All-wheel-drive AWD), or one of two axles can be linked to the engine [25]. Engine torque can be transmitted in this driveline by using inter-axle differentials or clutches. In four wheel drive systems yaw moment in the opposite direction to the turn can be generated using side to side torque transfer. Applying differential clutch torque to the turning vehicle, torque can be transferred from faster wheel to slower wheel generating a yaw moment in the opposite direction to the turn. In an all-wheel drive system, twin clutch torque biasing differentials can be used to transfer torque to inner or outer wheels [13]. Using a center differential, torque can be transferred between front and rear wheel. These differentials show a higher degree of variation for torque distribution, but can provide the precise required torque on a consistent basis.

In this research we investigate several differential torque based ESC systems considering these parameters: control strategy, control law and implementation complexity. The overall idea of the proposed system is included in the later sections of this chapter.

4.2 Related parameters and estimations

Research has been conducted on differential torque based stability controllers focusing on independent wheel control. Vehicle stability can be improved by controlling the driving force of the wheels which is the general concept of differential torque based ESC.

It is essential to know the required methods to determine the corrective wheel torque. Vehicle body dynamic forces are important parameters in ESC design and the value of these parameters is achieved using a vehicle model. Similarly to determine the tire forces a tire model is required. All these parameters such as vehicle body forces, tire forces, vehicle slip, yaw rate and coefficient of frictions are required for ESC.
It is important to discuss how these parameters are determined or estimated based on the previous research work. We discuss these parameters and their estimation methods below.

### 4.2.1 Vehicle dynamics

Though different methods of finding wheel torque are adopted in different research for ESC, the vehicle model is similar in most of the research works. Difference can be seen in the degree of freedom of the vehicle model. The vehicle model is required for the simulation to observe the responses of the vehicle body. This vehicle body model can then be used as a plant vehicle. A vehicle model with seven degree of freedom [34] is proposed for an all-wheel-drive vehicle. In other research work [35-37] a vehicle model is employed for a four-wheel-drive electric vehicle with the equation of motion of vehicle dynamics.

A simulation result is presented using Carmaker software [36]. Vehicle models are used in [38, 39] in view of a four-in-wheel-motor electric vehicle where the equations of vehicle dynamic are used for the observation of vehicle responses. Simulation then is done in the MATLAB/Simulink environment. The research work above adopted a simple vehicle model using equation of motion for the vehicle dynamics including steering angle, longitudinal vehicle velocity, and lateral vehicle velocity and yaw rate.

A numerical model is generated in [40] using MSC.ADAMS/Car for four in-wheel EV where electric motors are used for the power train. Though a detailed nonlinear model vehicle model is used for simulation in this research, a simplified vehicle model is considered for designing the controller. veDYNA is used in [41] for a four-wheel-drive vehicle along with MATLAB/Simulink based virtual controller and virtual instrument LabVIEW.

After designing the ESC controller, simulation can be done using numerical simulation software like CarSim or ADAMS/Car for detailed observation. A built-in database for standard passenger vehicles has made CarSim popular simulation software.
In the case of a four in-wheel motor EV developed in a laboratory presented in this research work, a built-in database may not help unless a customized model is used which does not fill the database with experimental data. Vehicle models available in different simulation software tools may not be entirely usable in this research. Also a controller designed to control such vehicles may not be available in these software systems to avoid co-simulation. Specific software with built in reconfigurable models and control methods from SIMULINK or from other software tools show some constraints in model definition, defining input and output variables, inputs and outputs within the required structure.

A vehicle model is required for designing an ESC controller which can provide vehicle responses such as longitudinal, lateral and yaw. To keep the model simple some of these research works have neglected the influence of aerodynamic drag, rolling resistance and suspension systems. As vehicle dynamics are much slower compared to the control loop [21], a simple vehicle model would be easier to implement. For analysis computer based simulations the vehicle model can be based on the empirical equations or an experimental data source, but a reference vehicle model needs to be hosted by an embedded controller. This requires a simple vehicle model.

According to the Newton’s second law the sum of the external forces acting on a body in a given direction is equal to the product of its mass and its acceleration.

\[ \sum F = m \cdot a \]  

(4.1)

Where,

\( F \) is force
\( m \) is mass
\( a \) is acceleration

This fundamental law of motion and with the geometric relationships, longitudinal velocity \( x \), lateral velocity \( y \), and yaw rate \( \Psi \) can be measured. The sum of the external forces acting on the vehicle body in the longitudinal and lateral axes is equal to the product of the vehicle mass and the acceleration.
This also includes the sum of torques acting on the vehicle body this is equal to the moment of inertia time rotational acceleration around the vehicle axis.

![Simplified vehicle model for analysis of transient motions](image)

**Figure 4.1 Simplified vehicle model for analysis of transient motions**

To handle the vehicle during a manoeuvre we require the equations of motion of the vehicle related to the transient state of the vehicle. This transient state is a state between the time of steering input by the driver and the time when the vehicle reaches steady-state motion. Vehicle handling differs qualitatively in the transient state of the vehicle. In the transient state the vehicle has responds quickly with minimum of oscillation while approaching steady-state motion. To analyse the transient response, the inertia properties of the vehicle are taken into consideration. During a turning manoeuvre, the vehicle undergoes both translation and rotation. To describe its motion, a set of axes fixed to and moving with the vehicle body is considered.
Equation of motion when the vehicle having plane motion is given here referring to Figure 4.1.

\[ m(\dot{x} - \dot{y} \dot{\Psi}) = F_{xf} \cos \delta + F_{xr} - F_{yf} \sin \delta \]  \hspace{1cm} (4.2)

\[ m(\dot{y} - \dot{x} \dot{\Psi}) = F_{yf} \cos \delta + F_{yr} + F_{xf} \sin \delta \]  \hspace{1cm} (4.3)

\[ l_z \dot{\Psi} = l_f F_{yf} \cos \delta - l_r F_{yf} + l_f F_{xf} \sin \delta \]  \hspace{1cm} (4.4)

where,

Front wheel steering angle is \( \delta \)

Longitudinal tire force for front wheels is \( F_{xf} \)

Longitudinal tire force for rear wheels is \( F_{xr} \)

Lateral tire force for front wheels is \( F_{yf} \)

Lateral tire force for rear wheels is \( F_{yr} \)

Mass of the vehicle \( m \)

Distance of front axle from CoG is \( l_f \)

Distance of rear axle from CoG is \( l_r \)

Yaw moment of inertia is \( l_z \)

Longitudinal velocity of vehicle is \( \dot{x} \)

Lateral velocity of vehicle is \( \dot{y} \)

Yaw rate is \( \dot{\Psi} \)

The formulation of these equations of motion is described in [42]. Here, the vehicle body is assumed to be symmetrical about the longitudinal plane, and the roll motion of the vehicle body is ignored. Using the values of different parameters of the vehicle like steering angle, and forces with the values of initial conditions of the vehicle the equations (4.2), (4.3) and (4.4) can be solved. By solving these differential equations yaw rate and lateral velocity of the vehicle can be determined.

As we are focusing four in-wheel-motors vehicle, we can use the individual wheel forces of the vehicle. Then equation (4.2), (4.3) and (4.4) can be re-written as given in [13].
Equations of motions of a vehicle body are re-written as:

\[ m\ddot{x} = (F_{xf1} + F_{xf2})\cos(\delta) + F_{xrt} + F_{xrr} - (F_{yfl} + F_{yfr})\sin(\delta) + m\Psi\dot{y} \]  \hspace{1cm} (4.5)  

\[ m\ddot{y} = F_{yrl} + F_{yrr} + (F_{xf1} + F_{xf2})\sin(\delta) + (F_{yfl} + F_{yfr})\cos(\delta) - m\Psi\dot{x} \]  \hspace{1cm} (4.6)  

\[ I_z\ddot{\Psi} = l_f(F_{xf1} + F_{xf2})\sin(\delta) + l_f(F_{yfl} + F_{yfr})\cos(\delta) - l_r(F_{yrl} + F_{yrr}) + \frac{l_w}{2}(F_{xf1} - F_{xf2})\cos(\delta) + \frac{l_w}{2}(F_{xrr} - F_{xrt}) + \frac{l_w}{2}(F_{yfr} - F_{yfl})\sin(\delta) \]  \hspace{1cm} (4.7)  

Where

Front wheel steering angle is \( \delta \)

Longitudinal tire force for:

Front left is \( F_{xf1} \)

Front right is \( F_{xf2} \)

Rear left is \( F_{xrl} \)

Rear right is \( F_{xrr} \)

Lateral tire force for:

Front left is \( F_{yfl} \)

Front right is \( F_{yfr} \)

Front left is \( F_{yrl} \)

Rear right is \( F_{yrr} \)

Mass of the vehicle \( m \)

Distance of front axle from CoG is \( l_f \)

Distance of rear axle from CoG is \( l_r \)

Distance between left and right wheels track length is \( l_w \)

Yaw moment of inertia is \( I_z \)

Longitudinal velocity of vehicle is \( \dot{x} \)

Lateral velocity of vehicle is \( \dot{y} \)

Yaw rate is \( \dot{\Psi} \)
Using equations (4.5), (4.6) and (4.7) vehicle motions can be determined after calculating wheel forces. Wheel models are discussed in the next section which facilitates calculation of the wheel forces.

4.2.2 Tire force

Tire forces and moments acting on the vehicle are significant parameters to determine as they strongly influence the dynamics of the vehicle. From equations (4.5), (4.6) and (4.7) we see that, individual tire forces are required to calculate the vehicle dynamics. To determine the tire forces, complex and non-linear tire models like Pacejka's Magic Formula, or Dugoff tire models are used in different vehicle simulations. For accuracy in simulation results the tire model must be accurate. Pacejka’s Magic Formula tire model described in [43] has been used in [35] where the model provides calculations for longitudinal and lateral force relation, and aligning moment. Pacejka’s Magic Formula tire model is used for the same purpose in [39, 44] for a perfect vehicle simulation result. This Magic Formula is a kind of experimental method which can derive tire road surface forces. This is a semi-empirical model developed using experimental data. This model is comparatively difficult to use as it requires many experimental coefficients.

To provide an overview this tire model is presented here briefly and a detailed description is available in “Tire and Vehicle Dynamics” [45].

According to the model, generated lateral or longitudinal force \( Y \) can be expressed as a function of the input variable \( X \) as,

\[
Y(X) = y(x) + S_Y
\]  

(4.8)

with

\[
y = D \sin[C \arctan(B_x - E(B_x - \arctan B_x))] \]

(4.9)

\[
x = X + S_H
\]

(4.10)

Where,

\( Y \) is the output variable; it can be \( F_x, F_y \) or \( M_z \)

\( X \) is the input variable; it can be \( \alpha \) or \( \sigma_x \)

\( \alpha \) is slip angle
\( \sigma_x \) is slip ratio
\( F_x \) is longitudinal force
\( F_y \) is lateral force
\( M_z \) is aligning moment
\( B \) is stiffness factor
\( C \) is shape factor
\( D \) is peak value
\( E \) is curvature factor
\( S_H \) is horizontal shift
\( S_V \) is vertical shift

This formula is capable of producing characteristics curves similar to the measured \( F_x \), \( F_y \). This tire model is not used for calculation or simulation in this research work because it is over complex.

Dugoff’s tire model is also popular and it is used in [34, 36] for vehicle simulation and tire force calculation. This model also describes the relation between tire forces with tire slip and the coefficient of friction using a function.
Dugoff's model calculates the combined longitudinal and lateral tire force using the friction circle concept. In this model vertical pressure distribution is assumed to be uniform on the tire contact patch. The model offers a significant advantage [46] by using independent values of tire cornering stiffness $C_\alpha$ and longitudinal stiffness $C_\sigma$. In Figure 4.2 wheel parameters are shown to describe the required parameter calculation.

To calculate the wheel forces let,
- Effective radius of the tire be $r_{eff}$
- Wheel angular velocity be $\omega_w$
- Longitudinal vehicle velocity be $\dot{x}$
- Lateral vehicle velocity be $\dot{y}$
- Longitudinal slip ratio be $\sigma$
- Tire slip angles be $\alpha$
- Front tire slip angles be $\alpha_f$
- Rear tire slip angles be $\alpha_r$
- Tire road friction coefficient be $\mu$
- Distance of rear axle from CoG be $l_r$
- Distance of front axle from CoG be $l_f$
- Yaw rate be $\Psi$
- Cornering stiffness of the tire be $C_\alpha$
- Longitudinal stiffness of the tire be $C_\sigma$
- Vertical force on tire be $F_z$
- Function variable be $\lambda$
- Longitudinal tire force be $F_x$
- Lateral tire force be $F_y$

Dugoff’s tire model provides longitudinal and lateral forces as output. To calculate the longitudinal and lateral forces in this model, it requires estimating:

- Wheel slip ratio
- Wheel slip angle.
Slip ratio

The difference between longitudinal velocity \( \dot{x} \) and the equivalent rotational velocity of wheel is called the longitudinal slip. Longitudinal slip ratios for each wheel for braking are

\[
\sigma = \frac{r_{eff} \omega_w - \dot{x}}{\dot{x}}
\]  
\[ (4.11) \]

And for longitudinal slip ratios for each wheel for acceleration are

\[
\sigma = \frac{r_{eff} \omega_w - \dot{x}}{r_{eff} \omega_w}
\]  
\[ (4.12) \]

Slip angle

Slip angles at the front tire is

\[
\alpha_f = \delta - \frac{\dot{y} + l \Psi}{\dot{x}}
\]  
\[ (4.13) \]

And Slip angles at the rear tire is

\[
\alpha_r = -\frac{\dot{y} - l \Psi}{\dot{x}}
\]  
\[ (4.14) \]

Function of \( \lambda \)

A function is used in Dugoff’s tire model with a variable \( \lambda \) in the wheel model which is calculated in a separate sub-block. Finally, with all these sub-blocks together we calculate the longitudinal or lateral wheel forces. The variable

\[
\lambda = \frac{\mu F_z (1 + \sigma)}{2 [ (c_\alpha \sigma)^2 + (c_\alpha tan \alpha)^2 ]^{\frac{3}{2}}}
\]  
\[ (4.15) \]

And function \( f(\lambda) = (2 - \lambda) \lambda \) If value of \( \lambda < 1 \) then function

Or, function \( f(\lambda) = 1 \) If value of \( \lambda \geq 1 \)
According to Dugoff’s tire model:

Longitudinal tire force $F_x = C_\sigma \frac{\sigma}{1+\sigma} f(\lambda) \quad (4.16)$

Lateral tire force $F_y = C_\sigma \frac{\tan \alpha}{1+\sigma} f(\lambda) \quad (4.17)$

In the case of Dugoff’s tire model, it requires fewer coefficients compared to the Magic Formula, to calculate longitudinal and lateral forces. A practical approach has been taken into account for a differential torque based ESC where complexity can be avoided by using fewer of coefficients. Though both of these tire models are very popular and accurate, preference is given to Duggoff’s tire model because it is less complex when creating a reference tire model in computer simulation, and for hosting by an embedded controller.

4.2.3 Control law parameters estimations

It is important to identify the required control parameters for a yaw stability controller. If the controller is designed using only yaw velocity as the control variable to maintain yaw rate equal to or close to the nominal value, it would fail to stabilize the vehicle on the road when tire road adhesion is very low on a slippery road. If the tire road coefficient is very low then the vehicle slip angle might increase rapidly without increasing or decreasing lateral force. This because on the slippery road yaw velocity and lateral acceleration do not correspond to each other [47]. Steer-ability of the vehicle depends on the side slip angle of the vehicle [48]. Steering angle input cannot change the direction of a vehicle if the side slip angle increases. The analysis in the paper [48] shows that on a dry surface vehicle steer-ability vanishes when the vehicle slip angle is larger than 10 degrees, and on an icy surface vehicle steer-ability vanishes when the vehicle slip angle is less than 4 degrees. So controlling yaw velocity or yaw rate only may not achieve stability of the vehicle in a critical situation.
To keep the yaw velocity and slip angle of the vehicle limited to the values that correspond to coefficient of friction of the road, the controller should take both the yaw velocity and slip angle as controller parameters [40, 47]. The majority of researchers have used similar control parameters like yaw rate and vehicle slip in control laws. Researchers in [35-37], [39] and [40] have used both the yaw rate and vehicle slip angle as their control parameters in an ESC controller, regardless of the types of the control law. This concept of minimizing the error of yaw rate and slip angle simultaneously in the control law is a more logical approach as they have to be limited to values which correspond to the tire road coefficient friction.

4.2.3.1 Yaw rate estimation

The primary objective of the yaw stability controller is to maintain the yaw rate close to the nominal value when the vehicle is unstable. If the controller is designed using yaw velocity as the control variable to maintain yaw rate equal to or close to the nominal value it would fail to stabilize the vehicle on the road where tire road adhesion is very low or on a slippery road. If the tire road coefficient is very low then the vehicle slip angle might increase rapidly without increasing or decreasing lateral force because on the slippery road, yaw velocity and lateral acceleration do not correspond to each other [47]. Steer ability of the vehicle depends on the side slip angle of the vehicle [48], steering angle input cannot change the direction of a vehicle if the side slip angle increases. The analysis in the paper [48] shows that on dry a surface vehicle steer-ability vanishes when the vehicle slip angle is larger than 10 degrees and on icy surfaces vehicle steer-ability vanishes when the vehicle slip angle is less than 4 degrees. So controlling yaw velocity or yaw rate only may not achieve stability of the vehicle in a critical situation. To keep the yaw velocity and slip angle of the vehicle limited to the values that correspond to coefficient of friction of the road, the controller should take both the yaw velocity and slip angle as controlled variables [40, 47].

Yaw rate of a vehicle is simply the change of angular direction in a time unit. If a vehicle changes its steering input about 180 degrees from north to south within 10 seconds then the yaw rate of the vehicle can be said 18 degree/ sec.
This is the actual yaw rate of a vehicle which can be measured directly using electronic yaw rate sensors. In the controller the difference between the actual yaw rate and desired yaw rate is used in most of the cases. Desired yaw rate has to be determined from the steady state relation between the steering angle and its generated radius of the vehicle’s trajectory. The desired yaw rate can be determined from the steering angle, radius, vehicle speed, vehicle mass, tire stiffness and vehicle’s physical dimensions. Desired yaw rate is calculated here in the control law for simulation.

\[
\Psi_{\text{desired}} = \frac{\dot{x}}{R} \quad (4.18)
\]

Where,
\[R\] is the radius of the circular road calculated using steering angle
\[\dot{x}\] is the longitudinal velocity.

Determining the radius \(R\) of the circular road, we need to consider the idea of steady state turning. Steady state turn is the turn taken at a constant speed and having a constant turn radius. A detail discussion on the relationship between the radius of the path and steady state steering angle is provided in [49]. A more simplified determination is given considering understeer gradient is discussed in [13]. The relation between steady state steering angle \(\delta_{ss}\) for a circular path and radius \(R\) of the path is

\[
\delta_{ss} = \frac{l_f + l_r}{R} + \left(\frac{m l_r C_{ar} - m l_f C_{af}}{2 C_{af} C_{ar} (l_f + l_r)}\right) \frac{\dot{x}}{R} \quad (4.19)
\]

Steady state steering angle \(\delta_{ss}\) is used in finding desired slip angle.

Where,
\(C_{af}\) and \(C_{ar}\) is the cornering stiffness of the front tire and rear tire respectively.

Vehicle mass is \(m\).
Distance from CoG to the front axle is \(l_f\).
Distance from CoG to the rear axle is \(l_r\).
Radius $R$ is determined from steady state cornering equations as given in [50]. Radius $R$ is determined as

$$R = \frac{mv^2}{F_y} \tag{4.20}$$

Where, $R$ is the radius of the circular road calculated using steering angle, $m$ is mass of the vehicle, $v$ is vehicle longitudinal velocity $F_y$ is lateral force on the vehicle.

### 4.2.3.2 Vehicle Side Slip angle

Vehicle body side slip angle is one of the most significant variables in vehicle stability. This is the angle between vehicle’s longitudinal axis and the direction of the velocity at the CoG. So, vehicle slip angle $\beta$ is can be determined as:

$$\beta = \frac{v_y}{v_x} \tag{4.21}$$

Where, $v_y$ is lateral velocity of the vehicle $v_x$ is longitudinal velocity of the vehicle

This side slip angle cannot be measured using any commercially available sensors. It is possible to measure using some optical sensors, but developing this kind of system is not cost effective. This can be used for development and test purposes rather than using it in the vehicle. In real time, a lateral accelerometer with a yaw rate sensor can be used to estimate the side slip angle of the vehicle. Output of a lateral accelerometer is linear lateral acceleration $a_y$ but with the effect of yaw rate. With the effect of yaw rate the output of the sensor is:

$$a_y = a_{y\:meter} - \Psi v_x \tag{4.22}$$
Where,

$a_y$ is lateral acceleration

$a_{y\_meter}$ is lateral acceleration from sensor

Lateral velocity $v_y$ can be found by integrating equation (4.22) which considers that the road surface is perfectly levelled. So, $v_y$ can be written as:

$$v_y = \int (a_{y\_meter} - \Psi v_x)dt$$  \hspace{1cm} (4.23)

The lateral velocity measurement is taken considering the imperfect level of a road surface and road the bank angle in paper [21]. This estimated lateral velocity is given below:

$$v_y = \frac{v_x m}{c_{af}+c_{ar}} \left(a_{y\_meter} - \frac{l_f c_{af} - l_r c_{ar}}{v_x} \Psi - \frac{c_{af}}{m} \delta \right)$$ \hspace{1cm} (4.24)

Where,

$c_{af}$ and $c_{ar}$ are the cornering stiffness of front tire and rear tire respectively.

Vehicle mass is $m$.

Distance from centre of gravity (CoG) to the front axle is $l_f$.

Distance from CoG to the rear axle is $l_r$.

Steering angle is $\delta$.

After measuring the longitudinal velocity of the vehicle and estimating the lateral velocity $v_y$ side slip angle $\beta$ can be determined from equation (4.21).

Desired side slip angle $\beta_{desired}$ can be obtained from the steady state steering angle $\delta_{ss}$ as given below.

$$\beta_{desired} = \frac{l_f \frac{mx^2}{2c_{ar}(l_f + l_r)}}{(l_f + l_r) + \frac{m^2 (l_r c_{ar} - l_f c_{af})}{2c_{af} c_{ar} (l_f + l_r)}} \delta_{ss}$$ \hspace{1cm} (4.25)
The actual or measured side slip angle needs to be compared with the nominal or desired side slip angle and the difference between them is used in the control law of ESC. Most of the researchers have used similar control parameters like yaw rate and vehicle slip in control laws. Some of the previous research into torque based ESC shows the use of different control variables. An effort is given here to realize those. In [44] a yaw control loop is used to control the yaw of the vehicle and employs yaw rate as the control variable. Another slip controller is used under the same yaw control loop to keep the slip ratios of the wheels within a stable region. This slip controller uses wheel slips and wheel angular velocity to make a total of three control variables. A yaw controller using yaw rate and side slip angle as control variables design is shown on [38]. An active front steering controller using the yaw rate as control variable is integrated with the yaw controller. We use the variable in more than one controller to achieve more robustness.

Other researchers in [35-37], [39] and [27] have used both the yaw rate and vehicle slip angle as their control parameters in an ESC controller, regardless of the types of control law. The concept of minimizing the error of yaw rate and slip angle simultaneously in the control law is a more logical approach as they have to be limited to values which correspond to the tire road coefficient friction.

An exception in control parameter can be seen in [34] where lateral acceleration is chosen along with yaw rate as a control variable to regulate the heading of the vehicle. In this paper the difficulty of measuring individual tire force and coefficient of friction is avoided by using measurable control variables from the vehicle like, yaw rate and lateral acceleration. Using this measurable variable is better than using estimated variables if it can provide the required parameter. Here it provides the mean longitudinal tire slip. In this research work the controller can provide the output as front to rear ratio and left to right ratio of torque.
4.2.3.3 Coefficient of friction estimation

Tire road friction coefficient is a complex phenomenon of tire road interaction. In vehicle dynamics it is an important parameter that helps to calculate the tire force and also helps to describe the condition of the road. Table 4-1 [51] shows the different frictional coefficient for different roads used by a passenger car. The adhesive capability a between the tire and ground plays a vital role in vehicle dynamics. Due to this adhesive capability vehicle moves forward when there is any tractive effort on the wheel. The maximum tractive effort that can be supported by the tire-ground contact patch is the product of vehicle load and the coefficient of road adhesion or tire-road frictional coefficient denoted as $\mu$. For further increase in tractive effort will cause sliding of the wheel.

<table>
<thead>
<tr>
<th>Surface</th>
<th>Peak Values, $\mu_p$</th>
<th>Sliding Values, $\mu_s$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Asphalt and concrete (dry)</td>
<td>0.8–0.9</td>
<td>0.75</td>
</tr>
<tr>
<td>Concrete (wet)</td>
<td>0.8</td>
<td>0.7</td>
</tr>
<tr>
<td>Asphalt (wet)</td>
<td>0.5–0.7</td>
<td>0.45–0.6</td>
</tr>
<tr>
<td>Grave</td>
<td>0.6</td>
<td>0.55</td>
</tr>
<tr>
<td>Earth road (dry)</td>
<td>0.68</td>
<td>0.65</td>
</tr>
<tr>
<td>Earth road (wet)</td>
<td>0.55</td>
<td>0.4–0.5</td>
</tr>
<tr>
<td>Snow (hard packed)</td>
<td>0.2</td>
<td>0.15</td>
</tr>
<tr>
<td>Ice</td>
<td>0.1</td>
<td>0.07</td>
</tr>
</tbody>
</table>

It is necessary to measure the value of tire road friction coefficient in real times mainly for safety limit warning, intelligent control and vehicle stability. It is difficult to determine the value of the tire road friction coefficient online. This coefficient is affected by different factors like the material used in the tire, surface conditions, tire pressure and vehicle load. The tire road friction coefficient is mostly estimated as the ratio of normal force of the vehicle and longitudinal or lateral force of the vehicle. Tire road friction coefficient is generally determined as

$$\mu = \frac{F}{N}$$

(4.26)

Where,

$\mu$ is friction coefficient, $N$ is tire normal force and $F$ is longitudinal or lateral force.
The longitudinal force of the vehicle equation (4.26) can be written for maximum value or the peak value of tire road friction coefficient as

$$\mu_{\text{max}} = \frac{F_x}{F_z}$$  \hspace{1cm} (4.27)

Where,

$\mu_{\text{max}}$ is maximum tire road friction coefficient, $F_x$ is the longitudinal tire force and $F_z$ is the normal load.

The longitudinal force generated at each tire is dependent on the tire road friction coefficient, longitudinal slip ratio and vertical load on the tire. Figure 4.3 shows the road friction coefficient versus slip ratio for different types of road.

![Figure 4.3 Longitudinal force versus tire slip ratio using Magic Formula tire model [13].](image)

This is related to the normal tire force or vertical load on the tire. From the known value of the normal load and tire slip ratio, tire-road friction coefficient can be estimated.

An accurate estimation of maximum tire road friction coefficient is required for optimum handling of the vehicle. Unfortunately very few research works related to ESC development or ESC simulations have discussed the accurate estimation of tire road friction coefficient by following any specified method.
Instead of covering the method of estimation in research works related to simulation and development of ESC, the subject is mostly covered in separate research works as tire road friction coefficient estimation itself a vast and complex area. To summarise the updates in tire road friction estimations, several research works are discussed here.

To estimate tire road friction coefficient many approaches have been proposed in different research works. According to the paper [52] these estimation methods can be categorized into caused-based and effect-based approaches. Caused based approaches use different sensors like optical, thermal, sonar or other sensors which have the benefit of estimating the friction of coefficient before the critical manoeuvres.

It is very difficult sometimes to be accurate because of the existence of foreign materials in the contact patch of the tire and the road. Expensive sensor based methods are also affected by tire’s age and condition. The other approaches are effect-based and they use vehicle and tire dynamics like tire slip ratio, slip angle and longitudinal forces. Effect based estimation can include sensor like GPS or DGPS along with an electronic gyroscope. Most of the research work is done on effect-based estimations and demonstrate several pros and cons.

A piezoelectric wireless tire sensor is used inside the tire on the contact patch in [53] to measure force, moment and slip angle variables. Then the brush tire model estimation can be done for tire road friction and slip angle of the vehicle. This paper also describes the on board sensor based estimation methods using engine torque, a steering angle sensor, wheel speed sensor, yaw rate sensor, GPS receiver, strain gauge and steering torque. The complexity of the system can be an extra load for the ESC in real time calculation for wheel force unless a separate controller is used for this purpose. Only then can this readily available data is fed to the ESC. Use of a GPS alone cannot help if GPS-based identification is used because of satellite drop-outs. This kind of system can be useful with the help of a gyroscope for estimation based on lateral tire models as done in [54].
Slip based estimation of tire road friction coefficient is widely used in several papers. Tire road coefficient can be estimated using wheel slip or the difference between the velocity of the different wheel [55]. It is based on the difference of driven and non-driven wheel velocity. In [56] tire road friction is estimated only using angular wheel velocity. A parameter used in the proposed equations along with other normalized parameters, like rubber longitudinal stiffness and change in road characteristics is identified.

Slip based estimation of maximum tire force is also done using the brake force during braking in [57]. This research concluded that maximum friction coefficient can be deduced from the slip curve when slip is relatively small and can be done during normal braking. A braking based estimation of road friction is kept as the secondary choice.

Estimation of tire road coefficient using longitudinal tire force is done in the paper [58]. Generation of differential longitudinal tire force between the left and right tires will affect the yaw rate which can be used to estimate cornering stiffness and friction coefficient. While running the vehicle straight with a slip angle of zero or a very small slip angle, tire slip angle can be changed by yaw moment which can be created by differential longitudinal force. This is the primary method used to find the cornering stiffness. For a large slip angle, when a vehicle is turning, an estimation of friction coefficient can be done. The method proposes a trial based estimation prior to ESC manoeuvres where estimated values can be stored in the controller’s memory and used later. Otherwise this estimation has to be made when the vehicle is being manoeuvred. This can be hard to achieve in real time stability control.

As the vehicle used here is a four in-wheel-motor electric vehicle where torque generation is measurable in real time, we can avoid very complex tire road friction estimation methods for ESC development. Comparison of the mechanical and electrical wheel torque of the plant vehicle model provides a flexible way of finding tire force. More generalized methods which are related to vehicle dynamics can be used so that the controller can estimate the friction coefficient without complexity. An example is given here by simplifying equation (4.26)
\[ \mu = \left| \frac{a}{g} \right| \]  \hspace{1cm} (4.28)

Where, \( a \) is the acceleration and \( g \) is the gravitational force.

### 4.3 Control law and Strategies

The key point in yaw stability control is to find the required corrective yaw moment. As a first step researchers have used vehicle dynamics parameters to find the required yaw moment. Different types of control law have also been used by researchers. Various control laws and strategies are addressed here.

Control laws, defined by the researchers in previous research works, use similar control parameters like yaw rate and vehicle slip angle. As we discussed previously, choosing both the yaw rate and vehicle slip is a logical approach for yaw stability. But dissimilarities occur in the use of control parameters in a single controller or multiple controllers.

#### 4.3.1 Previously used strategies

In Figure 4.4, a fuzzy logic based direct yaw control for all wheel drive EV is proposed in [40], [55] which can be taken as case 1 for discussion. Here fuzzy logic is used considering the non-linearity of the system, possible increase in number of sensors, development cost and time. A novel strategy is adopted where yaw rate and wheel slip ratio are controlled in the control loop. Separate fuzzy logic based controllers for yaw rate and wheels’ slip ratio are used in this yaw motion control loop, where the yaw rate controller determines the required torque.
Applied corrective torque from the yaw controller can saturate the tire force. To avoid this problem a slip controller is used to keep the slip ratio within the stable region by generating weakening torque for the wheels. In the yaw rate controller, reference yaw rate is generated by a neural-network. This neural network can generate the reference yaw based on the vehicle speed and steering angle. The neural network must be developed by running the vehicle on the road. Results show that a yaw-controller and a slip controller can bring the vehicle to alignment without blocking the wheels.

In case 2, the discussion is based on hierarchical vehicle stability control systems using a fuzzy logic based controller designed in the paper [35, 36]. Here, the controller is designed to calculate the desired yaw moment from yaw rate error and vehicle slip angle error.

Desired wheel slip ratios are calculated based on desired yaw moment using fuzzy logic based control allocation. It is assumed that ABS/TCS is available to manipulate braking or wheel traction to achieve the yaw moment. The flowchart provided in this research work as given in Figure 4.5, shows that the difference between the actual and the desired slip angle and yaw rate is considered as controller input to calculate desired yaw moment. The controller uses yaw rate and slip angle to find yaw moment and is responsible for controlling the individual wheel slip.
In [36] a hierarchical vehicle stability control system is designed where the difference of actual and desired yaw rate is taken as input to the first layer of the controller. This finds the desired yaw moment and calculates the desired longitudinal wheel slip rate for each wheel. The second layer, or control allocation layer, uses lateral force and yaw moment in a sliding mode controller for actuation signals.

Both the yaw rate error and slip angle errors as control parameters used in [34, 38], which is taken as case 3, is shown in Figure 4.6. A fuzzy controller is used to determine the corrective yaw moment. For more robustness an active front steering controller based on the sliding mode controller is used which provides the correct steering angle. Using a steering control and a fuzzy controller together is a rarely used method.
Another strategy is used for yaw rate control employing independent in-wheel motors in [59] for case 4 which is shown in Figure 4.7.

A PID controller for lateral acceleration error, a sliding mode controller for yaw rate error and another PID controller for wheel slip error are used to generate the correcting wheel torque. Corrective wheel torques, generated by these three controllers, are then combined and used as the total corrective torque to control the yaw rate of the vehicle.

There are many different strategies of torque based ESC for EV. Novel ideas are used in several research papers concerning the control of yaw moment, wheel slip ratio, and vehicle side slip angle and steering angle in different combination.
Several controllers like fuzzy logic, sliding mode, neural network or PID are used here for non-linear behaviour of the system. They can also be used in parallel or inner loop fashion to control vehicle dynamics.

A very complex control strategy for a real time environment may incur cost in terms of time and processing capability which is hardly considered in these research works. Being more concerned about the simulation of ESC would be a possible reason for this. Different types of actuations for creating a combined compensation to control the system eventually may get over complex. A simplified drive-train with four in-wheel electric motors in an EV can have a simple control strategy for stability by controlling the torque of each individual wheel. A model based approach for estimating vehicle and wheel dynamics is required to identify the vehicle state and using these models can reduce the error and complexity.

These reference models are supposed to provide the desired values of the required parameters. Prior setting of a targeted value of the parameter removes the uncertainty of achieving the desired values. It would not be possible to entirely achieve the nominal yaw rate but an ESC may be able to make the actual yaw rate closer to the expected yaw rate.

### 4.4 Wheel torque control technique

Actuations in yaw stability control are typically related to control the wheels’ rotational speed. In braking based stability control, yaw moment is created by differential braking in each wheel. Braking based ESC is a popular technique for conventional vehicles. A number of research works have been published on this topic.

The vast number of research works on differential braking based ESC for conventional vehicles have given us a comprehensive description on the technique. But differential torque based ESC is currently the focus of the researchers and industry as advanced EVs are introduced. A type of advanced EV is assumed to have individual motor in the hub of the wheel.
This architecture makes possible the control of wheel rotation in each wheel. In a differential torque based stability control, actuation is accomplished by controlling the wheel rotation which generates differential torque between left and right wheel. Typically, differential torque based ESC related research works described wheel torque calculations or force calculations where a detailed description of torque generation is absent in most of the cases.

One reason for these shortcomings is because this approach has only been simulated and never had to face the challenges of practical implementation. The majority of these research works validated their ESC systems by running computer based simulations, where algebraic calculation of corrective torque and force is done and fed into the plant for stabilization. As a result the details of wheel rotation control or controlling the electric motors are kept very brief in most of the cases. Differential torque generation technique by actuating individual wheels is an important part of this research. The techniques of wheel control should be different in four wheels independent control EV compared to conventional single electric motor based vehicles. Implementation of motor control poses a major challenge to in-wheel motors on all four wheels, and most OEMs' preference for centralised motors made this problem more critical.

As part of an investigation into differential torque based ESC, several research works which include torque control, wheel rotation control and motor control are discussed here. A permanent magnet motor model and motor torque dynamics model are used in [35] to provide the required torque in the stability system of an all-wheel drive EV. A similar motor torque dynamics model is also used in [60] for stability control of a four-wheel-drive EV. Both of the above mentioned research papers have addressed torque control dynamics, but more work might be needed to clarify their relation to the main control system, especially if an electric motor is used. It can be assumed that the anticipated torque is based on the tire model output [35].

Dynamics of wheel speed are presented in [61] and wheel torque calculation is made considering both the drive and brake torque.
Despite the absence of a more detailed description of actuation, this research work clearly shows the relation of the wheel dynamics and wheel torque to the control law.

A clearer idea of wheel control for demanded torque is depicted in [39] for vehicle stability control. Here the tire model is used for force calculation and the simplified motor model constructed using the MAP of torque, PWM and rotational speed of the motor. The torque model is given below:

\[ T_e = T_{es} \cdot \frac{1}{\tau_m s + 1} \]  

(4.29)

Here, \( T_e \) is motor torque, \( T_{es} \) is motor steady torque and \( \tau_m \) is time constant.

The demanded torque is generated from the simplified model using PWM and RPM as inputs of the motor drive control system. This research work covered a detailed wheel torque control technique only using simulation.

More research work on four in-wheel motor stability control has been done to understand the wheel control techniques needed to fulfil the required torque.

Though an innovative concept of yaw control system is presented by determining the torque of the wheel in [59], the torque and wheel rotational dynamics are ignored. Other research work on vehicle stability using differential torque has been considered. This includes the wheel torque control where only wheel angular velocity is mentioned in [44] and is overlooked in [27] and [38].

After investigating a number of research works, it is found that wheel mechanical and electrical torque calculation should be included to provide a clear idea on wheel torque control technique. A proper description of wheel speed control technique is also required. In the next sections, these proper issues are clarified.
4.4.1 Mechanical torque calculation of wheel

The total torque acting on the wheel divided by the moment of inertia of the wheel equals to the wheel angular acceleration. The total torque consists of torque from motor, brake torque and torque from wheel friction force. Figure 4.8 shows wheel rotational dynamics under the influence of torque. The dynamic equation for wheel angular motion is given below:

\[ \dot{\omega}_w = \frac{[T_d - T_b - r_{eff}F_x]}{J_w} \quad (4.30) \]

Here,
- \( T_d \) is the drive torque of the wheel
- \( T_b \) is the braking torque of the wheel
- \( F_x \) is the wheel longitudinal force
- \( r_{eff} \) is the effective radius of wheel.

To discuss wheel torque control in a four in-wheel motors EV by controlling the wheel rotation. We need to know wheel rotation, wheel force and angular velocity. From vehicle model and tire model, force of the wheel and the angular velocity is estimated. Desired torque or required mechanical torque is estimated from the control system. This data is compared with the actual data to achieve the expected wheel torque.

Actual wheel rotation is determined via a built-in Hall Effect sensor in the in-wheel motor. From electrical data of the wheel, electrical torque is calculated. Once the actual wheel rotation is determined, then controllers control the wheel speed to achieve the actual torque which is close to the expected torque. When all the four wheels achieve the expected wheel speed then the vehicle comes close to the expected direction of motion.
In this section we determine the mechanical torque from wheel rotational dynamics by re-writing the equation (4.30) as [62]:

\[ J_w \ddot{\omega}_i = T_{di} - T_{bi} - r_{eff} F_{xi} \]  

(4.31)

Here, \( J_w \) is the wheel inertia, \( \dot{\omega}_i \) is the angular acceleration, \( T_{di} \) is the drive torque of the wheel, \( F_{xi} \) is the wheel longitudinal force and \( r_{eff} \) is the effective radius of wheel. The subscript ‘i’ is representing any of wheels like front left- FL, front right- FR, real left- RL or rear right-RR.

From equation (4.31) it can be seen that wheel torque is expressed in terms of wheel velocity, so, the difference in torque between two wheels can be determined from the control law and numerical total torque can be determined for each wheel. Wheel angular velocity from the total torque can be calculated and part of the controller can create and track wheel velocity.
4.4.2 Proposed wheel speed control technique

Wheel angular velocity or wheel speed control is achievable simply by controlling the motor voltage. In practice, the in-wheel motor or hub wheel motor is run by a motor driver and the motor driver has a variable input voltage which changes the speed of the motor. So, control of the individual input voltage of the motors expected wheel speed is achievable for each wheel.

![Diagram of in-wheel motor with driver and throttle](image)

**Figure 4.9 An example setup of in-wheel motor with driver and throttle**

As a result the expected differential torque is achieved from the differential wheel speeds. Figure 4.9 is an example setup of a motor with driver. An operating supply voltage is required for the in-wheel motor, and a voltage varying input signal as throttle varies the speed of the motor. From a practical point of view these motors can be operated with a higher voltage to change from normal speed to turbo mode when the speed of the motor is greater than normal mode for the same input throttle signal. The lower controller can be designed to control the rotation of the wheels.
4.4.3 Wheel electrical torque calculation

The torque feedback derived from the electrical characteristics of the motor can be applied if the motor is studied in detail. The optimum torque can be derived from the electrical characteristics of the motor to provide the wheel control system. A permanent magnet brushless DC motor in the hub of the wheel is used in this research work and its torque can be measured in the DC motor form equation (4.32) given below:

\[ T_{surf} = K_a \times \Phi_t \times I_{dc} \text{ NM} \]  

(4.32)

Here, \( T_{surf} \) is the is the surface torque
\( K_a \) is armature winding constant, \( K_a = \text{Number of Poles} \times \frac{N_c}{(2\times\pi)} \), \( N_c \) is current-carrying conductors
\( \Phi_t \) is total flux per pole in webers
\( I_{dc} \) is the average dc bus current
A detail of this torque calculation can be found in [63] which is helpful to estimate electrical torque to ensure the expected generated torque along with the angular velocity of the wheel.

4.5 Overall discussion and scope of current research

An overall view of ESC systems developed or proposed in the previous research works for four in-wheel motors EV is shown in Figure 4.10. This overview shows the related parameters of ESC found in previous research works. This includes different levels of ESC systems development. The levels represent the phases of ESC development.

Discussion in the previous sections of this chapter is based on the limited amount of research related to ESC systems. Some views on different levels of ESC are given below.
Table 4-2 Levels or phases of ESC system

<table>
<thead>
<tr>
<th>Level</th>
<th>Phase related to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Finding vehicle dynamics by using of vehicle model.</td>
</tr>
<tr>
<td>2</td>
<td>Parameter estimation e.g. slip angle and yaw rate to determine vehicle state</td>
</tr>
<tr>
<td>3</td>
<td>Reference values of different parameters to make comparison.</td>
</tr>
<tr>
<td>4</td>
<td>Control laws and variables</td>
</tr>
<tr>
<td>5</td>
<td>Actuation</td>
</tr>
<tr>
<td>6</td>
<td>Real time control</td>
</tr>
</tbody>
</table>

These speculations provide overviews on what are the parameters are used in each phase and how they are used. This eventually determines the amount of research work done and the possible options or scope for conducting research on ESC systems.

**Level 1:**

A variety of software tools have been used in vehicle modelling and simulation. This software uses experimental data and empirical equations to find vehicle states. Vehicle body, wheel and other components from software tools like ADAMS CAR, CARSIM and SIMULINK are used in several researches.

**Level 2:**

Different types of sensors and critical estimations are done for several parameters like tire-road coefficient, vehicle body slip angle, wheel-slip angle and yaw rate. Different parameter estimations have been done in several research works. These estimations introduced new ideas and offered precise outcomes. But the degree of complexity of these innovative ideas sometimes overwhelms the ESC system in terms of implementation.
Figure 4.10 Overall coverage on differential torque based ESC done in previous works
Level 3:

A number of items are used in different ESC systems as reference value of vehicle dynamics. They are wheel slip angle, yaw rate, body slip angle and lateral acceleration. These parameters are used in several researches on stability control systems to find the vehicle states with different combinations. Sometimes more than two types of reference value are used to compare the vehicle state.

Level 4:

Innovative and novel ideas have been used in ESC control laws. Numerous types of control laws like fuzzy logic control, sliding mode control and neural network based control law have been used in different research works. Multiple variables in a single control law have been used in most of the research works. Several strategies have been adopted in different research activities addressed in this chapter. Some strategies have used multiple control parameters and sometimes more than one control law. The control laws are introduced ignoring the complexity of implementation in real-time.

Level 5:

In actuation, mostly numerical calculations are done on the wheels for driving torque. In some research works the details of actuation is kept very limited and assumed pre-existing ABS or TCS to provide brake-torque or driving-torque. Sometimes the steering-angle is also considered for actuation with the wheels. More detail of the steps in actuation would allow us to do further research work in an easier way. However, this lack of detail at actuation level opens the scope for more research.

Level 6:

In real-time control hosting of the controller, reference value or nominal value generating and controlling wheels or steering are briefly discussed, at best, in some research works.
It is essential to have more details of the real time controller of ESC systems where crucial challenges arise. It is possible to rethink the proposed idea of ESC at this level reaching the limitations associated with the host e.g. the digital controller. There is more opportunity for further research for a simplified system than with conventional systems.

4.6 Proposed vehicle stability control method

Previously in this chapter we discussed control parameters, strategies and controller types used in vehicle stability control. From the discussion a simple and effective method for yaw stability control is proposed. This method is for a simplified four in-wheel drive train. The proposed yaw stability control is described here which is less complex in terms of parameters, strategy and implementation. Complex parameter estimation is avoided for different physical properties of the vehicle and the environment. Multiple parallel controllers to control vehicle dynamics like yaw rate, lateral acceleration, vehicle slip angle or wheel slip are avoided to reduce the complexity of real-time implementation of the system.

4.6.1 Overview of the method

The proposed stability controller calculates the expected and actual direction of vehicle motion and the difference between them. From the vehicle model and tire model it calculates vehicle and wheel dynamics. Using the directional difference and other vehicle dynamics the controller calculates the corrective yaw moment. To bring the vehicle close to the expected direction of motion, the controller calculates difference in longitudinal force between left and right wheels. Longitudinal force difference is created by changing wheel torque. To create the expected change in torque of the wheel, the controller manipulates the wheel speed. Wheel speed is changed by varying the voltage input and current flow through the electric motors. This is a brief overview of the method; the next sections elaborate the method.
4.6.2 **Controller structure**

From the proposed method overview we have seen that there are several stages of calculation before actuation. To make the controller efficient, we have decided to design a hierarchical controller. This hierarchical controller has two levels. They are:

- Top level controller
- Lower level controller

4.6.3 **Top level controller**

The top level of the controller determines the required longitudinal force difference. Figure 4.11 shows the top level controller which takes vehicle dynamics as input and calculates required longitudinal wheel force differences.

![Figure 4.11 Top level of the proposed controller](image-url)
Actual vehicle dynamics and nominal vehicle dynamics are calculated using these models. The vehicle model is used to generate vehicle longitudinal and lateral velocity and yaw-rate.

4.6.3.1 Top level input and output

The top level of the controller takes yaw rate error and side slip error as input to a single control loop. These errors are calculated using vehicle model, tire model and sensors. Actual yaw rate, expected yaw rate, slip angle and some other required parameters are derived from the steering wheel angle input. The control loop in this top level controller determines errors in yaw rate of the vehicle. A change in steering angle input will change the error between the expected and actual yaw rate which changes the output of the control loop as long as significant error exists.

As ESC does not assist in automated driving, it does not neglect the steering input during the implementation time. So, when the steering input is changed at the time of control, the controller considers the new steering angle as the intended direction of the driver. The controller then determines the required values for this new input and then actuation take place. Typically, ESC tries to bring the vehicle back to the expected direction by actuating wheel rotation within a very small amount of time or within the transient state before the driver realizes. The driver will have a natural view of driving and the steering angle input. If it takes multiple iterations of actuation then it is possible for the driver to change the steering angle. If the vehicle faces a higher slip angle in that time there will be almost no change in the actual direction of motion. The ESC will run iteration in this intended direction. Or, if the vehicle follows the intended direction based on the steering angle at that time then the vehicle is under control.

This discussion also involves the control loop variables. Yaw velocity describes the lateral motion of the vehicle which could be used as the only control variable. But when driving on a slippery road, lateral acceleration and yaw rate do not correspond with each other. This happens because on slippery road, slip angle increases rapidly.
Vehicle slip angle and yaw rate both must be within the limits that correspond to the tire road friction coefficient. Hence, both the parameters are taken as control variables in a single control as suggested by several researchers [11, 64]. Desired or nominal yaw rate and nominal side slip angle is generated by the reference vehicle model. Then possible maximum values of desired yaw rate and slip angle are determined.

4.6.3.2 Required longitudinal force calculation

The proposed strategy of a hierarchical control system structure is depicted partially here in Figure 4.12.

![Proposed control strategy](image)

**Figure 4.12 Proposed control strategy (partial) using yaw rate and vehicle slip angle**

The top level controller as shown in Figure 4.12 determines the actual vehicle dynamics and derives nominal vehicle dynamics from the sensors and physical properties of the vehicle. The figure shows that the top level of the controller takes the input of yaw rate error and slip angle error. A sliding mode based control law in the top level controller determines the longitudinal force difference to correct the yaw of the vehicle. These force differences are used in wheel rotational dynamics calculation to determine the required torque and wheel velocity.
4.6.4 Lower level controller

In section 4.4.1 discussion is done on mechanical torque calculation of the wheel where the wheel rotation is related to the torque and force of the wheel. The proposed lower level controller achieves the required torque by changing the wheel speed. Figure 4.13 shows the lower level of the controller. The figure shows that the lower level controller determines expected wheel speed to create the torque.

![Proposed lower level controller](image)

Figure 4.13 Lower level of the proposed controller

4.6.4.1 Lower level input and output

The lower level takes the required longitudinal force differences between the wheels as input. Then it determines the differential torque based on the input from the upper level. Here a control variable is used to determine the corrective wheel driving torque for left and right wheels taking into consideration the required longitudinal wheel force.
This variable determines the amount of torque on the wheels to create the required force difference which essentially used the wheel rotational dynamics.

The lower level controller calculates the expected wheel speed and provides output to the interface of the wheel speed controller. This part of the lower level controller takes the current wheel speed as input. Based on the current wheel speed this interface provides voltage as output to change rotation of the wheel.

4.6.4.2 *Wheel speed control interface*

This proposed interface makes it possible to achieve the torque difference by changing the wheel angular velocity. This can be done by controlling the wheel rotation. The expected wheel rotation is achieved by manipulating the input voltage of each individual motor. This change of input voltage changes the current flow in the electrical motor which changes the rotational speed. Change of current flow in the electric motor is achieved by changing the input throttle voltage of the motor controller.

![Figure 4.14 Proposed wheel speed controller](image)

*Figure 4.14 Proposed wheel speed controller*
This wheel speed control requires a higher level of accuracy and speed. To achieve this, a PID controller is used with a feedback signal from the wheel. The proposed wheel speed controller is given in the Figure 4.14.

Figure 4.15 Proposed method of stability control with the overall system
This figure shows the method of proposed yaw stability control. It includes the top level controller and lower level controller. For the sake of clarity this proposed system is shown including vehicle and controllers.

### 4.6.5 Proposed method summary

A simplified determination of differential driving torque for the wheels to compensate for corrective yaw moment is proposed for the lower level control. A simple and fast wheel rotational controller is proposed as a part of the lower level control.

For the real-time processing a high performance Field Programmable Gate Array (FPGA) based controller with higher accuracy input output modules is proposed here as an embedded host. The entire proposed system is shown in the Figure 4.15.

In brief it can be said that, in this hierarchical control system a simplified vehicle model and tire model are used to generate tire dynamics. Desired or nominal yaw rate and nominal side slip are generated by the reference model. These desired values are kept inside a certain limit to achieve the target. After calculating the corrective yaw moment, difference between left and right wheel forces are derived from it. These force differences are used in wheel dynamics to determine the required torque and wheel velocity.

### 4.7 Summary

This chapter has included a wide investigation of differential torque based stability control. We reviewed vehicle dynamics and tire force related parameters and estimations. This discussion included the different strategies to reduce complexity for a four in wheel EV.

In this chapter some strategies are discussed on control methods using multiple parameters to show the importance of adopting them for this EV.
From the discussion made in this chapter, we can gain some idea of the amount of research in torque based ESC. The possibility of wheel speed control for compensating in yaw correction is not discussed in detail or tested in a four in-wheel motor. Discussion on differential torque based strategies has been most helpful for selecting suitable methods of estimations and determinations of vehicle dynamics.

A method of vehicle stability control technique is proposed in this chapter with the necessary interfaces. Based on this proposed method, an in-wheel ESC system is simulated in the next chapter. As in the simulation we require a virtual vehicle to provide the responses, so a reference vehicle model is created. Front and rear wheels are modelled to provide the wheel dynamics in the simulation.
Chapter 5
Modelling the components of four in-wheel EV for ESC
5 Modelling the components of four in-wheel EV for ESC

5.1 Introduction

Despite numerous multi-body simulation packages, a vehicle model to simulate vehicle handling control systems is presented here. We also consider the existing constraints in available simulation software. The relevant tasks and challenges for modelling and simulating the stability controller for electric vehicle with four in wheels are explained. Co-simulation is a common solution for multi-dimensions system like this. Interchange of data between these multiple systems can cause problems with input and output structure. This may also impose limitations in the control algorithm for an advanced vehicle. Detailed coverage of the model is necessary to counter the phenomenon of under-prediction in simulation. A vehicle model was developed using SIMULINK basic blocks. Longitudinal and lateral motion, front and rear wheel with steering angle, electric motor and controller, physical environment and a stability controller were included. Modelling and simulation of the vehicle and the stability controller are discussed in this chapter. Some of the sections in this chapter are based on the paper “Vehicle modelling for ESC in a four in-wheel electric vehicle” [65]. At the end of this chapter simulation results are discussed.

5.2 Requirement of Vehicle modelling

Modelling and simulation is always an important issue in the engineering process. In order to reduce the labour and cost in terms of money and time, different simulation tools have been used to simulate the entire system before starting physical development.

An electric vehicle, consisting of four in-wheel motors has a greater control flexibility [2, 66] but physical multiple testing in the design phase is not required if it can be simulated. In order to perform a vehicle stability test using a control method, it is necessary to create the vehicle virtually in a software environment which can provide responses in terms of the required vehicle dynamics.
In this case a four in-wheel virtual vehicle is created for a vehicle stability test where vehicle body dynamics and wheel dynamics are observed and controlled. It is inevitable that this virtual vehicle would respond in a predictable way based on the mathematical model but if detailed coverage of the wheel models and body models are considered in the program, this would be sufficient for simulation. The challenge faced for modelling the entire environment is to create the simulation environment from scratch so that it can meet the current and future requirements in terms of accessing the parameters.

The most common way of simulating a vehicle stability system is co-simulation. In such co-simulations, for example, controllers are developed in SIMULINK for those vehicle models available in ADAMS car [67], CarSim [68] and other advanced vehicle simulating software. Without a comprehensive model for the advanced vehicle mentioned here, the electrical and mechanical properties, stability control law and algorithm may not be simulated and verified properly. SIMULINK is a powerful tool to model parts of the vehicle system according to the design. Here, the model has to be developed taking into consideration the reconfiguration of the vehicle body, wheels, environment and controller. Reconfiguration is needed for a hardware test to analyse the performance of the actuator and sensors.

Control analysis and controller design for vehicle motion rely on vehicle dynamics. Based on the investigation of the vehicle model described in section 4.2.1, some requirements of modelling are identified:

- A detailed and comprehensive vehicle model is required to reproduce the behaviour of individual components of the in-wheel EV as exactly as possible.
- The model should provide longitudinal motion, lateral motion and yaw.
- The model should provide interactions between the subsystems, and allow access to the parameters.
- Modelling should be done taking into consideration the reference vehicle model which will be embedded in the electronic digital controller.
Such a vehicle model can be developed using mathematical equations for vehicle dynamics. Using these mathematical equations a computer model is made that helps to analyse the controller before prototyping.

5.3 Overview on the proposed simulation

Vehicle models can be very specific. Instead of using a generalized vehicle model a specific vehicle model is developed here for ESC. It is necessary to present the entire ESC system model with data and control signal flow, parameter estimations, control law and actuations. Here, the simulation environment and calculations are created in MATLAB-SIMULINK where most of the component blocks are invisible from the top level. For example, the vehicle body block has three sub blocks for calculating longitudinal velocity, lateral velocity and yaw rate. The overall top view of the entire system cannot visualize the associated sub systems. Keeping this problem in mind, the entire settings of the simulation environment with the associated sub-systems are visualized here in a schematic diagram. The schematic diagram is given in Figure 5.1 for ESC controller and vehicle model.

The self-explanatory simulation block proposed here shows the inputs for simulation, major vehicle component blocks like wheels and body, desired value generation blocks and controller blocks. Wheel blocks generate the longitudinal and lateral forces with the initial value of angular wheel velocity $\omega$ and vehicle longitudinal velocity $x$. The vehicle body block calculates longitudinal and lateral velocity using the longitudinal and lateral forces coming from wheel blocks with the given initial vehicle velocity. The vehicle body block also calculates the yaw rate from the wheel forces and steering wheel angle input. From the calculated longitudinal and lateral velocity, vehicle slip angle is determined. All these outputs from the vehicle body go into the controller where they are then compared with the nominal values. The control law and other blocks of the controller then determine the differential torque to create the corrective yaw moment.
Figure 5.1 All components of proposed simulation for ESC
5.3.1 Vehicle body modelling

The fundamental law of motion and the geometric relationships, longitudinal velocity, lateral velocity and yaw rate can be measured. The vehicle body model includes the modelling of vertical load or normal forces on the rear and front of the vehicle body.

In order to avoid the complexity of modelling a large system like the vehicle full body, is divided into five sub-models or sub-systems, they are:

- Velocity in X axis
- Velocity in Y axis
- Yaw rate
- Vertical load in the front
- Vertical load at the rear

After creating the computer model of these sub-models from their mathematical model, they are encapsulated as a subsystem of the vehicle body model.

It is important to mention that the entire vehicle is treated as a single mass concentrated in its CoG. So, longitudinal velocity and lateral velocity are measured around the CoG. The sum of the external forces acting on the vehicle body in the longitudinal and lateral axes is equal to the product of the vehicle mass and acceleration in a given direction. This also includes the sum of torques acting on the vehicle body, which is equal to the moment of inertia times by the rotational acceleration about the vehicle axis. The equation of longitudinal motion of the vehicle is given in equation (4.5) in the previous chapter 4. This simplified equation of motion to find the longitudinal velocity of the vehicle is considered without the influence of air drag and rolling resistance. It is done with the assumption that this equation will provide the vehicle velocity with an influence of a speed controller which will try to maintain constant speed. The influence of road bank is not considered in vehicle longitudinal velocity calculation because this simulation only uses a plane surface to test the vehicle stability. But in calculation of normal force on the wheel, aerodynamic resistance and road bank is considered to get a realistic output.
It is worth mentioning that the computer model for vehicle motions is developed using SIMULINK, where the forces from the mathematical model are time-varying functions. So, by integrating equation (4.5), (4.6) and (4.7) with respect to time $t$, equations (5.1), (5.2) and (5.3) are given below to find the longitudinal velocity of the vehicle, lateral velocity of the vehicle and yaw rate respectively.

$$\dot{x} = \Psi \dot{y} + \frac{1}{m}[(F_{xfl} + F_{xfr})\cos(\delta) + F_{xrl} + F_{xrr} - (F_{yfl} + F_{yfr})\sin(\delta)]$$  \hspace{1cm} (5.1)$$

$$\dot{y} = -\Psi \dot{x} + \frac{1}{m}[F_{yrl} + F_{yrr} + (F_{xfl} + F_{xfr})\sin(\delta) + (F_{yfl} + F_{yfr})\cos(\delta)]$$  \hspace{1cm} (5.2)$$

$$\Psi = \frac{1}{l_z}l_f(F_{xfl} + F_{xfr})\sin(\delta) + l_f(F_{yfl} + F_{yfr})\cos(\delta) - l_r(F_{yrl} + F_{yrr}) +$$

$$\frac{l_w}{2}(F_{xfr} - F_{xfl})\cos(\delta) + \frac{l_w}{2}(F_{xrr} - F_{xrl}) + \frac{l_w}{2}(F_{yfl} - F_{yfr})\sin(\delta)$$  \hspace{1cm} (5.3)$$

Figure 5.2 Forces acting on the vehicle

Equation (5.1) describes the motion in X axis of the vehicle. Assuming that the net force from the wheels for driving the vehicle in X axis are $F_{xfl}$, $F_{xfr}$, $F_{xrl}$ and $F_{xrr}$. Equation (5.2) shows the motion in y axis, here $F_{yfl}$, $F_{yfr}$, $F_{yrl}$ and $F_{yrr}$ which are the induced
lateral forces in the Y axis. Equation (5.3) shows the relationship between acting torques on vehicle body Z axis and the moment of inertia of the vehicle from where the yaw rate is derived. Figure 5.2 shows the forces acting on a vehicle.

Vehicle body modelling also includes modelling vertical load on a vehicle. Loads on the front and rear axles work in the Z axis of the vehicle. According to SAE these forces acting downward are positive. Modelling of vertical load in the front of the vehicle $F_{zf}$ and vertical load at the rear of the vehicle $F_{zr}$ are discussed here.

![Figure 5.3 Illustration of vehicle body subsystem block](image)

Let,
Inclined angle or angle of slope of the vehicle with ground be $\theta$
Gravitational force be $g$
Height of CoG is be $h_{CoG}$
Height of the point of application of the aerodynamic resistance be $h_{aero}$
Aerodynamic drag force be $F_{aero}$
Then the vertical load acting on the front axle while climbing up of a slope with angle $\theta$

$$F_{zf} = \frac{1}{(f_f + f_r)} \left[ mg l_f \cos \theta - F_{aero} h_{aero} - h_{CoG} m \frac{d}{dt} \dot{x} - mgh_{CoG} \sin \theta \right]$$ \hspace{1cm} (5.4)

And the vertical load acting on the rear axle

$$F_{zr} = \frac{1}{(f_f + f_r)} \left[ mg l_f \cos \theta + F_{aero} h_{aero} + h_{CoG} m \frac{d}{dt} \dot{x} + mgh_{CoG} \sin \theta \right]$$ \hspace{1cm} (5.5)

Combining all these equations (5.1), (5.2), (5.3), (5.4) and (5.5) the vehicle body block is constructed as shown in Figure 5.3.

Table 5-1 Numbered labels in the figure 5.3 describe the attributes of input and output parameters in the vehicle body modelling.

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Longitudinal wheel forces</td>
<td>Wheel subsystem</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Lateral wheel forces</td>
<td>Wheel subsystem</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Steering angle</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Vehicle initial velocity</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Distance from front axle to CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Longitudinal velocity</td>
<td>Vehicle Body subsystem</td>
<td>Wheel subsystem, desired</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>yaw rate subsystem, User/simulation</td>
</tr>
<tr>
<td>7</td>
<td>Yaw rate</td>
<td>Vehicle Body subsystem</td>
<td>Sliding-mode controller subsystem, User/simulation</td>
</tr>
<tr>
<td>8</td>
<td>Lateral velocity</td>
<td>Vehicle Body subsystem</td>
<td>Wheel subsystem, Vehicle</td>
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<td></td>
<td></td>
<td></td>
<td>Body subsystem</td>
</tr>
<tr>
<td>9</td>
<td>Gravitational force</td>
<td>User/simulation</td>
<td></td>
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<td>10</td>
<td>Vehicle mass</td>
<td>Vehicle properties</td>
<td></td>
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<td>---</td>
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</tr>
<tr>
<td>11</td>
<td>Slope angle</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Aerodynamic drag force</td>
<td>Air-drag Subsystem</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>Height of CoG and height of the point of application of the aerodynamic resistance</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Normal load on rear axle</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>Normal load on front axle</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>Wheel base width</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>Moment of inertia</td>
<td>Vehicle properties</td>
<td></td>
</tr>
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</table>

### 5.3.2 Wheel modelling

Simulation of ESC for 4 in-wheel-motor electric vehicle requires details of the involved forces acting on the wheel. Dugoff’s model is used here. This popular tire model can provide combined longitudinal and lateral forces for vehicle simulation. The most significant advantage of using this tire model is that it allows the use of independent values for tyre cornering stiffness and longitudinal stiffness [46]. Another advantage is that it requires a fewer number of coefficients compared to Magic Formula. This tyre model is comparatively simple to use to create a reference tyre model in the computer simulation and to host in an embedded controller.

Wheel model provides longitudinal and lateral forces as output. To calculate the longitudinal and lateral forces in this model requires estimating wheel slip ratio and wheel slip angle. These two estimations are done in two sub-blocks. A function is used with a variable \( \lambda \) in the wheel model this is calculated in a separate sub-block. Finally all these sub-blocks together are together to calculate the longitudinal or lateral wheel forces.
Using equation (4.11) and (4.12), a sub block is created to calculate slip ratio. Using equation (4.13) and (4.14), a sub block is created to calculate slip angle. Equation (4.15) is used to create a sub block which takes slip angle and slip ration as input.

**Longitudinal tire force and Lateral tire force**

Using equations (4.11), (4.12), (4.13), (4.15), (4.16) and (4.17) the front wheel model block subsystem is constructed as given in the Figure 5.4.

**Figure 5.4 Front wheel subsystem model**

Using equations (4.11), (4.12), (4.14), (4.15), (4.16) and (4.17) the rear wheel model block subsystem is constructed as given in the Figure 5.5.

**Figure 5.5 Rear wheel subsystem model**
Table 5-2 Numbered labels in the figure 5.4 and figure 5.5 describe the attributes of input and output parameters in the wheel modelling.

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Effective radius</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Wheel angular velocity</td>
<td>Initial value/Wheel rotation controller</td>
<td>Slip ratio block in wheel subsystem/ wheel torque calculation block</td>
</tr>
<tr>
<td>3</td>
<td>Longitudinal velocity of vehicle</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Lateral velocity of vehicle</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Distance of front axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Yaw rate</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Steering angle</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Longitudinal slip ratios for each wheel</td>
<td>Slip ratio block in wheel subsystem</td>
<td>Wheel force calculation,</td>
</tr>
<tr>
<td>9</td>
<td>Slip angles at the front tire</td>
<td>Slip angle block in front wheel subsystem</td>
<td>Front wheel force calculation,</td>
</tr>
<tr>
<td>10</td>
<td>Normal load on front axle</td>
<td>Vehicle Body subsystem</td>
<td>Calculation of a wheel variable $\lambda$</td>
</tr>
<tr>
<td>11</td>
<td>Tire road friction coefficient</td>
<td>User/simulation</td>
<td>Calculation of a wheel variable $\lambda$</td>
</tr>
<tr>
<td>12</td>
<td>Longitudinal stiffness of the tire</td>
<td>User/simulation</td>
<td>Wheel force calculation,</td>
</tr>
<tr>
<td>13</td>
<td>Cornering stiffness of the tire</td>
<td>User/simulation</td>
<td>Wheel force calculation,</td>
</tr>
<tr>
<td>14</td>
<td>Function of $\lambda$</td>
<td></td>
<td>Wheel force calculation,</td>
</tr>
<tr>
<td>15</td>
<td>Longitudinal front tire</td>
<td></td>
<td>Vehicle Body subsystem</td>
</tr>
</tbody>
</table>
### Desired values model block for yaw rate and slip Angle

The difference between the actual vehicle dynamics and the expected vehicle dynamics would represent the state of stability of the vehicle in simulation. The following discussion determines the expected values of yaw rate and vehicle body slip angle which are two key parameters in the controller.

Desired side slip angle has relation with steady state steering angle. So, before going to calculate the desired slip angle, we need to determine steady state steering angle. Using equation (4.19) a sub block is created to determine steady state steering angle as shown in figure. From equation (4.19), it is seen that this steady state steering angle is dependent on the radius of the vehicle’s trajectory $R$. When the vehicle is traveling at high speed, the radius of turn is much larger than the wheel-base. The difference between steering angles on the outside and inside of the front wheel is negligible and can be assumed as a one steering angle. On the other hand, high speed cornering introduces lateral acceleration and to counteract this wheels develop some lateral forces.
Figure 5.6 Steady state steering angle calculation block

Table 5-3 Numbered labels figure 5.6 describe the attributes of input and output parameters in the steady state steering angle block

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>radius of the vehicle’s trajectory</td>
<td>Calculation in vehicle body</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Distance of front axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Distance of rear axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Cornering stiffness of rear tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Cornering stiffness of front tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Longitudinal vehicle velocity</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Vehicle mass</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>steady state steering angle</td>
<td></td>
<td>Desired and targeted slip angle</td>
</tr>
</tbody>
</table>
Considering these facts, according to [50], the cornering equation of vehicle can be found as given in the equation below.

\[ \sum F_y = \frac{mx^2}{R} \]  \hspace{1cm} (5.6)

From equation (5.6), the radius of the vehicle’s trajectory is calculated and used in the simulation to determine the steady state steering angle.

** Desired and targeted yaw rate:**

Actual yaw rate and desired yaw rate determination is required to maintain the stability by the controller. For a real-time controller an actual yaw rate can be measured directly using electronic yaw rate sensors. But for simulation the actual yaw rate can be found from the vehicle model. Desired yaw rate is determined by comparing the steady state relation between the steering angle and its generated radius of the vehicle’s trajectory by using steering angle, radius, vehicle speed, vehicle mass, tire stiffness and vehicle’s physical measurements.

Let, the cornering stiffness of front tire and rear tire respectively be \( C_{af} \) and \( C_{ar} \)

Front wheel steering angle be \( \delta \)

Distance of front axle from CoG be \( l_f \)

Distance of rear axle from CoG be \( l_r \)

\[
\Psi_{desired} = \frac{\dot{x}}{l_f + l_r + \frac{mx^2(l_rC_{ar} - l_fC_{af})}{2C_{af}C_{ar}(l_f + l_r)}} \delta \]  \hspace{1cm} (5.7)

As the desired yaw rate may not be entirely achievable, a limit or target of this value is set. The expected value is then fed into the control-law block [69]. An upper limit of the achievable yaw rate is set for simulation as given below:

\[
\Psi_{target} = 0.85 \frac{\mu g}{x} \]  \hspace{1cm} (5.8)
Where, 
\( \dot{x} \) is longitudinal velocity of vehicle, \( g \) is gravitational force and \( \mu \) is tire road friction coefficient. This target can be set to lower in the simulation if it becomes unachievable.

**Figure 5.7 Targeted yaw rate calculation**

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steering angle</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Distance of front axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Distance of rear axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Cornering stiffness of front tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Cornering stiffness of rear tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Longitudinal vehicle velocity</td>
<td>Vehicle body subsystem</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Vehicle mass</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Targeted yaw rate</td>
<td>Controller block</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Desired Yaw rate</td>
<td>User/simulation</td>
<td></td>
</tr>
</tbody>
</table>

**Table 5-4 Numbered labels in the figure 5.7 describe the attributes of input and output parameters in modelling desired yaw rate.**
**Desired and targeted slip angle:**

Desired side slip angle $\beta_{desired}$ is obtained using steady state steering angle $\delta_{ss}$, which is given below in the equation (4.25).

As the desired slip angle may not be achievable, a limit or targets of this value are set. The expected value is then fed into the control-law block.

$$
\beta_{targeted} = \tan^{-1}(0.02 \mu g)
$$

(5.9)

Where,

$g$ is gravitational force and $\mu$ is tire road friction

**Table 5-5 Numbered labels in the figure 5.8 describe the attributes of input and output parameters in modelling desired slip angle.**

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
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<td>1</td>
<td>Steady state steering angle</td>
<td>Steady state steering angle calculation block</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Distance of front axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Distance of rear axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Cornering stiffness of front tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Cornering stiffness of rear tire</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Longitudinal vehicle velocity</td>
<td>Vehicle Body subsystem</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Vehicle mass</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Targeted slip angle</td>
<td>Controller block</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Desired slip angle</td>
<td>User/simulation</td>
<td></td>
</tr>
</tbody>
</table>
5.4 Modelling Controllers for ESC in 4 in-wheel EV

In this section the sliding mode control approach and modelling of the control law are discussed in the following sections:

- Sliding mode control law
- Modelling the controller blocks

5.4.1 Sliding mode controller law

Simplified vehicle model used in the entire system modelling in this research work. This model is used for an actual nonlinear vehicle dynamics. Due to the simplicity of the plant modelling, uncertainty about the plant may arise. ESC is to provide assistance when the vehicle manoeuvres beyond the normal limit. Nonlinearity of tire force generation can create unexpected lateral dynamics of the vehicle. Therefore, a robust control technique like sliding mode can provide more improvements [49] when we use simplified models.
To keep the yaw velocity and slip angle of the vehicle limited to the values that correspond to the coefficient of friction of the road, the yaw velocity and slip angle are considered as controlled variables for the controller. The following sliding surface has been chosen as the control law which has a weighted combination of yaw rate and slip angle as proposed in different research [13, 70]. Sliding mode surface defined as follows:

\[
S = \dot{\Psi} - \Psi_{target} + \xi(\beta - \beta_{target})
\]  

(5.10)

In the above control law, the difference between actual and expected yaw rate and slip angle are used. If the desired yaw rate and slip angle are achieved by the controller, this sliding surface will converge to zero. So, we need to ensure that \(S\) converges to zero. For this purpose we need to calculate \(S\) using some assumptions.

Differentiating equation (5.11) we get,

\[
\dot{S} = \ddot{\Psi} - \ddot{\Psi}_{target} + \xi(\dot{\beta} - \dot{\beta}_{target})
\]

(5.11)

Further calculation (5.3) can be rewritten as

\[
\ddot{\Psi} = \frac{1}{l_z} [l_f (F_{xfl} + F_{xfrr}) \sin(\delta) + l_f (F_{yfll} + F_{yfrr}) \cos(\delta) - l_r (F_{yrll} + F_{yrll}) + \frac{1}{2} l_f (F_{xff} - F_{xrr}) \cos(\delta) + \frac{1}{2} l_w (F_{xrll} - F_{xrl}) + \frac{1}{2} l_w (F_{yfl} - F_{yfr}) \sin(\delta)]
\]

(5.12)

As driving torque is used for actuating corrective yaw moment, a fixed ratio \(\rho_t = 1\) of front to back torque is assumed, as each wheel is similar in terms of generating torque. So, \(F_{xrl} = \rho_t F_{xfll}\) and \(F_{xrr} = \rho_t F_{xfrr}\). 

114
Term \( l_f(F_{xf} + F_{xfr}) \sin(\delta) \) and \( \frac{l_w}{2}(F_{yf} - F_{yfr}) \sin(\delta) \) can be ignored assuming small steering angle.

So, after these assumptions (5.12) can be written as

\[
\Psi = \frac{1}{I_z}[l_f(F_{yf} + F_{yfr}) \cos \delta - l_r(F_{yr} + F_{yrr}) + (\cos \delta + \\
\rho_t)\left(\frac{l_w}{2}(F_{xfr} - F_{xf})\right)]
\]  \( (5.13) \)

Let, the term \( \frac{l_w}{2}(F_{xfr} - F_{xf}) = M_{\psi_t} \), which is the yaw torque generated from differential torque.

Then (5.13) can be written as

\[
\Psi = \frac{1}{I_z}[l_f(F_{yf} + F_{yfr}) \cos \delta - l_r(F_{yr} + F_{yrr}) + (\cos \delta + \rho_t)M_{\psi_t}]
\]  \( (5.14) \)

Equation (5.12) can be rewritten by substituting \( \Psi \) from (5.14) as given in equation (5.15) to calculate yaw moment, \( M_{\psi_t} \).

\[
M_{\psi_t} = \frac{I_z}{(\cos \delta + \rho_t)} \times \left[\frac{1}{I_z}[F_{yf} + F_{yfr}) \cos \delta + l_r(F_{yr} + F_{yrr})] + \dot{s} + \\
\Psi_{target} - \xi(\dot{\beta} - \dot{\beta}_{target})\right]
\]  \( (5.15) \)

All the variables such as \( F_y, \delta, I_z, l_f, l_r, \dot{\beta}, \dot{\beta}_{target}, \Psi_{target} \) and \( \rho_t \) are known. If we design a sliding mode surface as

\[
\dot{S} + \eta S = 0
\]  \( (5.16) \)
Then the control input $M_{\psi t}$ is given by the equation (5.17).

$$M_{\psi t} = \frac{I_z}{(\cos \delta + \rho_i)} \times \left[ \frac{1}{I_z} \left[ -(F_{yf1} + F_{yfr}) \cos \delta + l_r(F_{yrl} + F_{yrr}) \right] - \eta S + \dot{\psi}_{\text{target}} - \xi (\dot{\beta} - \dot{\beta}_{\text{target}}) \right]$$

Equation (5.16) gives the sliding mode surface by

$$S = e^{-\eta t} \quad (5.18)$$

This ensures that the sliding mode surface converges to zero. Then the desires yaw rate is achieved according to the sliding mode surface given by equation (5.10).

5.4.2 Modelling the controller blocks

The controller is modelled here to determine the differential wheel torque. Initially longitudinal wheel force difference is calculated using the vehicle, wheel and desired values block. Then possible differential torque is determined from the longitudinal force difference. To do so, a controller block is designed here with four different sub blocks, they are:

- Control law
- Longitudinal force difference calculation
- Torque calculation
- Wheel rotation controller

5.4.2.1 Modelling control law

Sliding surface as controller law is given in equation (5.10). The model of the block is given in Figure 5.9. In this block yaw rate and slip angle are the inputs. The actual value of yaw rate is achieved from the vehicle body modelling and the slip angle is derived from
vehicle velocities. Vehicle side slip angle $\beta$ is the angle between vehicle’s velocity in longitudinal and lateral axis at the CoG, which is calculated as

$$\beta = \tan^{-1} \frac{v_y}{v_x} \quad (5.19)$$

Here, $v_y$ is the lateral velocity and $v_x$ is the longitudinal velocity of the vehicle.

![Figure 5.9 Control law block](image)

**Table 5-6 Numbered labels in the figure 5.9 describe the attributes of input and output parameters in modelling control law block.**

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Yaw rate</td>
<td>Vehicle body block</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Targeted Yaw rate</td>
<td>Desired yaw rate block</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Slip angle</td>
<td>Calculation</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Targeted Slip angle</td>
<td>Desired slip angle block</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Control law variable</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Control law</td>
<td></td>
<td>Controller block</td>
</tr>
</tbody>
</table>

**5.4.2.2 Longitudinal force difference calculation:**

Longitudinal wheel force difference is required to calculate the torque difference between the left and right wheels so that a corrective yaw moment can be created. The
The proposed method is to create yaw moment primarily using wheel drive torque. The required differential longitudinal wheel force is determined from the control law by the proposed controller.

A corrective yaw moment $M_{yt}$ is generated by using differential torque to track the targeted yaw rate and slip angle.

The extra differential longitudinal tire force $F_{xf} - F_{xf_t} = \Delta F_{xf}$ can be calculated as

$$\Delta F_{xf} = \frac{l_w M_{yt}}{2}$$  \hspace{2cm} (5.20)

![Figure 5.10 Longitudinal wheel force calculation block](image-url)
Table 5-7 Numbered labels in the figure 5.10 describe the attributes of input and output parameters in the modelling of force difference calculation block

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
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<tr>
<td>2</td>
<td>Vehicle moment of inertia</td>
<td>Vehicle properties</td>
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</tr>
<tr>
<td>3</td>
<td>Control law variable</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Lateral force in front left wheel</td>
<td>Front wheel block</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Lateral force in front right wheel</td>
<td>Front wheel block</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Lateral force in rear left wheel</td>
<td>Rear wheel block</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Lateral force in rear right wheel</td>
<td>Rear wheel block</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Distance of front axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Distance of rear axle from CoG</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Steering angle</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>11</td>
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<td>Targeted yaw rate block</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Slip angle</td>
<td>Calculated</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>Targeted slip angle</td>
<td>Targeted slip angle block</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Wheel base</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>Ratio between front and rear wheel torque</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>Control law</td>
<td>Control law block</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>Longitudinal force difference between front wheels</td>
<td>Wheel controller</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>Longitudinal force difference between rear wheels</td>
<td>Wheel controller</td>
<td></td>
</tr>
</tbody>
</table>
5.4.2.3 Wheel torque calculation

Differential wheel force is calculated from the required longitudinal wheel force which can track the expected yaw rate and slip angle. This is an important part of the simulation as it requires some observation and analysis to determine the expected differential torque.

The relationship between the mechanical torque and the wheel rotation can be written as:

\[ J_w \dot{\omega} = T_d - T_b - r_{eff} F_x \]  \hspace{1cm} (5.21)

Here, \( J_w \) is the wheel inertia, \( \dot{\omega} \) is the angular acceleration, \( T_d \) is the drive torque of the wheel, \( F_x \) is the wheel longitudinal force and \( r_{eff} \) is the effective radius of wheel.

From this equation it is seen that wheel driving and braking torque create the wheel rotation. In this research work yaw stability is achieved primarily by using differential driving torque.

Differential driving torque calculation

Differential driving torque between front right and front left wheels is determined using the following equations.

Front left differential driving torque

\[ T_{dfl} = T_{dfl,init} + (1 - a) \Delta F_x r_{eff} \]  \hspace{1cm} (5.22)

Front right differential driving torque

\[ T_{dfr} = T_{dfr,init} - a \Delta F_x r_{eff} \]  \hspace{1cm} (5.23)

Here, 
\( T_{dfl} \) is the differential driving torque for the front left wheel
$T_{dfr}$ is differential driving torque for front right wheel
$T_{dft,init}$ is initial driving torque for front left wheel
$T_{dfr,init}$ is initial driving torque for front right wheel
$a$ is control variable
$\Delta F_x$ is controller calculated required longitudinal wheel force
$r_{eff}$ is wheel effective radius

Figure 5.11 Differential wheel torque calculation block

Table 5-8 Numbered labels in figure 5.11 describe the attributes of input and output parameters in Differential driving calculation block

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
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<th>Output going to</th>
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<td>2</td>
<td>Controller variable</td>
<td>User estimation</td>
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<td>3</td>
<td>Effective rolling radius</td>
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</tr>
<tr>
<td>4</td>
<td>Differential longitudinal tire force</td>
<td>Controller part</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Differential driving torque for wheel</td>
<td>Wheel rotation</td>
<td>calculation block</td>
</tr>
</tbody>
</table>
Differential braking torque calculation

Differential braking torque between front right and front left wheels is determined using the following equations. Front left differential braking torque

\[ T_{bfl} = T_{bfl,\text{init}} - b \Delta F_x r_{eff} \]  \hspace{1cm} (5.24)

Front right differential braking torque

\[ T_{dfr} = T_{dfr,\text{init}} + (1 - b) \Delta F_x r_{eff} \]  \hspace{1cm} (5.25)

Here,
- \( T_{dfl} \) is differential braking torque for front left wheel
- \( T_{dfr} \) is differential braking torque for front right wheel
- \( T_{dfl,\text{init}} \) is initial braking torque for front left wheel
- \( T_{dfr,\text{init}} \) is initial braking torque for front right wheel
- \( b \) is control variable
- \( \Delta F_x \) is controller calculated required longitudinal wheel force
- \( r_{eff} \) is wheel effective radius

![Figure 5.12 Differential braking torque Calculation block](image)

Figure 5.12 Differential braking torque Calculation block
Table 5-9 Numbered labels in figure 5.12 describe the attributes of input and output parameters in differential braking torque calculation block

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Wheel initial braking torque</td>
<td>user</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Controller variable</td>
<td>User estimation</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Effective rolling radius</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Differential longitudinal tire force</td>
<td>Controller part</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Differential braking torque for wheel</td>
<td></td>
<td>Wheel rotation calculation block</td>
</tr>
</tbody>
</table>

The value of $a$ and $b$ is difficult to determine without analysis as possible optimum values are acceptable for this controller. Analysis is shown by simulating with different values of $a$. Detail of this analysis is included in the simulation section of this chapter.

**Wheel rotational dynamics**

Wheel angular velocity or wheel rotational dynamics is related to mechanical driving and braking torque. These relations are rewritten from equation (5.21) as:

$$
\omega = \int \frac{1}{J_w} (T_d - T_b - r_{eff} F_x)
$$

(5.26)

Here, $J_w$ is the wheel inertia, $\dot{\omega}$ is the angular acceleration, $T_d$ is the drive torque of the wheel, $F_x$ is the wheel longitudinal force and $r_{eff}$ is the effective radius of wheel.

As wheel torque can be expressed in terms of wheel velocity, the difference in torque between two wheels can be determined from the control law. Wheel angular velocity from torque is calculated to control the wheel rotation via the hardware interface.
Figure 5.13 wheel rotation calculation block

Table 5-10 Numbered labels in figure 5.13 describe the attributes of input and output in parameters require wheel rotation calculation

<table>
<thead>
<tr>
<th>Label</th>
<th>Name</th>
<th>Input from</th>
<th>Output going to</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Wheel inertia</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Wheel torque</td>
<td>wheel torque calculation</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Effective rolling radius</td>
<td>Vehicle properties</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Longitudinal tire force</td>
<td>Wheel block</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Brake torque for wheel</td>
<td>User/simulation</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Wheel angular velocity</td>
<td></td>
<td>Wheel block</td>
</tr>
</tbody>
</table>

5.5 Simulation

Simulation is done here in for different situations they are normal operation, stability test without the controller and stability test with controller. In these cases of simulations, vehicle parameter is used as given in the table below. The parameters chosen here are similar to a large vehicle like a standard SUV but different parameters can be used to simulate the behaviour of any passenger vehicle.
### Table 5-11 Parameter used in simulation

<table>
<thead>
<tr>
<th>Vehicle Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unsprang Mass</td>
<td>1500 Kg</td>
</tr>
<tr>
<td>Yaw inertia</td>
<td>2480 kg-m²</td>
</tr>
<tr>
<td>Distance of front axle form CoG, L_f</td>
<td>1.180 m</td>
</tr>
<tr>
<td>Distance of rear axle form CoG, L_r</td>
<td>1.770 m</td>
</tr>
<tr>
<td>CoG height</td>
<td>0.700 m</td>
</tr>
<tr>
<td>Track Width</td>
<td>1.800 m</td>
</tr>
<tr>
<td>Frontal area</td>
<td>1.6 m²</td>
</tr>
<tr>
<td>Coefficient of Aerodynamic Resistance</td>
<td>0.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tire Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effective rolling radius</td>
<td>0.380 m</td>
</tr>
<tr>
<td>Cornering stiffness</td>
<td>80000</td>
</tr>
<tr>
<td>Longitudinal stiffness</td>
<td>20000</td>
</tr>
<tr>
<td>Wheel inertia</td>
<td>2.166 kg-m²</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Road and wind parameters</th>
<th>values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire-road friction coefficient μ</td>
<td>0.85</td>
</tr>
<tr>
<td>Rolling resistance coefficient</td>
<td>0.015</td>
</tr>
<tr>
<td>Wind velocity</td>
<td>1 m/s</td>
</tr>
<tr>
<td>Air density</td>
<td>1.22</td>
</tr>
</tbody>
</table>

### 5.5.1 Vehicle simulation for longitudinal motion

In the first part of the simulation the vehicle is driven to show the different dynamics of the vehicle and the wheels. In this part of the simulation, the vehicle is moved forward for 10 seconds with zero steering angle and initial torque of 50 NM in each of the wheels. Output from this normal operation is given below to evaluate the behaviour of the vehicle. Initially the vehicle speed was 80 Km/h. Output from this simulation is described in the resultant graphs below.
The steering angle is an input for the simulation and it is kept at zero in this simulation which is shown in the Figure 5.14.

Another input for this simulation is wheel driving torque as shown in the Figure 5.15. This pulse-like wheel torque is the applied to each of the four wheels as in this study we have identical in-wheel motors.
Some output like vehicle velocities, wheel forces, wheels angular velocity and yaw rate is given in the following figures and discussed.

![Figure 5.16 Vehicle longitudinal velocity](image)

Figure 5.16 Vehicle longitudinal velocity

Longitudinal velocity of the vehicle is given in figure Figure 5.16. This shows that the vehicle started getting slower from the initial speed of 22.2 m/s as no driving torque was applied to the wheels. The vehicle started gaining velocity as the driving torque was applied. At the end this vehicle slowed down when torque was not applied. This behavior shows that the vehicle model is working as expected.

The vehicle behavior for the given input is valid and it is seen by observing the lateral motion of the vehicle. The lateral vehicle motion is shown in Figure 5.17 which shows there is no lateral velocity. This is due to the steering angle input which was kept at zero for this entire simulation.
In the Figure 5.18 (a), longitudinal wheel forces are given, these are outputs from the wheel blocks. Wheel force is derived from wheel slip ratio which depends on braking or driving torque. Wheel longitudinal force exists in this figure when driving torque is applied. Figure 5.18 (b) shows there is no wheel lateral force which justifies the validity of the vehicle motion for the input provided.
Wheel angular velocity is an important output which expresses the wheel’s rotational characteristics. The primary idea of the proposed method is to manipulate the wheel rotation to gain control of the vehicle direction. Figure 5.19 shows the wheel angular velocity of the four wheels. This output shows that wheel angular velocity has increased slowly at the time when driving torque is applied. Wheel velocity has decreased. This is similar to the behavior of the vehicle velocity we saw Figure 5.16. The proposed vehicle model includes the rolling resistance and air drag force. Due to these opposing forces vehicle velocity has decreased as well as wheel angular velocity. This is included in the initial vehicle simulation to observe the vehicle characteristics. In the simulation for stability control manoeuver, these opposing forces are not considered, assuming that a speed controller is applied for a constant speed.
From the behaviour of the vehicle shown in Figure 5.20, it is seen that the vehicle is performing normal operations without any inaccuracy as there is no change in direction of the vehicle due to zero steering angle. Also the vehicle responded to the input driving torque. From this simulation result it is concluded that this vehicle model has no basic problems and is suitable for performing any kind of simulation for stability.

5.5.2 Vehicle Simulation for Stability

Simulation for stability of the vehicle is done here in two different parts. At first an open loop simulation is done to compare the actual vehicle states with the nominal or expected values of the vehicle dynamics. In the second part a closed loop simulation is done to compare the actual and nominal vehicle dynamics with the calculated differential torque. Both of these simulations are described in the next sub sections.

For stability control simulation, properties of the vehicle, wheel, road-tire friction coefficient and wind velocity are required as input in the other part of the simulation. To perform the stability test on this vehicle, we initially used the guide line FMVSS 126. It is a prescribed test for vehicle stability.
In this test a series of steering angles, as shown in the Figure 5.21, are provided as input for the vehicle and then analysis is done on the outputs like steering angle, yaw rate and lateral displacement.

![Steering angle series](image1)

**Figure 5.21 Steering angle series**

![Wheel driving torque](image2)

**Figure 5.22 Wheel driving torque**

Figure 5.22 shows the wheel torque which is provided as an input in this simulation.
Figure 5.23 Vehicle longitudinal velocity without ESC in FMVSS 126

Figure 5.23 shows the vehicle longitudinal velocity which decreases as the vehicle turns. In Figure 5.24 expected and actual yaw rate are plotted to observe the difference in case of a manoeuvre similar to the FMVSS126.
Here, the main component yaw rates are brought into focus to identify the vehicle stability without presenting other parameters. From the output Figure 5.24 it is seen that the output series are difficult to analyze without emphasizing a particular case. To avoid this difficulty, open loop simulation and closed loop simulation is considered as a single case of “Sine with dwell” in the later subsections. To evaluate the simulation output we have reference values of yaw rate and slip angle, these reference values are used to compare the expected results of yaw rate and slip angle. These will help to determine the vehicle stability.

5.5.2.1 Open loop simulation

In this stage, an open loop of simulation is done to observe the vehicle responses with a sine with dwell steering angle as input, as shown in Figure 5.25. The previous test with the steering angle series helped to find a proper sine with dwell steering angle input that can create some instability in the vehicle.

![Figure 5.25 Sine with Dwell Steering input for open loop test](image)

In Figure 5.26 the provided wheel driving torque is shown. Wheel torque is kept at zero after an initial peak and then the vehicle motion was observed in longitudinal and lateral axes to identify the instability and to determine required longitudinal force difference.
This instability is taken from the vehicle yaw rate and vehicle slip angle. Differences in expected vehicle responses with the nominal values are taken as the controller input to generate longitudinal wheel force differences.

Figure 5.26 Driving wheel torque in open loop

Figure 5.27 Vehicle longitudinal velocity in open loop simulation
Resultant longitudinal velocity of the vehicle is given in Figure 5.27 which shows a change in velocity due to the steering angle input. Changes in front and rear wheel forces due to steering angle are seen from the longitudinal wheel forces given in Figure 5.28 (a).

![Figure 5.28 (a) Longitudinal wheel forces and (b) Wheel angular velocity in open loop simulation](image)

These forces affected the wheel angular velocity which can be observed from the output of the wheels angular velocity as given in Figure 5.28 (b). The method we have used here to determine the wheel force is dependent on wheel slip ratio and wheel angular velocity. Wheel longitudinal force, wheel angular velocity and wheel slip ratio are strongly related to each other in this simulation. They are in a loop which was a major challenge technically.

Using the vehicle dynamics generated from the model, the controller has determined the expected difference in longitudinal wheel forces between the right and left wheel to assist vehicle stability as shown in figure Figure 5.29.
Figure 5.29 Expected longitudinal wheel force to minimize instability

Figure 5.30 shows the comparison of actual yaw rate from vehicle body block, calculated desired yaw rate and targeted yaw rate.

Figure 5.30 Desired, targeted and actual yaw rate in open loop simulation
From Figure 5.30 it is seen that the actual yaw rate is not converging with the desired and targeted yaw rate this means the vehicle is slightly unstable in following the direction.

The following Figure 5.31, shows vehicle slip angle is much lower than the expected slip but not entirely zero which indicates an instability.

Figure 5.31 Desired, targeted and actual vehicle slip angle in open loop simulation

5.5.2.2 Closed loop simulation

The closed loop simulation is performed to measure the vehicle responses from this we calculate the required differential torque so that the differential torque can create the expected longitudinal force difference between the wheels and make the vehicle more stable in terms of changing direction. As one input of this closed loop simulation the same steering profile has been used as in Figure 5.25, which was used for the open loop. The output of the closed loop simulation, longitudinal wheel forces are shown in Figure 5.32 (a) and angular velocity of the wheels are shown in Figure 5.32 (b).
Longitudinal wheel force and wheel angular velocity is dependent on the calculated differential wheel torque. It is seen that there are force differences in the left and right wheels as well as the wheel angular velocity. These differences in angular wheel velocity are expected to be created by the lower level controller. This controls the wheel speed according to the required wheel angular velocity as shown here. Another control PID loop will listen to the expected angular velocity or wheel speed and perform control intervention by changing the voltage and current of the motor. In the next chapter this process is described in details.
Figure 5.33 Desired, targeted and actual yaw rate in closed loop simulation

From the above figure it is observed that actual yaw rate is within the limit of the targeted yaw rate and almost similar to the expected yaw rate. This indicates an improvement in yaw rate to stabilize the vehicle.

Figure 5.34 Desired, targeted and actual vehicle slip angle in closed loop simulation
Figure 5.34 compares desired, targeted and actual vehicle slip angle. The actual slip angle is not zero but it is within the expected limit.

**Figure 5.35 Differential Torque in closed loop simulation**

The calculated wheel driving torque which is used as a feedback control input for the vehicle is presented in Figure 5.35.
Finally the vehicle longitudinal velocity is shown in Figure 5.36 which shows an increase in vehicle velocity.

![Figure 5.36 Vehicle velocity in closed loop simulation](image)

5.6 Performance of the proposed system

It is difficult to compare the proposed ESC method used in this four in-wheel motor electric vehicle with the other ESC systems. The other ESC systems are mostly used in conventional electric vehicles. Some ESC systems have been used in similar drive train described in this work. We used those ESC systems reported here to make comparison. In Chapter 4, we discussed several different methods to provide a similar solution. A comparison of this proposed method with the methods reported in this thesis is provided here to evaluate the overall performance. In this section, we have compared vehicle longitudinal velocity and expected yaw rate of the proposed system with the other systems.
To evaluate the performance in achieving expected yaw rate, we have compared this proposed ESC method with other methods reported in this literature. A linear quadratic regulator based stability control with a sliding mode wheel slip controller is presented in [71] to control the yaw rate. The goal of this research work was to develop a Vehicle Dynamics Control system (VDC) system for improving vehicle dynamic stability. From this research work, yaw rate response from simulation result is given in Figure 5.37.

![Figure 5.37 Yaw rate response of a linear quadratic regulator based controller with sliding mode wheel slip controller](image)

Another motor torque based fuzzy controller [72] is considered for comparison. The responses of vehicle with four-wheel-drive electric vehicle ESP (EV ESP) from simulation result of this system is given in Figure 5.38.

![Figure 5.38 Yaw rate response of a torque based ESC with fuzzy controller](image)
The yaw rate response from the simulation of the proposed method is given in Figure 5.39.

Figure 5.39 Yaw rate response from the simulation of the proposed method

As a standard test, a sine with dwell test has been performed the simulations we have included in our discussion. The controller should keep the yaw rate of the vehicle close to the expected yaw rate. From the beginning, the rise in the yaw rate should be tracked and at the end of this sinusoidal movement, yaw rate should become zero. It is found that the controller maintains the yaw rate close to the targeted yaw rate. Comparing the results of the proposed method with other results, it can be said that the proposed controller has maintained the yaw rate close to the expected yaw rate. Hence, the proposed controller for the electric vehicle with four in wheel motors is performing the job properly.

We have compared different types of ESC in section 3.5 and then selected one type. One of the significant disadvantages in conventional ESC is to get the expected longitudinal response. From the simulation result, vehicle longitudinal velocity and yaw rate of the proposed system is presented in Figure 5.40.
Figure 5.40 Vehicle longitudinal velocity of the proposed system

The vehicle velocity of the proposed method can be compared with a motor torque based fuzzy controller for vehicle stability presented in [72]. Simulation result in Figure 5.41 shows the vehicle velocity decreases when the controller has performed a yaw control. From these comparisons for vehicle velocity it is clear that the vehicle is getting better longitudinal velocity at the moment of yaw rate control, compared to the other methods.
5.7 Summary

The simulation shows improvement in vehicle dynamics in critical manoeuvres like sine with dwell. In this simulation vehicle dynamics as well as wheel dynamics are presented in detail to see how the vehicle performs. Wheel torque is calculated and provides a feedback input in the plant vehicle. In this simulation, differential driving torque has been calculated using equation (5.29) and (5.30). Here, control variable ‘a’ has been considered as low as 0.001. However, it has been reported that the value of this control variable ‘a’ should be limited from 0 to 1 in case of differential brake pressure [13]. Nevertheless, this simulation has been performed with several values of the control variable. It shows that better controllability and stability is found at 0.001 with possible maximum driving torque ranges. This improvement in the vehicle is created by changing the applied driving torque only. For a more critical situation, differential
braking is expected to be used in the future. Simulation can be used with designing a
desired vehicle for optimum response in handling manoeuvres. In the next Chapter an
experiment has been performed to generate differential wheel torque in terms of wheel
velocity which we simulated here.
Chapter 6
Hardware setup and experiments
6 Hardware setup and experiments

6.1 Introduction

To assess the possibility of making this proposed stability control method work in a four in-wheel motors electric vehicle, and to identify the potential challenges, experiments have been done at different stages and the results are included in this chapter.

After finishing the basic development of a four in-wheel vehicle-like platform, normal driving tests like following a circular path with acceleration, and following a straight path with braking and acceleration were performed. A real life stability test with the developed platform on a driving track was far more complicated and expensive than it was expected to be at the start of the experiment. It requires precise input of steering angle over time, an extra sensor to realize vehicle conditions, and time and human resources. Overall it requires a much large input of resources than anticipated.

Instead of a test on the track, a different approach is taken to understand the performance of the proposed stability controller. A hardware test was done using the digital controller and in-wheel motor to observe the controller demand and the responses of the actuations. In this test the important vehicle physical properties for instance weight and CoG are measured and determined. The behaviour of the in-wheel motors is also tested. In this chapter the measurements for the experiment, experiment setup, experiment results and challenges are discussed in the different sections.

6.2 Proposed hardware system for experiment

The simulation results in chapter 5, shows that the differential torque can improve vehicle dynamics in case of manoeuvres. At the top level a reference vehicle model is used to provide dynamic responses like velocities and yaw rate to the upper level controller and that the upper level controller provides the required longitudinal force differences. The lower level of the controller generates the expected differential driving torque of the four wheels. From the required corrective differential driving torque, wheel angular velocity is derived and plotted.
The differential wheel angular velocity or differential wheel rotations for four wheels are the generated demand from top level of the controller to the lower level of the controller. In the experiment, the reference vehicle is created using LabVIEW code in Virtual Instrument (VI) so that it can be written in the digital controller which mimics the vehicle response according to the steering angle input. This virtual reference vehicle provides the necessary inputs like yaw rate, expected yaw rate, longitudinal and lateral velocities and expected slip angle to the controller, and the controller generates the required wheel rotation. The system with the proposed testing method is shown in the Figure 6.1. Here the input for steering angle, vehicle velocities and yaw rate are considered from the LabView environment as well as from the physical world so that the controller can be used for testing and real-time operation.

Figure 6.1 Proposed experiment setup

To meet the demand, actuation is done by the lower level controller operating the in-wheel motor using an embedded interface in the digital controller.
Based on these concepts, experiments have been carried out for a wheel speed controller to meet the demand using an in-wheel test rig. The advantages of this test is, we can observe the possible expected output on the test bench without going to the driving track which saves time and cost. This method of testing can be used to check the functionality of the stability controller before operation [73]. To do this bench test the in-wheel platform is measured to determine its physical properties as shown below.

### 6.3 Measurement of experimental platform

The measuring the vehicle’s physical properties like normal load on wheels, CoG, height of CoG, wheel base and track width are important parameters to be included in the digital controller. In order to do that the vehicle weight is measured using scale pads under the wheels and then the CoG is determined [74].

Figure 6.2 Physical measurement of the four in-wheel platform

This figure shows the measurement of the vehicle to determine the CoG and CoG height. A manual crane is used to determine the CoG height as shown in (d).
CoG height is also determined after measuring the vehicle using scale pads which lift the rear axle. These measured parameters of the unloaded vehicle are included in the table below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Track width</td>
<td>106 mm</td>
</tr>
<tr>
<td>Wheel base</td>
<td>178 mm</td>
</tr>
<tr>
<td>Load on front left wheel</td>
<td>53 kg</td>
</tr>
<tr>
<td>Load on front right wheel</td>
<td>52 kg</td>
</tr>
<tr>
<td>Load on rear left wheel</td>
<td>86 kg</td>
</tr>
<tr>
<td>Load on rear right wheel</td>
<td>84 kg</td>
</tr>
<tr>
<td>Total load on front axle lifting rear axle</td>
<td>114 kg</td>
</tr>
<tr>
<td>Total weight of the vehicle</td>
<td>275 kg</td>
</tr>
<tr>
<td>CoG longitudinal from front axle</td>
<td>1103.6 mm</td>
</tr>
<tr>
<td>CoG Height from ground</td>
<td>345 mm</td>
</tr>
</tbody>
</table>

### 6.3.1 In-wheel test Rig

An experimental setup is created as a wheel test rig where in-wheel motors are tested. In this setup a multi-channel digital oscilloscope is used to measure the change of wheel speed and the change of input voltages of the motor over time. A customized filter is used to get the actual frequency of the hall-effect sensor. The motor driver is connected to the motor and powered by a power supply which can be varied from 48 volts to 60 volts.
Figure 6.3 Current measurement of the in-wheel motor to determine the electrical torque

To measure the current consumption, a shunt with 50m ohm and 100A tolerance is used in between the battery and motors as shown in the Figure 6.3. A low value capacitor is connected across the shunt to get a stable output value of voltage drop across the shunt. This voltage drops across the shunt changing with the current flow. From the measured current the electrical torque of the wheel is measured for further comparison [75]. The wheel control test rig is connected to the FPGA controller and a PC with the required software environment.

Figure 6.4 In-wheel test rig for observation and control of the wheel
6.3.2 Observation of the throttle pedal signal to a desired wheel velocity

Transforming the throttle pedal signal in terms of voltages to a desired wheel velocity is very complex and involves tested data of the motors with all possible throttle inputs [76]. A comprehensive experiment using several identical in-wheel motors, motor drivers and throttle. An experiment has been conducted on all the motors and controllers to find the wheel speed provided by the internal Hall Effect sensor signals for a certain throttle voltage input [77]. The experiment was done using four identical in-wheel motors and four drivers for these motors. Observation firstly is done here with a single motor driver and then with the four identical motors. In the second observation the drivers only was changed. All the possible values of throttle voltage that can be generated by the physical throttle pedal, and rotation of the motor is collected by the internal Hall Effect sensor of the motor.

Figure 6.5 Test data of motors shows similarity in responses against throttle input voltages varying from 0.8 V to 3.6 V
Figure 6.5 shows each set of data collected by running the throttle voltages from 0.8Volt to 3.8Volt. The input voltage is increased gradually in each set of data to observe the frequency and the period of the signal under the supplied voltage. This procedure is repeated across 25 sets using the setup mentioned above. This experimental data is to improve performance by finding the required throttle input for a certain wheel speed. Testing has been done to find the motor and controller combination pair to get the most desirable output.

6.4 Digital Controller

6.4.1 Embedded host with I/O and programming environment

In order to develop a portable, real-time, accurate controller for the 4 in-wheel motor electric vehicle, a National Instruments Compact Real-time Input Output (NI cRIO) device was used. This programmable automation controller is used in different real-time control applications as it is known to be a robust technology [78, 79]. The NI cRIO device is a modular device which is reconfigurable with analogue and digital modules for both input and output. The device allows a program developed in NI-LabVIEW to be deployed to the device to be run in real-time independently of a computer. The NI cRIO device and modules used are as follows.

- NI cRIO-9004 chassis
- NI 9401 Digital Input/output Module
- NI 9201 Analogue Input Module
- NI 9263 Analogue Output Module

The NI cRIO-9004 chassis is a field-programmable gate array (FPGA) system, and the modules used with it are an 8 channel digital input/output, 4 channel analogue input, and 4 channel analogue output modules respectively. An FPGA system such as this has advantages and disadvantages over a standard microprocessor. Due to the fact that an FPGA system uses dedicated hardware for processing logic, much faster simultaneous execution of multiple processing threads is possible when compared to a standard microprocessor [80].
It also allows higher performance with hardware to perform specific logic and computations in a single processing clock, rather than a microprocessor which may have to perform the calculation over multiple clocks.

One disadvantage however is that certain data types are not able to be handled by the FPGA device. Things like floating point numbers and variable length arrays are incompatible as the hardware doesn’t have the capacity to process them. A workaround for floating point numbers on an FPGA device is the use of the fixed point number system. Fixed point numbers represent a decimal number as two integers. The first integer represents the whole number component to the left of the decimal point, and the second integer represents the decimal component to the right of the decimal point. However, as a result of the number being represented in this way, the accuracy of the decimal component is limited by the number of bits available in the integer representing it. As a result, integration and derivation of continuous numbers becomes immensely complex, and isn’t always feasible. Additionally, existing control algorithms that use floating point numbers may be useable on FPGA devices after including some number system conversion functions. Otherwise it may not be possible to use higher level of algorithm in this kind of FPGA.

The program used for developing the controller was NI LabVIEW 2011. Labview is a development environment for the graphical programming language “G”. In this graphical programming language, data flow during execution is defined by a block diagram in which functions are joined together with lines representing data flow. Once past the learning curve, LabVIEW allows for simple and effective programming of a multithreaded FPGA deployable program. It also allows for a layered programming approach. Sub tasks, or functions are easily exchangeable if the input and output variables are of the same data type. These functions are generally contained in Virtual Instruments or VIs which can be called from other VIs to give the program a layered structure.
6.4.2 Feed-back controller for wheel

6.4.2.1 Requirement of a controller

For a higher level control system such as vehicle stability control or traction control to perform efficiently and effectively, the lower level control of the motor needs to be both accurate and fast. Without an accurate and highly responsive motor controller, the efficient high level control of this vehicle will not be possible. This experiment aims primarily to provide a simple interface for a higher level control system for the four in-wheel motors.

6.4.2.2 Challenges

Vehicles with a centralised motor achieve this passively with a differential, which allows the powered wheels to spin at different speeds when requires stabilization. With a 4 in-wheel motor vehicle, this difference in wheel speed needs to be actively controlled by the controller. Control of a four in-wheel motor vehicle has many potential complications. Firstly, the controller needs to be able to accurately control four motors to the exact speed as demanded by the top level controller in normal operation and in other scenarios. If the speeds of the wheels are off, even by a small amount, the vehicle will have a natural tendency to turn unexpectedly. This creates instability which will in turn result in a loss of performance and extra wear on the tyres.

Another potential complication with a 4 -in-wheel motor vehicle is the speed of change. If the motor is too slow to react to a change in demand, then not only will the performance of the higher level controller be impacted, but the overall safety of the vehicle may be jeopardised. Implementation of digital controllers in an embedded environment is difficult because of the inherent problems related to analogue and digital signals interfacing in hard-real-time, therefore, the control algorithms sometimes use approximations [81]. The most important and challenging issue is existing control algorithms which use floating point numbers may not be useable on a FPGA devices. Consequently, a decision was made to develop a discrete-time PID controller from scratch.
There are many ways of developing a control algorithm, each has its own advantages and disadvantages. The two most popular methods for control are PID and Fuzzy logic. Fuzzy logic uses degrees of truth of inputs to generate an output [82]. In terms of control of an electric motor, fuzzy logic can be used to determine whether the motors are spinning faster than the desired speed, slower than the desired speed, at the desired speed, or a partial truth or combination of any of these. However, fuzzy logic, while good for determining truths, is not ideal for precision control. Fuzzy logic is however useful in a higher level controller to determine which action to take when the vehicle is cornering, or for something such as stability control to determine which motors are slipping, and the corrective action to rectify the situation.

PID control is by far the most widely used control method in use today [83]. In case of ESC, a PID controller is chosen for the actuation in the lower level control in most of the popular methods. One of the examples where a test rig used with a PID controller for actuation seen in [84]. This is partly because the performance of the control method and the relative simplicity of the application of a PID controller.

PID controllers are named as such because of the three components making up the controller. These components are the Proportional error, Integral error, and Derivative error. These three components are then summed to create the controller output. A PID controller takes a feedback variable and adjusts the output of the controller so that the feedback variable reaches a specified set point.

The controller is able to provide an accurate closed loop control by tuning. This is done by adjusting three coefficients which multiply each of the proportional, integral and derivative components of the controller in order to achieve the desired properties of the controller output, i.e. accuracy, rise time, settling time, overshoot, and stability. These coefficients are commonly referred to as $K_p$, $K_i$ and $K_d$ for the proportional, integral and derivative components respectively.
For use in motor control, the PID can be used to adjust the input voltage and current to
the motor and monitor an output sensor or encoder to set the motor at the desired speed.
PID control is most useful for creating an interface between a smarter, higher level
controller and allowing this controller to set a specific speed for the motor to rotate. The
proportional, integral and derivative components of the controller each have a different
effect on the performance of the motor output. Increasing the gain of each component
has the effect described in the Table 6-2 [85].

Table 6-2 PID controller tuning effects

<table>
<thead>
<tr>
<th></th>
<th>Rise Time</th>
<th>Overshoot</th>
<th>Settling Time</th>
<th>Steady State Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Proportional</td>
<td>Decrease</td>
<td>Increase</td>
<td>No effect</td>
<td>Decrease</td>
</tr>
<tr>
<td>Integral</td>
<td>Decrease</td>
<td>Increase</td>
<td>Increase</td>
<td>Eliminate</td>
</tr>
<tr>
<td>Derivative</td>
<td>No effect</td>
<td>Decrease</td>
<td>Decrease</td>
<td>No effect</td>
</tr>
</tbody>
</table>

6.4.2.4 In-wheel motor and driver

The motors currently used in the vehicle come with a simple motor controller. The
motor driver takes an input of 0 - 3.6V, and outputs a DC voltage of 48V or 60V. These
simple drivers do not allow for feedback, or for accurate control of the motors. The
motor itself takes a 48V or 60V DC input. The motor also contains a Hall Effect sensor
which generates a pulse output. The Hall Effect generator creates 23 pulses for each
revolution of the motor. The pulses have a direct, linear relationship to the number of
revolutions. This feedback pulse can be then used as a control variable for the
controller.
6.4.2.5 Filtering motor feedback signal

To remove noise from the feedback pulse, the signal is passed through a simple low pass filter. This can be done in the software, but generally, a physical filter is easier and faster to make [86]. The low pass filter that was used was made using a capacitor and two resistors as shown below.

![R-C filter diagram](image)

Figure 6.7 R-C filter to remove noise from hall sensor feedback signal

For a given situation, the resistance and capacitance will vary. The resistors are required to adjust the voltage across signal output, and also to prevent a short circuit. The difference between the resistances $R_1$ and $R_2$ should be adjusted so that the voltage across $R_2$ is the required input voltage to the digital input module.
In the case of this experiment, that input voltage is 10V, so the resistances were adjusted accordingly. The capacitor filters out any high frequency noise in the signal. If the capacitance in the capacitor is too high, then the output signal will be distorted. If the capacitance in the capacitor is too low, then the noise on the signal will be unable to be filtered out, and some residual noise will remain on the output signal. It is therefore necessary to test and check the shape of the output signal to ensure that it doesn’t carry any residual noise. The shape of the signal is otherwise consistent with the original signal and not significantly distorted. After checking and testing multiple capacitors a capacitor with a suitable capacitance was selected.

6.4.2.6 Programming the PID controller

Program is developed that works as a PID controller. It includes several subroutines which are described below as different functions.

![Digital PID controller for wheel control with the subroutines](image)

**Figure 6.8 Digital PID controller for wheel control with the subroutines**
These functions or subroutines collectively work as a digital controller that controls the speed of the in-wheel motors. Figure 6.8 shows the proposed wheel speed controller with the associated subroutines. The associated subroutines or functions are described below.

**Function for sensing the number of pulses:**

As the filtered pulse is essentially a square wave, it can be read using the digital input module. This also helps remove any minor inconsistencies as the digital input uses thresholds to determine whether the signal is high or low, and transcribes this to a Boolean true or false. In LabVIEW, code was generated in a Virtual Instrument to read this series of Boolean pulses, and to determine first the period of the signal, and from there, determine the frequency of the pulses.

![Figure 6.9 function for In-coming pulse counting of frequency counter for hall sensor](image)

The above function takes one input (the Boolean pulse) and gives one output (the determined frequency of the pulse). In order to do this, it uses a tick counter, which
generates a tick every $\mu$ second. This clock tick is stored in a variable which updates when the input Boolean changes from yes to no, or no to yes. This variable is used to calculate the high time, and the low time of the pulse, by calculating the difference between the current and previous values of the clock tick variable on either the leading or trailing edge of the pulse. The sum of these two times gives the period of the wave. With the clock tick set at 1 $\mu$ second, this period is in $\mu$ seconds. In order to determine the frequency, this number is first converted into seconds, and then the period is reciprocated to give the frequency. This frequency has a direct linear relationship to the speed of the motor, and as such, is ideal for use as the process variable of the PID controller. This frequency is assumed as the speed of the wheel in the experiment. For exact speed control, a higher level controller can use this variable, along with the relationship between the number of pulses per revolution, and the circumference of the wheel to determine the velocity of the surface of the tire as it moves along the road. As this will vary depending on the motor, and the size of the wheel, it has been left to a higher level controller, or an intermediary controller to determine this.

Once the pulse is translated into a frequency, it can be passed as the process variable to the PID controller. The PID controller is made up of 3 components they are proportional, integral and derivative. These three components are added together to give the controller output.

**Proportional component**

Firstly, and most simply the proportional component is a linear multiple of the error between the set-point (demand), and the process variable (actual). In a discrete system, this is triggered by a sampling time. This simple function generates the proportional component of the PID controller. First it calculates the difference between the set-point and the process variable. These two parameters are passed to it from the calling VI, which are in turn defined by demanded speed, and the pulse respectively. This error is then multiplied by the proportional gain. Finally, the output of the proportional component of the program is set to only update when triggered by the calling VI.
Figure 6.10 Proportional component of the PID controller

The proportional component of the PID controller will have the following effects on the controller performance.

- Decrease rise time
- Increase Overshoot
- Decrease Steady State Error
- Minor effect on settling time

**Integral component**

The second component of the PID controller is the integral. The integral component in a discrete PID controller is effectively a cumulative form of the proportional controller. It attempts to add control based on cumulative previous values of error. Unlike the proportional controller, the integral component of the controller accumulates over time, and will take into account the current error as well as all previous values of the error.
This function works in a similar way to the function that calculates the proportional component. It takes the difference between the set-point and the process variable, and multiplies it by a constant gain. Next, rather than simply updating when triggered by the calling VI, this value is updated, and added to its previous value creates the integral component. Finally, because the integral component is cumulative over time, a reset trigger is needed. This reset will set the output of the integral value to zero. The reset is useful because if the PID controller is not responsive the integral component can regress over time. The integral component of the PID controller will have the following effects on the controller performance.

- Decrease rise time
- Increase Overshoot
- Eliminate Steady State Error
- Increase settling time

**Derivative component**

The third and final component on the PID controller is the derivative. The derivative component attempts to predict future values of error, for use in the control algorithm.
This makes the derivative component of the controller the most complicated component to calculate. In a discrete system, the true derivative must be calculated using a backwards difference equation.

\[ \frac{e(t) - e(t-1)}{T_s} \]  

(6.1)

Again, the output is multiplied by a constant Derivative gain, and is only set when triggered by the calling VI. Additionally, a reset is needed to clear the previous error value so as not to distort the current derivative component when resetting the controller.
The integral component of the PID controller will have the following effects on the controller performance.

- Decrease Overshoot
- Decrease settling time
- Minor effect on rise time
- Minor effect on steady state error

**Collective proportional, integral and derivative functions**

To bring these three components together to form a PID controller, the following function was developed.

![Figure 6.13 Merging function for proportional, integral and derivative components](image)

This function takes the process variable and set point from its calling function, and passes them through to each of the proportional, integral, and derivative calculators. Also passed to these calculations VIs are the proportional, integral, and derivative Gain respectively. These constants are again passed to this function from the VI that calls it.
The other part of this VI is the calculation and implementation of the sampling time. Because of the way that LabVIEW works, this function is passed by a clock tick from the function that calls it, along with a sampling time in seconds. These are both passed to another function which generates a pulse once every sample.

![Diagram of calculation and implementation of the sampling time](image)

**Figure 6.14 Calculation and implementation of the sampling time**

The easiest way to do this in LabVIEW is by converting the sampling time to the same time as a single clock tick (in this case a micro-Second) using the remainder of an integer division to generate a single pulse when the remainder is zero.

**Individual motor control function**

In order to keep the system modular when controlling multiple motors, another function was created to control a single motor just by linking the appropriate digital input, analogue output, and the necessary tuning parameters for the PID controller.

The components making up this function are the pulse reader, the PID controller, and an accumulator finally.
The accumulator is necessary to give the motor driver a voltage which corresponds to
the desired speed of the motor, rather than a value relative to the difference between
the desired and actual speeds.

The accumulator takes the output from the PID controller, and updates once per sample.
The accumulator has two functions. Firstly, it takes the output of the PID and
continuously accumulates it over time. Secondly it limits the output which is tied to the
analogue output module. This prevents the controller from overloading the motor driver,
and also prevents the controller from accumulating too far in one direction.
Additionally, by adjusting the minimum output voltage so that the motor is powered, but not enough to overcome its inertia, the initial start time from zero motion can be improved slightly.

**Overall system**

Bringing it all together, with 4 single motor controls, 4 motors, the motor drivers, and simple low pass filters on each digital feedback pulse, the following system can be put together. The system is designed such that each of the higher level controllers can interact with the interface controller. The higher level controllers can be developed either on the same FPGA device, or separately, and interface with the FPGA device using an analogue or digital input module. Additionally, any higher level controller only needs to set a demand for the motor controller, and the interface controller will take care of the monitoring and control of each motor individually to ensure they reach, and remain at the demanded speed.

![Figure 6.17 Overall system for motor control using digital controller](image)
6.4.3 Results and discussion

The tuning, and testing of this controller are heavily interlinked, as they are both dependent on each other. In order to prove the success of the controller, the requirements need to be quantified. The requirements can be broken down to speed and accuracy. The speed of the system can be quantified by the rise time, and the settling time, while the accuracy can be explained by overshoot and steady state error. The best indicators are the settling time and the steady state error. The tuning process involves testing adjusting, and retesting [87]. This can be a long drawn out process, especially if the effects of each component are not researched beforehand.

The generally accepted method for tuning the PID controller is to first start with a proportional only controller. The proportional gain is adjusted to give the controller the desired rise time. Secondly, the derivative component is added. The derivative component has the effect of decreasing settling time, and decreasing the overshoot, while having a very minor, insignificant effect on the steady state error and the rise time. Finally, the integral component is added in. The integral component has the effect of eliminating the steady state error, but the cost of this is that the settling time and rise time are increased. Finding the balance between settling time and steady state error is the trickiest part of the testing process. Depending on the situation, it usually comes down to a decision between which is more important for the application of the controller. In the case of this application, response time is the most critical, provided the steady state error is relatively low (<0.5RPM).
In order to find an initial value for the proportional gain, some information needs to be known about the relationship between the process variable and the controller output. In order to determine this, the motor was tested without the controller by setting an output voltage, and monitoring the feedback frequency. This test is done on all the motors as explained in previously in section 6.3.2 of this chapter. From those observations, general characteristic data was obtained for all the motors as shown in Figure 6.18.

As can be seen from this graph, excepting the initial voltage where the motor is stationary, the relationship between the voltage throttle and feedback frequency is more or less linear. Therefore, by selecting two points and calculating the gradient between them, a rough starting point for the proportional gain can be determined. The gradient of this line is roughly 115 Hz / V, which results in an initial proportional gain somewhere in the vicinity of 0.008. 
With this in mind, the controller was initialised with a proportional gain of 0.008, and integral and derivative gains of zero. The result was a controller which had a very quick rise time, but conversely, a very high overshoot, which lead to a relatively high settling time. As the system allows for live adjustment of the tuning variables, the proportional gain was adjusted to 0.005. This still gave a very quick rise time, but also gave considerably less overshoot. After initially taking a long time to settle, the steady state error was very low, and was in the vicinity of 1 - 2 Hz, which relates to a wheel speed of around 0.05RPM which is well within the requirements.

The second component to be added was the derivative. With the controller still running, the derivative gain was adjusted, initially to a very low 0.001. This had a clear and obvious effect on the settling time and overshoot of the controller output while not having a significant effect on either the rise time or the steady state error. Thirdly, the integral gain was adjusted, again to 0.001. This integral gain had a serious detrimental effect on the overshoot, and settling time of the motor, but effectively reduced the steady state error to <1 pulse. However, since the controller was performing with a steady state error well within the acceptable range, it was decided that the detrimental effect of the integral component was not worth the positive effect, and it was re-adjusted to 0 integral gains.

With the controller tuned to a proportional gain of 0.005 and a derivative gain of 0.001, the motor was run at the demanded series of speeds, and the motor response was monitored. As the physical motor gives a pulse output, the voltage output across the analogue output terminals was measured. This will give an accurate indication of the most important monitoring characteristics of the controller, the settling time, and the steady state error.
Figure 6.19 Motor speed control test 1 shows rise time vs. settling time

Figure 6.20 Motor speed control test 1 shows overshoot (voltage) and steady state error (voltage)
The graphs in Figure 6.19 and Figure 6.20 indicate the motor response with relation to the difference in demanded speeds. For example if the speed was changing from 150Hz - 200Hz, this would be a speed difference of 50Hz. As can be seen from these graphs, the greater the speed difference, the higher the overshoot and the longer the settling time. The rise time and steady state error are mostly consistent across the board. An interesting thing to note, when decreasing the motor speed, the settling time and overshoot are considerably higher. This is because the motor does not actively decelerate, but instead relies on a loss of rotational inertia. In unloaded conditions this can take considerable time. With loading on the wheel, this response will be faster but it is unlikely that the motor speed will be symmetrical when rising and falling.

After a little bit more tuning, the tests were run again. This time with a proportional gain of 0.003 and a derivative gain of 0.003. The results of the second test are much better with a quicker settling time, and a lower overshoot. Steady state error is roughly the same, at around 0.1V.

Figure 6.21 Motor speed control test 2 shows rise time vs. settling time after tuning
Figure 6.22 Motor speed control test 2 shows overshoot vs. steady state error voltages after tuning.

Again, when decreasing the speed of the motor, the overshoot and settling time are considerably higher than when increasing the speed of the motor. The main reason for this asymmetric behaviour is that these tests were conducted without any load on the motor.

In a real world application, these motors will be subject to a load due to the inertia of the vehicle. This load can be calculated from known vehicle dynamics including but not limited to the weight of the vehicle, suspension, centre of gravity, friction coefficient with the road surface, and wind drag. As many of these dynamics vary from vehicle to vehicle, the decision was made to do a loaded test of the wheel performance using an arbitrary constant wheel load. This load was applied vertically to the wheel so that the load on the wheel was at right angles to the tangent of the wheel at the point of load contact [88].

The tests were run again with a load of 20kg applied to the wheel in the manner described above using the same proportional and derivative gains as the previous test.
Immediately, we can see that there is a significant performance improvement during deceleration of the wheel.
Rather than having a large overshoot that bottoms out at the specified minimum voltage, the wheel decelerates fast enough that the voltage overshoot is much less significant. Additionally, the settling time during deceleration is significantly improved, in some situations by up to 10 times. This is due to the fact that the wheel deceleration is much more rapid whilst the wheel is under load. This is expected, as the wheel load will effectively induce a force opposing the direction of motion of the wheel surface at the point of contact of the load. This in turn will result in a moment opposing the motor torque, which will aid in the slowing down of the motor. However, due to this opposing moment, the acceleration time of the wheel will be negatively impacted. This is apparent in the results, as we can see; the settling time when the speed is increasing is higher. Also important to note, is that the maximum speed of the wheel is also impacted when the wheel is under load when compared to the unloaded testing. This was apparent when trying to reach the maximum testing speed of 240Hz. In the loaded tests, setting the demand for 240 Hz ended up with the motor speed maxing out at approximately 230 Hz, falling short of its intended speed. As such, the overshoot in this case was 0, but the steady state error was 10. To overcome the problem of achieving acceleration or maximum wheel speed, the introduction of electric gearshift can be considered [89]. This could assist in achieving maximum speed by changing the operating voltage difference of the motor.

**Discussion**

When observing the result of the validation of the controller, we see that when decreasing the motor speed the controller performs much less effectively than when increasing the speed of the motor. This is because when the motor actively accelerates, it passively decelerates. This situation is very common when using PID controllers. One method of mitigating the impact of this phenomenon is to over-damp the controller. The idea behind over-damping is that the positive overshoot will be minimised, and therefore the settling time can be reduced. This method however does not help in situations when a rapid decrease in speed is required.
Whilst a motor under load will decelerate much faster than an unloaded motor such as the one used in these tests, it is highly unlikely that the motor will be symmetrical with respect to its acceleration and deceleration rates. The controller itself needs to account for this asymmetrical operation. Following are a few potential, but untested ideas for future development of this motor controller.

One method for accounting for the asymmetrical behaviour of the motor is to dynamically adjust the sampling time in the discrete system [90]. By increasing the sampling time, the rate of change of the output of the controller will decrease. If this sampling time can be dynamically adjusted to suit the rate of change of the feedback variable, then the controller response can be adjusted accordingly, which will result in a voltage output which much more closely matches the actual rate of change of the motor speed.

A further method for accounting for asymmetry is to add an active braking of the motor during deceleration [91]. This can be done either with a physical disc or drum brake, or electronically with a resistive load across the motor. If this method is used, not only will the controller perform better during acceleration, but the braking should be able to regenerate some of the momentum of the wheel as charge for the battery [92]. This could have a significant impact on the range of the vehicle, as less of the output power would be dissipated as heat.

### 6.5 Summary

A different approach to the experiment is proposed here to determine the capability and performance of wheel speed control as the key actuation process of vehicle stability control. To do so, the behaviour of the in-wheel motor is examined and vehicle physical properties are measured to facilitate the proposed bench test. The proposed embedded host is described here along with its required digital and analogue input-output modules. Also the possibilities and limitations of using the digital controller are included.
The approach towards PID wheel speed controller development is described along with the user-defined functions in the required software for the proposed digital controller. Designing a modular and flexible system to interface with different types of motors and multiple higher level controllers requires a portable, layered design. Satisfactory experimental results are discussed and a proposal for an improved actuation method is discussed.
Chapter 7
Advanced features of ESC
7 Advanced features of ESC

7.1 Centre of gravity and its effect on electronic stability control

A vehicle performance and energy enhancement system in an EV is much dependent on load characteristics. The distribution of load, placement of parts and components on a vehicle will not remain the same and will diametrically change, changing the centre of gravity. It is important to consider the shifting of centre of gravity (COG) position for an advanced electric vehicle, as it shifts dynamically depending on change of load distribution, running and road condition and requires a control logic analysis for the best controllability and performance of the motors.

The change of COG not only affects the motor control strategy but also affects the ESC (ECS) of the vehicle. This section addresses the effect of a dynamically changing COG on the real time processing of a vehicle’s performance and its effects on the ESC system based on the paper “Fuzzy logic controller in an Electric Vehicle with dynamically changing centre of gravity” [93].

The centre of gravity of vehicle is an important property when considering vehicle handling and accident prevention. It is the point where the vehicle's weight is balanced in all three directions: front to rear; side to side; and up and down [94]. It is worth mentioning that with an in-wheel motor electric vehicle or a hub wheel motor vehicle it is important to take into account the shifting or changing of the centre of gravity position.

Automotive manufacturers provide the measurements of vehicles including CoG after estimating it in different ways. In fact CoG varies depending on the load distribution on the vehicle. Shifting or changing of centre of gravity is not limited to static load and the number of passenger in the vehicle; it also varies depending on the orientation on the load and passenger [95]. An active accident prevention technology like ESC prevents the vehicle from spinning and drifting out and helps to prevent single vehicle accidents.
To do this a controller has to be developed considering the wheel speed, lateral acceleration, yaw rate and steering angle as input using different sensors. These parameters will be used for the calculation and the output will be the brake pressure or torque in different wheels. To achieve these outputs, the controller has to calculate desired yaw rate, desired slip-side angle and other important parameters which depend on the dynamics of the vehicle [96]. Required parameters from the car dynamics have a strong connection to CoG. A predetermined position of CoG may not be perfect for ESC where the output will be brake pressure or torque in different wheels calculated using CoG. An imperfect result may come from the controller which may create hazard instead of preventing an accident. This study proposes the importance of the shifting behaviour of CoG. A model based simulation of a vehicle is used to analyse the change to the CoG and its effect on ESC.

Stability control systems minimize and prevent vehicles from skidding, spinning and drifting out. This computerized technology detects the loss of steering control and applies brake pressures to wheels to maintain the intended direction according to the steering angle. The main objective of these three systems is to control yaw by creating difference between left and right wheel forces by using differential braking or differential torque.

In differential braking systems, clock-wise and counter clock-wise yaw moment is generated by increasing the brake pressure between left and right wheels. The block diagram given in Figure 7.1 shows the generalized structure of yaw stability control. From the structure of ESC, it is noticeable a targeted yaw rate and targeted slip angle are the most important parameter for the controller to determine wheel forces. These parameters are determined based on (5.15) - (5.18).

In determining the targeted yaw rate and slip angle this controller needs to know certain properties of the vehicle which are related to its dynamics like longitudinal distance from centre of gravity to front tyres $l_f$, and longitudinal distance from centre of gravity to rear tyres $l_r$ which involves of the position of the CoG.
ESC development is an evolution of the Antilock Braking System (ABS) concept with additional sensors such as steering wheel angle sensor, and a gyroscopic sensor. If the gyroscopic sensor detects that the direction taken by the car does not coincide with that of the steering wheel or there is an unpredicted inclination of the car, the sensor acts and ESC software will apply brake pressure to the necessary individual wheel(s) so that the vehicle continues in the intended direction [97]. This action requires optimum analysis for the optimum controllability and performance of the electric motors in each wheel.

Figure 7.1 Structure of Electronic Stability Control
7.1.1 Vehicle model and analysis

A model based simulation using MATLAB Simulink is presented here to verify the effect for various extra loads on the car and slope of road. The CoG of a vehicle on a horizontal axis can be located by calculating distance $l_f$ from the front axle to the centre of gravity and distance $l_r$ from rear axle to the centre of gravity shown in Figure 7.2.

![Figure 7.2 Vehicle model for finding centre of gravity](image)

Let,

- $l_f$ be the distance from front axle to the centre of gravity
- $l_r$ be the distance from the rear axle to the centre of gravity
- $Load_{rear}$ be the load on rear axle
- $Load_{front}$ be the load on front axle
- $Load_{total}$ be the total load of the car

Then,

$$l_f = \frac{Load_{rear} \times l_f + l_r}{Load_{total}}$$

(7.1)
Hence,

\[ l_f = (l_f + l_r) \left(1 + \frac{\text{Load}_{\text{front}}}{\text{Load}_{\text{rear}}}\right) \tag{7.2} \]

\[ l_r = (l_f + l_r) - l_f \tag{7.3} \]

Using equation (7.2) and (7.3) the CoG can be located on the bisect line of the car. Point of mass is another way of locating the centre of gravity which might be used for an asymmetric load [98]. Load can be calculated online using axial sensors in rear and front axles. Figure 7.3 shows the online load determining system.

![Figure 7.3 Weight distribution by axial sensors](image)

### 7.1.2 Simulation result of vehicle with centre of gravity shifting

Simulation has been done in two separate steps. In the first step the CoG is calculated using a MATLAB program. In the second step the obtained centre of gravity is used in Simulink block based model to observe the effect. Another model has been developed using electric motors as engines for each wheel as shown in Figure 7.4.
Different scenarios have taken into consideration the issues of incorporating the load in the axles and the slope of the road. In this simulation, parameters for vehicle dynamics are taken as shown in Table 7-1.

![Vehicle model with in wheel electric motor](image)

**Figure 7.4 Vehicle model with in wheel electric motor**

<table>
<thead>
<tr>
<th>Vehicle Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of the car</td>
<td>1500 kg</td>
</tr>
<tr>
<td>Centre of gravity height</td>
<td>0.5 meter</td>
</tr>
<tr>
<td>Frontal area</td>
<td>$3^2$ meter</td>
</tr>
<tr>
<td>Drag coefficient</td>
<td>0.4 for air drag</td>
</tr>
<tr>
<td>Effective rolling radius of tyre</td>
<td>0.4064 meter</td>
</tr>
<tr>
<td>Rated vertical load on the tyre</td>
<td>3000 N</td>
</tr>
<tr>
<td>Maximum power of the engine</td>
<td>150000 watt with speed of 4500 rpm</td>
</tr>
</tbody>
</table>

186
In this simulation, we assume normal conditions where there is no extra load on any of the axles and the car is moving on a plain road with zero degree inclined angle. Later on, for the extra load in rear, it is assumed that distance from the front axle to the centre of gravity $l_f$ and distance from the rear axle to the centre of gravity $l_r$ have moved for 10 centimetres towards the rear axle with a 5 degree inclined angle. Simulation shows noticeable output for vertical tire force on the rear and front axles and vehicle velocity. The result given in Figure 7.5 and Figure 7.6 shows the change of vehicle speed and significant difference in the vertical tyre force $F_z$ in the two different scenarios.

**Figure 7.5 Case-1:** Centre of gravity in normal position and the incline angle is 0 degree

**Figure 7.6 Case-2:** Centre of gravity shifted horizontally by 10 cm and the incline angle is 5 degree
The significant difference in the vertical tyre force $F_z$ has an influence on longitudinal and lateral tyre forces. These tyre forces also have an influence the torque of the wheels. A relation between torque and the vertical tyre force can be found from the wheel force calculation using equation (5.11), (5.12) and (5.14) in section 5.3.2 of the Dugoff tyre model. According to the simulation, if the centre of gravity changes towards rear, the vertical force on each tyre changes. Actual required torque can be calculated considering vertical tyre force while using dynamically calculated $l_f$ and $l_r$ from equation (7.2) and (7.3). Using the torque balance from equation (6.14) the required brake torque can be calculated to be applied via ESC. The ESC system activates when the difference between the desired path and actual path is too great. In ESC using differential braking system, brake pressure is increased to the left wheels compared to the right wheels to create a counter clockwise yaw moment. For a clockwise yaw moment brake pressure is increased at the right wheels compared with the left wheels.

These yaw moments help to keep the car on the desire path. ESC systems have to use vehicle data such as CoG precisely. How much intervention is necessary on the brake pressure in the action of stability control has to be determined based on the vehicle data. Instead of a pre-estimated CoG in most of the stability controller, a fuzzy logic based controller using real time data for CoG will be more effective in controlling the stability of a car.

### 7.1.3 Summary

The effect of shifting the CoG due to load on the vehicle has been shown using the simulation. The vertical force on the tyres changes as the centre of gravity shifts. Active accident prevention control such as electric stability control will determine the required brake pressure or driving torque needed on each wheel taking into consideration the shifting of centre of gravity to avoid possible errors. Four independent wheels with independent motors will give better controllability if independent brake pressure is applied after calculating the required force correctly. Better controllability will ensure better performance in stability. Implementation of regenerative braking in a vehicle will lower the energy consumption. The number of intelligent sensors in determining the
dynamics of vehicle enhanced the same controller for stability control. Further work can be done to calculate the vertical shifting of the centre of gravity for stability control.

7.2 Effect of CoG in vehicle mass distribution

In the previous section it is observed that the position of CoG is a key parameter for stability and safe handling. Distribution of components on an EV determines the position of CoG which is influenced by the architecture of the vehicle [99]. In four in-wheel motors EV, it is much easier to maintain the symmetric architecture to place the components. Problems regarding CoG may occur due to placement of battery packs for range coverage. The mass of battery packs and controllers changes the position of CoG in longitudinal, lateral and in vertical directions.

Other factors like vehicle track width, vehicle weight and the length of the wheelbase have strong influences on the CoG but necessary changes in these parameters are sometimes difficult to overrule. Focus can be given to EVs converted from ICEVs, which are currently favourable to solve the carbon emission problem immediately along with the new production of EVs. For these retrofitted EV, vehicle track width and length of wheelbase are difficult to alter.

An obvious option is an efficient distribution of mass for positioning of CoG. Research has been carried out to measure the effects when mass distribution is changed in paper [100] and this section discussed the effect of mass on CoG based on this paper. By changing the positions of vehicle components like battery packs and the motor controller, shifting of CoG is achieved and the effects on the vehicle have been analysed in this research by using computer simulation. Analysis is done here to observe the effect of shifted CoG on the vehicle, which will be helpful for controlling the stability of the vehicle.
7.2.1 Mass Distribution and the centre of gravity

Changing the location battery pack on the vehicle creates a major change on CoG as battery packs are generally 20 to 25% of the weight of small EVs. By placing them in different location on the vehicle a change in the position of CoG causes different longitudinal mass distribution.

- front to back mass distribution ratio 60:40 when the battery pack at the front
- front to back mass distribution ratio 50:50 when the battery pack at the middle
- front to back mass distribution ratio 40:60 when the battery pack at the rear

It is worth mentioning that lateral mass distribution is maintained with a ratio of 50:50 for left and right. An unequal mass distribution between left and right will create unequal normal force due to weight transfer and will cause the vehicle to skid while turning [101, 102].

7.2.2 Calculation of CoG in case of converted EVs:

In this calculation the front axle is assumed as the longitudinal reference and the ground is the vertical reference for CoG calculation. Distance from these references and the mass of each added or removed competent determines the CoG position of the vehicle.

Let,

Distance of CG from front axle be X
Distance of CG from ground be Z
Total vehicle weight be M
Mass of each component be \( m_n \)
Corresponding CG distance of component from front axle be \( x_n \).
Corresponding CG distance of component from ground be \( z_n \).
Then, longitudinal position $X$ of CoG is determined using the following equation:

$$X = \frac{\sum m_n x_n}{M} \quad (7.4)$$

Vertical position $Z$ of the CoG is determined using the following equation:

$$Z = \frac{\sum m_n z_n}{M} \quad (7.5)$$

7.2.3 Simulation

A model based simulation is done using MATLAB/SIMULINK and the effect of CoG shifting is observed while turning in a circular path. In the simulation, forces are applied on the wheel of the model vehicle in both longitudinal ($X$) and lateral ($Y$) directions to analyse the effect of different mass distribution on vehicle stability while turning [103]. In the beginning, the longitudinal and vertical positions of CoG are calculated by using MATLAB then these values are used in the vehicle body model for three different scenarios. A multi-body based vehicle model is used here which is consists of SIMULINK body blocks and body sensor blocks [104]. Unsprung and sprung mass are connected using a body block with the wheels and a body sensor is attached. These virtual sensors are capable of providing the vehicle speed and yaw rate of the vehicle in the simulation. This yaw rate helps to determine the radius of the curved path followed by the vehicle by using the following equation:

$$R = \frac{V_x}{\Psi} \quad (7.6)$$

Here,

Radius of the path is $R$

Vehicle longitudinal; velocity is $V_x$

Yaw rate is $\Psi$
7.2.4 Results and Analysis

In three different cases of simulation, only different longitudinal mass distribution has been considered and the followed path is plotted in an X-Y graph to demonstrate the effects of CoG positions on vehicle.

In Figure 7.7 the path of the vehicle is shown for the first case where 60% of total weight of the vehicle is at the front and 40% is at the rear. As the rear tires face a greater slip angle than the front tires, the vehicle over steers while turning.

![Figure 7.7 Case 1 of ratio 60:40 with over steering behaviour](image)
The outcome of the second case is given in the Figure 7.8, where the front to rear weight ratio is 50:50. The vehicle turned with neutral steering and slight over steering because of the weight transfer effect on front tires.

The third case is shown in Figure 7.9 where the vehicle has a front to rear weight ratio of 40:60. The vehicle is under steered as the rear tires face a greater portion of the vehicle weight. This creates greater slip in the rear tires. In this situation the vehicle follows a curved path of a larger radius than the projected trajectory of the vehicle.
Figure 7.9 Case 3 of ratio 40:60 with under steering behaviour

7.2.5 Summary:

Different mass distribution of EV has been considered to analyse the effect of CoG shifting on the behaviour of the vehicle in dynamic condition. Simulation shows the different behaviours of the vehicle due to these changes in CoG which helps to determine proper vehicle mass distribution for better handling.
Chapter 8
Conclusion
8 Conclusions and future work

8.1 Objectives

Electric vehicles are a possible alternative to reduce the pollution of the environment. Advances have been made in electric vehicle technology which provides flexibility in designing drive trains. Advanced EVs which have four in-wheel motors require individual control for an effective drive train. The increased simplicity and efficiency of using four in-wheel motors creates some challenges too. Being a cutting edge technology the four in-wheel motors EV are yet to gain approval from OEM. Most OEMs' prefers a centralised motor over in-wheel motors. This poses a major challenge in getting a vehicle stability controller implemented by controlling individual wheels. Braking based vehicle stability does not provide the expected longitudinal response and slows down the vehicle. Loss of energy will occur if only braking based stability control is used in four in-wheel motor EV. Steer-by-wire is good for intended path detection but stability control based on this method is far more complex. Considering these shortcomings, differential driving torque based vehicle stability is considered here. But it is not possible to use a conventional differential driving torque based vehicle stability control in an advanced four in-wheel EV which is completely free of any kind of gear ratio or differential in its drive train.

Hence an advanced controller is proposed that is capable of controlling the in-wheel motors individually, and performs vehicle stability so that accidents can be avoided. A simplified version of ESC is proposed that uses differential torque to stabilize the vehicle in sudden manoeuvres by controlling the wheel speed. A generalized method of differential driving torque calculations for in-wheel electric motors is proposed here.

8.2 Achievements

Several outcomes have been included in the previous relevant sections. From the initial discussions in Chapter 2 it is proposed that electric vehicles are comparatively more environmental friendly for their efficiency and can be a viable form of transport.
Advances in architecture are the main focus of current technology where a drive train consisting of four in-wheel motors is used. So the necessity of a vehicle stability control is demonstrated for this new type of vehicle. A comparison of different types of ESC focusing four in-wheels EV is shown here considering sensors, actuators, complexity and challenges in Chapter 3. From the discussion on existing different types of stability control techniques, a differential driving torque based ESC is selected which would be suitable a four in-wheel vehicle to counteract the need to slow down while the stability control is active.

Deeper analysis is done on the differential driving torque based stability controller to find a simple and viable solution. In this section updated research works related to differential torque based vehicle stability are discussed to propose a simplified version. From the discussion on available simulation software used for vehicle stability test it is seen that a detail definition in vehicle model is required to simulate the stability test in case of a four in-wheel motor EV. A decision has been made to use only yaw rate and slip angle error in the control law from the detailed discussion on control parameters.

This discussion provided the guidelines to focus specifically on the least covered area in differential torque based stability control. In order to do this a new overall system architecture is designed for a vehicle stability controller which is described at the end of Chapter 4.

To verify the proposed method and identify the possibility of implementation, steps have been taken to simulate the system and the experiment. In order to perform the vehicle stability test using a control method, a model of four in-wheel vehicle is created with rear and front wheels in MATLAB/SIMULINK, this is described in Chapter 5.

From this simulation, vehicle body dynamics and wheel dynamics are observed and the control law is implemented to determine the corrective longitudinal wheel force. Required differential torques for the wheels are determined from the control law to compensate for creating difference in longitudinal wheel force.
A very simplified determination of differential driving torque for the wheels to compensate against corrective yaw moment is achieved. Simulation is done here to observe the controllability of the proposed vehicle stability control. A satisfactory result of yaw control is achieved from the simulation in the case of the yaw stability of the vehicle. A slight improvement is also observed in the vehicle body slip angle, though it is supposed to control both these yaw rate and slip angle simultaneously. From this simulation the required wheel angular velocity is observed and achieved by using the in-wheel motor.

Due to the complexity of a field test using the developed platform, wheels speed control is measured in a bench test. A simple wheel speed controller is designed and implemented to control the wheel speed. The advantage of individual wheel rotational control is taken as the main actuation method to create the differential torque in the different wheels for stability. In order to verify the proposed method wheel speed, a control experiment was done as explained in Chapter 6. Initially the behaviour of the in-wheel motor is examined and vehicle physical properties are measured to use this information in the proposed bench test. A National Instrument’s Compact Real-time Input Output (NI cRIO) device was used in order to develop a portable, real-time, accurate controller for the 4 in-wheel motor electric vehicle. NI LabVIEW 2011 is used to program a PID based digital feed-back controller as the interface for the upper level stability controller and the actuating wheels. LabVIEW allows for simple and effective programming of a multithreaded FPGA deployable program in multiple layers. A test rig is developed to perform a test on the wheel so that observations can be done on the wheel speed control which depends on the upper level of the controller.

From the conducted bench test it is seen that the wheel decelerates fast enough so that the voltage overshoot is much less significant. The acceleration time of the wheel decreased as the settling time also decreased. This may not be a precise control of wheel speed but these outcomes indicate the possibility of creating rotational difference which also is capable of compensating for correcting the yaw. This method is not implemented here.
Several other outcomes have been reached from different analyses which are included in this research work as advanced features. One of them is the effect of the shifting centre of gravity due to load on the vehicle. This has been shown using a simulation. To rectify this active accident prevention control like the electronic stability control are proposed to avoid possible errors and determine the required brake pressure or driving torque needed on each wheel.

Another analysis for finding the effects of mass or load distribution on CoG is carried out by simulating the different examples of load on the vehicle. It is realized from the previous outcome that CoG is one of the significant parameters which is affected by vehicle mass and influences vehicle handling.

In the case of four in wheel vehicles a change in CoG location depends on the different architectural arrangements of drive train and sub-systems like the battery pack which affect the mass distribution of the vehicle. We analyse the effect of changing CoG on the vehicle path while turning. This will be useful to develop a more robust control strategy for vehicle stability. Different architectural layouts with 60%, 50% and then 40% of total weight at the front are considered for the simulations and the over steer or under steer situations are observed while turning. This experiment helps to correct vehicle mass distribution for better handling.

### 8.3 Contributions

This research work is focused on the emerging technology of EV with an advanced drivetrain. A recent report included in Chapter 1 on the increasing number of vehicles indicated the rise of passenger vehicles in the recent years. This research work focuses on developing a passenger EV because it is the largest group of vehicle in the world. Hence replacing the conventional passenger vehicles with EVs will contribute more to control pollution.
In this research work, an advanced drivetrain with four in-wheel vehicle is proposed after analysing its advantages. Later on a vehicle platform with four in-wheel motor is developed to use in the research. This platform is intended for use in further research like autonomous control by incorporating steer-by-wire. This platform will help to explore more in four in-wheel control techniques.

This four in-wheel vehicle requires a stability controller for normal operation and most importantly for sudden manoeuvres. Because of having a different in drivetrain, this vehicle demands of a novel method of stability control. So, analysis is done on different torque based vehicle stability controls. We concluded the findings in chapter 4. This analysis can be used to see the research has been done in the area of torque based stability control.

A differential driving torque based stability control is proposed with the overall system configuration suitable for a vehicle which uses four in-wheel electric motors. A differential driving torque based stability control is proposed with the simple strategy of using yaw rate error and side slip error in a single control loop which is used in other research for a differential braking method. This works as an upper level controller. A very simplified determination of differential driving torque for the wheels to compensate against corrective yaw moment is proposed in the lower level control.

In order to verify the proposed stability method, simulation is done as the first step. As this is a four in-wheel motor vehicle it requires an analysis of vehicle body dynamics and individual wheel forces in lateral and longitudinal axes to determine the status of the vehicle.

To meet the requirements a reference vehicle with front and rear wheels is modelled using the most widely used graphical programming language MATLAB/SIMULINK. This model is created with a modular approach so that more subsystems can be added or removed if required in future research.
To simulate a stability control test, an overall environment is created including subsystems to determine the nominal dynamic behaviour of the vehicle body, a subsystem of control law to determine the corrective longitudinal wheel force and manipulation of the torque for each wheel. This entire simulation environment is capable of simulating behaviours of the body, wheel and status of the vehicle and can visualize variety of data. This includes vehicle longitudinal velocity, lateral velocity, vehicle body slip angle, front and rear wheel slip ratios, wheel slip angles, longitudinal and lateral wheel forces, vehicle yaw rate, nominal yaw rate and so on. The simulation is also capable of providing differential wheel torque and wheel angular velocity. This vast and detailed simulation environment will play a vital role in the current and future research in the area of vehicle dynamics. The model can be used for different control laws or for different type of actuation methods in a stability control development.

Testing of the proposed non-conventional yaw stability control, based on differential torque, is a crucial challenge as it required satisfying regulations of the test, more resources to achieve signal or data, and most importantly more time.

Due to the inevitable problems related to the field test for vehicle stability, a different testing method is proposed here. As the testing is mostly related to the actuation on the wheel based on the demand generated by the stability controller, a wheel control test rig with the facility to assign wheel load is developed in order to perform the experiment. A digital controller, which is common for field test or bench test is developed using a high performance FPGA based controller by National Instruments. The wheel in this test rig is then connected to the FPGA through an I/O module. The software program is written to control the wheel in the graphical LabVIEW program. The FPGA is then programmed and controlled from computer. The hosted software program in the FPGA is used to measure the demand of wheel control from the upper level of stability control and control the wheel according to this demand.

The program is highly modular and the wheel feed-back control interface for four individual wheels is reusable for other control methods of the drivetrain, for example, controlling the wheels’ speed while cornering.
The proposed embedded program for vehicle control is suitable for further research into vehicle dynamics by adding more program modules and more sensors.

8.4 Suggestions for further research

The control strategy of using differential torque and wheel speed controls developed in this work has worked well for the vehicle stability and longitudinal dynamics. Further effort has been invested into discovering an optimal method of generating the required corrective yaw moment using both differential driving torque and braking torque. The method is described in the simulation but the electromagnetic braking actuation for the drum brake of this in-wheel motor has not been implemented.

In case of the wheel control experiment it is found that the motor actively accelerates and it passively decelerates. A symmetrical behaviour with respect to its acceleration and deceleration rates is required for highly accurate wheel angular velocity. To eliminate this asymmetry, active braking of the motor can be added during deceleration [105]. An electromagnetic actuator to control the physical disc or drum brake is proposed earlier for improved performance.

An electronically resistive load can be added across the motor for active braking in order to improve performance in wheel speed control during deceleration [106].

In future the most important future work will be to examine the effectiveness of the stability controller by doing a field test using the developed four in-wheel platform. To test the prototype requires the inclusion of some more sensors, data acquisition and a computer within the vehicle to observe the status of the vehicle and actuation signals of the voltage.

It is also important to identify the tire road friction coefficient by using different estimation methods. In this regard, an advanced sensor can be included to identify the
runtime friction coefficient between the tire and the road. This could provide more flexibility in the control system [107].

The effect of ESC on multiple vehicle crashes requires more analysis as research in this area is shows both the positive and negative effects. According to some surveys, ESC is responsible for increasing the risk of crash. To overcome this problem a new paradigm of ESC could be suggested which would include obstacle detection sensors [108] and assigning limitation on lateral displacement [109] during the operation of ESC.
Reference


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