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UMI
MODELING AND CONTROL OF POWERTRAINS WITH STEPPED AUTOMATIC TRANSMISSIONS

DISSERTATION

Presented in Partial Fulfillment of the Requirements for
the Degree Doctor of Philosophy in the Graduate
School of The Ohio State University

By
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The Ohio State University
1999

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ABSTRACT

This dissertation addresses the modeling and control of automotive powertrains with emphasis on stepped automatic transmissions. An extensive survey of the state-of-the-art of modeling and control issues with emphasis on automatic transmission is provided, the survey indicating the need for improved modeling and control of powertrain systems, especially the need for integrated powertrain control.

Development of model-based approaches to integrated engine - transmission control design offers the potential for addressing more demanding control applications and significantly greater gains. Simulation models of the automatic transmission shift process with varying levels of complexity, suitable for dynamic response simulation, controller design and validation, are implemented using Matlab/Simulink. Linearized models of transmission dynamic response during the inertia phase of a gear shift are obtained directly from the Simulink block diagram.

Closed-loop shift control algorithms for the inertia phase of a gear shift are developed based on these linearized models and implemented in the powertrain simulation. An open-loop control strategy for the torque phase of a gear shift is developed and shown to be effective in reducing the torque drop in the torque phase. The results indicate the ability of integrated engine and transmission control to achieve satisfactory shift quality.

Powertrain neutral idle control improves fuel economy in urban driving and minimizes the engine vibrations transmitted to the driveline. Neutral-idle control requires fast and smooth application of the forward clutch in coordination with engine control functions. Understanding of pressure control system dynamics is essential to devise a control strategy for the neutral-idle process, as well as for other shift control applications.
such as clutch-to-clutch control. Detailed analysis of the dynamic response of a clutch pressure control system resulting in a nonlinear simulation model is presented and validated by experiments. Closed loop control of clutch pressure using feedforward plus feedback control is implemented in a test setup. Experimental results show that clutch pressure can be accurately controlled.

The formulation of the powertrain neutral-idle control problem is presented. A multivariable control strategy involving coordinated engine and transmission control is developed based on quantitative models of powertrain dynamic responses. Simulation results show that satisfactory neutral-to-first gear shift quality is achieved.
To my heavenly Father
and
My parents
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FIELDS OF STUDY

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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Abstract</td>
<td>ii</td>
</tr>
<tr>
<td>Dedication</td>
<td>iv</td>
</tr>
<tr>
<td>Acknowledgments</td>
<td>v</td>
</tr>
<tr>
<td>Vita</td>
<td>vii</td>
</tr>
<tr>
<td>Table of Contents</td>
<td>viii</td>
</tr>
<tr>
<td>List of Tables</td>
<td>xii</td>
</tr>
<tr>
<td>List of Figures</td>
<td>xiii</td>
</tr>
<tr>
<td>Nomenclature</td>
<td>xxii</td>
</tr>
<tr>
<td>Chapters:</td>
<td></td>
</tr>
<tr>
<td>1. Introduction</td>
<td>1</td>
</tr>
<tr>
<td>1.1 Motivation for proposed research</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Objectives of the research</td>
<td>4</td>
</tr>
<tr>
<td>1.3 Organization of the dissertation</td>
<td>5</td>
</tr>
<tr>
<td>2. Literature review</td>
<td>7</td>
</tr>
<tr>
<td>2.1 Introduction</td>
<td>7</td>
</tr>
<tr>
<td>2.2 A representative automotive powertrain</td>
<td>8</td>
</tr>
<tr>
<td>2.3 Modeling of automotive powertrain systems</td>
<td>12</td>
</tr>
<tr>
<td>2.3.1 Engine models for transmission related research</td>
<td>13</td>
</tr>
<tr>
<td>2.3.2 Modeling of torque converters</td>
<td>15</td>
</tr>
<tr>
<td>2.3.3 Modeling of transmission hydraulic systems</td>
<td>21</td>
</tr>
<tr>
<td>2.3.4 Clutch/band friction characteristics for different types of clutches</td>
<td>29</td>
</tr>
<tr>
<td>2.3.5 Modeling of transmission mechanical systems</td>
<td>34</td>
</tr>
<tr>
<td>2.3.6 Modeling of vehicle dynamics</td>
<td>35</td>
</tr>
<tr>
<td>2.3.7 Integrated powertrain modeling</td>
<td>41</td>
</tr>
</tbody>
</table>

2.4 Review of control issues for automatic transmissions ...........................................43

2.4.1 Shift control........................................................................................................44

2.4.1.1 Pressure manipulation .................................................................................45
2.4.1.2 Inertia phase control ..................................................................................49
2.4.1.3 Clutch-to-clutch shift control .....................................................................55
2.4.1.4 Shift schedule control ................................................................................76

2.4.2 Torque converter clutch control ........................................................................78
2.4.3 Estimation techniques .......................................................................................88
2.4.4 Integrated powertrain control during shifts ......................................................95
2.4.5 Integrated powertrain control – neutral idle control .....................................106

2.5 Summary .............................................................................................................117

3. Powertrain model with emphasis on the automatic transmission ...............................119

3.1 Overview of the powertrain model.......................................................................119
3.2 Mean-value engine model ..................................................................................121
3.3 Torque converter model .....................................................................................125
3.4 Transmission mechanical system model ............................................................127

3.4.1 Planetary gear train model .............................................................................127
3.4.2 Models of transmission shift dynamics ..........................................................131

3.4.2.1 Dynamic model of the transmission in the first gear..............................131
3.4.2.2 Dynamic model of first-to-second gear upshift......................................134
3.4.2.3 Dynamic model of the transmission in the second gear .......................137
3.4.2.4 Dynamic model of second – third gear upshift .....................................137
3.4.2.5 Dynamic model of the transmission in the third gear .........................142
3.4.2.6 Dynamic model for third-to-fourth gear upshift .....................................143
3.4.2.7 Dynamic model of the transmission in the fourth gear .........................146
3.4.2.8 Transmission mechanical system model overview ...............................147
3.5 Transmission shift hydraulic system model .......................................................149
3.5.1 Clutch and band pressure profiles ................................................................149
3.5.2 Clutch/accumulator assembly model ............................................................150
3.6 Drivetrain and longitudinal vehicle dynamics model .........................................155
3.6.1 Gross vehicle dynamics model ....................................................................155
3.6.2 Vehicle dynamics model with tire-road interaction ......................................157
3.7 Model integration ................................................................................................159

3.8 Simulation results for power-on upshift .............................................................162
3.8.1 First-to-fourth gear power-on upshifts .........................................................162
3.8.2 First-to-second gear upshift including hydraulic system simulation ..........169
3.8.3 First-to-second gear power-on upshifts with compliant shaft model ..........183
3.9 Conclusion ..........................................................................................................189

4. Control strategies for shift control in an automatic transmission .............................191

4.1 Introduction ..........................................................................................................191
4.2 Determination of linearized models for the inertia phase of the 1-2 upshift .......193
4.3 Controller design for the inertia phase of the 1-2 power-on upshift .................199
6.4 Conclusion ..............................................................................................................382

7. Conclusions and recommendations .......................................................................386

7.1 Summary ...................................................................................................................386
7.2 Contributions of the research ................................................................................387
7.3 Recommendations for future research .................................................................389

Bibliography .............................................................................................................................391

Appendices ...............................................................................................................................402

Appendix A  Transmission dynamics simulator ...............................................................402
Appendix B  Neutral idle pressure control valve model..................................................409
Appendix C  Simulink block diagrams for neutral idle control system .........................410
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1 Operation conditions of the A350E transmission (Hojo, et al., 1992)</td>
<td>73</td>
</tr>
<tr>
<td>3.1 Clutch engagement schedule for automatic transmission</td>
<td>128</td>
</tr>
<tr>
<td>3.2 Alternative representations of steady state speed and torque relationships for the planetary gear train</td>
<td>132</td>
</tr>
<tr>
<td>3.3 Inputs and outputs of subsystems</td>
<td>160</td>
</tr>
<tr>
<td>3.4 Shifting Status Signal</td>
<td>161</td>
</tr>
<tr>
<td>4.1 Eigenvalues of Linearized Models for Different Instants in the Inertia Phase (Figure 4.1)</td>
<td>198</td>
</tr>
<tr>
<td>4.2 Eigenvalues of Linearized Models for Different $P_{a2}$ values (50% into inertia phase, Figure 4.2)</td>
<td>198</td>
</tr>
<tr>
<td>4.3 Eigenvalues of linearized models for different friction characteristics</td>
<td>201</td>
</tr>
<tr>
<td>4.4 Poles and zeros of the linearized $n$th order system model</td>
<td>202</td>
</tr>
<tr>
<td>5.1 List of symbols used in the experiment and simulation</td>
<td>248</td>
</tr>
<tr>
<td>5.2 Model parameter at the operating conditions chosen</td>
<td>272</td>
</tr>
<tr>
<td>5.3 Poles and zeros of transfer functions $\frac{P_{a2}}{P_{p}}(s), \frac{P_{b}}{P_{p}}(s)$</td>
<td>273</td>
</tr>
<tr>
<td>5.4 Pressure trace</td>
<td>284</td>
</tr>
<tr>
<td>6.1 List of operating points chosen for deriving the linearized models</td>
<td>331</td>
</tr>
<tr>
<td>6.2 Poles and zeros of system $(F_1, G_1, H_1, J_1)$</td>
<td>332</td>
</tr>
<tr>
<td>6.3 Poles and zeros of system $(F_2, G_2, H_2, J_2)$</td>
<td>333</td>
</tr>
<tr>
<td>6.4 Poles and zeros of system $(F_3, G_3, H_3, J_3)$</td>
<td>333</td>
</tr>
<tr>
<td>6.5 Poles and zeros of system $(F_4, G_4, H_4, J_4)$</td>
<td>334</td>
</tr>
<tr>
<td>6.6 Poles and zeros of system $(F_5, G_5, H_5, J_5)$</td>
<td>334</td>
</tr>
<tr>
<td>6.7 Poles and zeros of system $(F_6, G_6, H_6, J_6)$</td>
<td>335</td>
</tr>
<tr>
<td>6.8 Poles and zeros of system $(F_7, G_7, H_7, J_7)$</td>
<td>335</td>
</tr>
<tr>
<td>6.9 Poles and zeros of reduced order system $(F_{l1}, G_{l1}, H_{l1}, J_{l1})$</td>
<td>338</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>2</td>
</tr>
<tr>
<td>2.1</td>
<td>9</td>
</tr>
<tr>
<td>2.2</td>
<td>9</td>
</tr>
<tr>
<td>2.3</td>
<td>10</td>
</tr>
<tr>
<td>2.4</td>
<td>11</td>
</tr>
<tr>
<td>2.5</td>
<td>11</td>
</tr>
<tr>
<td>2.6</td>
<td>12</td>
</tr>
<tr>
<td>2.7</td>
<td>14</td>
</tr>
<tr>
<td>2.8</td>
<td>19</td>
</tr>
<tr>
<td>2.9</td>
<td>19</td>
</tr>
<tr>
<td>2.10</td>
<td>20</td>
</tr>
<tr>
<td>2.11</td>
<td>22</td>
</tr>
<tr>
<td>2.12</td>
<td>23</td>
</tr>
<tr>
<td>2.13</td>
<td>24</td>
</tr>
<tr>
<td>2.14</td>
<td>25</td>
</tr>
<tr>
<td>2.15</td>
<td>27</td>
</tr>
<tr>
<td>2.16</td>
<td>27</td>
</tr>
<tr>
<td>2.17</td>
<td>28</td>
</tr>
<tr>
<td>2.18</td>
<td>28</td>
</tr>
<tr>
<td>2.19</td>
<td>29</td>
</tr>
<tr>
<td>2.20</td>
<td>29</td>
</tr>
<tr>
<td>2.21</td>
<td>31</td>
</tr>
<tr>
<td>2.22</td>
<td>31</td>
</tr>
<tr>
<td>2.23</td>
<td>32</td>
</tr>
<tr>
<td>2.24</td>
<td>32</td>
</tr>
<tr>
<td>2.25</td>
<td>33</td>
</tr>
<tr>
<td>2.26</td>
<td>33</td>
</tr>
<tr>
<td>Section</td>
<td>Description</td>
</tr>
<tr>
<td>---------</td>
<td>-------------</td>
</tr>
<tr>
<td>2.27</td>
<td>Subcommittee round-robin tests results on friction characteristics (Fanella, 1994c)</td>
</tr>
<tr>
<td>2.28</td>
<td>An example of lever analogy</td>
</tr>
<tr>
<td>2.29</td>
<td>Bond graph representation of the planetary gear set shown in Figure 2.28</td>
</tr>
<tr>
<td>2.30</td>
<td>(a) Behavior of a tire under a driving torque (b) Behavior of a tire under a braking torque (Wong, 1978)</td>
</tr>
<tr>
<td>2.31</td>
<td>Variation of tractive effort coefficient with longitudinal slip of a tire</td>
</tr>
<tr>
<td>2.32</td>
<td>Variation of braking effort coefficient with skid of a tire on various surfaces</td>
</tr>
<tr>
<td>2.33</td>
<td>Automatic transmission dynamic model (Shindo, et al., 1979)</td>
</tr>
<tr>
<td>2.34</td>
<td>Comparison of shift characteristics during power-on downshift</td>
</tr>
<tr>
<td>2.35</td>
<td>Line pressure control during 1st to 2nd shift (Shinohara et al., 1989)</td>
</tr>
<tr>
<td>2.36</td>
<td>Line pressure control system (Shinohara et al., 1989)</td>
</tr>
<tr>
<td>2.37</td>
<td>Line pressure for each shift (Shinohara et al., 1989)</td>
</tr>
<tr>
<td>2.38</td>
<td>Acceleration map during power-on upshift (Shinohara et al., 1989)</td>
</tr>
<tr>
<td>2.39</td>
<td>Clutch hydraulic pressure control system (Kondo et al., 1990)</td>
</tr>
<tr>
<td>2.40</td>
<td>2nd - 3rd Upshift with clutch hydraulic pressure control (Kondo et al., 1990)</td>
</tr>
<tr>
<td>2.41</td>
<td>Shift controller block diagram (Hrovat and Powers, 1988)</td>
</tr>
<tr>
<td>2.42</td>
<td>Experimental and simulation results for 1st -2nd power-on upshift</td>
</tr>
<tr>
<td>2.43</td>
<td>Chrysler ultradrive clutch and gear arrangement (Leising et al., 1989)</td>
</tr>
<tr>
<td>2.44</td>
<td>Computer simulated adaptive control (Leising et al., 1990b)</td>
</tr>
<tr>
<td>2.45</td>
<td>Power-on upshift base control strategy (Hebbale and Kao, 1995)</td>
</tr>
<tr>
<td>2.46</td>
<td>Power-on downshift base control strategy (Hebbale and Kao, 1995)</td>
</tr>
<tr>
<td>2.47</td>
<td>Vehicle test results showing power-on upshift adaptation (Hebbale and Kao, 1995)</td>
</tr>
<tr>
<td>2.48</td>
<td>Vehicle test results showing power-on downshift adaptation</td>
</tr>
<tr>
<td>2.49</td>
<td>Longitudinal acceleration: open-loop vs. transmission-only closed-loop (Cho, 1987)</td>
</tr>
<tr>
<td>2.50</td>
<td>Longitudinal jerk: open-loop vs. transmission-only closed-loop</td>
</tr>
<tr>
<td>2.51</td>
<td>Required clutch torques: transmission-only closed-loop (Cho, 1987)</td>
</tr>
<tr>
<td>2.52</td>
<td>Required clutch pressures: transmission-only closed-loop (Cho, 1987)</td>
</tr>
<tr>
<td>2.53</td>
<td>Longitudinal acceleration: open-loop vs. integrated engine</td>
</tr>
<tr>
<td>2.54</td>
<td>Longitudinal jerk: open-loop vs. integrated engine and transmission closed-loop (Cho, 1987)</td>
</tr>
<tr>
<td>2.55</td>
<td>Clutch pressures: integrated engine and transmission closed-loop</td>
</tr>
<tr>
<td>2.56</td>
<td>Spark command from mbt (negative=retard): integrated engine and transmission closed-loop (Cho, 1987)</td>
</tr>
<tr>
<td>2.57</td>
<td>Percent reduction in engine torque required for integrated engine and transmission closed-loop control (Cho, 1987)</td>
</tr>
<tr>
<td>2.58</td>
<td>(a) Vehicle acceleration, 25 Shifts, Direct inertia adaptation, 25% $\mu_2$ Error (b) Normalized $\mu_2$ behavior, 25% error (Glitzenstein and Hedrick, 1990)</td>
</tr>
<tr>
<td>2.59</td>
<td>(a) Vehicle acceleration, 8 Shifts Composite inertia adaptation, 25% $\mu_2$ Error</td>
</tr>
<tr>
<td>2.60</td>
<td>(a) Vehicle acceleration, 5 Shifts Composite torque and inertia phase adaptation, 25% $\mu_2$ error (b) Normalized $\mu_2$ behavior, 25% error (Glitzenstein and Hedrick, 1990)</td>
</tr>
<tr>
<td>2.61</td>
<td>Clutch-to-clutch load transfer (Brown and Hrovat, 1988)</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
</tr>
<tr>
<td>---------</td>
<td>-----------------------------------------------------------------------</td>
</tr>
<tr>
<td>2.62</td>
<td>Schematic drawing of the A350E transmission (Hojo, et al., 1992)</td>
</tr>
<tr>
<td>2.63</td>
<td>Transient characteristics, shifting from 2&lt;sup&gt;nd&lt;/sup&gt; to 3&lt;sup&gt;rd&lt;/sup&gt; gear (Hojo et al., 1992)</td>
</tr>
<tr>
<td>2.64</td>
<td>(a) Shift characteristics: the completion of shift by the front gear unit lagged</td>
</tr>
<tr>
<td>2.65</td>
<td>Clutch-to-clutch cooperative control system (Hojo et al., 1992)</td>
</tr>
<tr>
<td>2.66</td>
<td>Effectiveness of compensation for delays (Hojo et al., 1992)</td>
</tr>
<tr>
<td>2.67</td>
<td>Damper clutch control system (Hiramatsu et al., 1985)</td>
</tr>
<tr>
<td>2.68</td>
<td>$\mu$-$\omega_{slip}$ Characteristic of clutch facing material (Hiramatsu et al., 1985)</td>
</tr>
<tr>
<td>2.69</td>
<td>Hunting of slip control system (Hiramatsu et al., 1985)</td>
</tr>
<tr>
<td>2.70</td>
<td>Block diagram of damper clutch control system (Hiramatsu et al., 1985)</td>
</tr>
<tr>
<td>2.71</td>
<td>Feedback control system (Kono et al., 1995)</td>
</tr>
<tr>
<td>2.72</td>
<td>Comparison of PID and $H_\infty$ controllers (Kono et al., 1995)</td>
</tr>
<tr>
<td>2.73</td>
<td>Noise and vibration evaluation results (Kono et al., 1995)</td>
</tr>
<tr>
<td>2.74</td>
<td>Torque converter bypass clutch slip control system: overview</td>
</tr>
<tr>
<td>2.75</td>
<td>Torque converter bypass clutch slip control system: subsystem 1</td>
</tr>
<tr>
<td>2.76</td>
<td>Torque converter bypass clutch slip control system: subsystem 2</td>
</tr>
<tr>
<td>2.77</td>
<td>Torque converter bypass clutch slip control system: subsystem 3</td>
</tr>
<tr>
<td>2.78</td>
<td>Hydraulic torque converter characteristics (Minowa et al., 1994)</td>
</tr>
<tr>
<td>2.79</td>
<td>Simulated results for estimated turbine torque (Minowa et al., 1994)</td>
</tr>
<tr>
<td>2.80</td>
<td>Simulated results for estimated driven shaft (transmission output shaft) torque (Minowa et al., 1994)</td>
</tr>
<tr>
<td>2.81</td>
<td>Two methods of torque estimation (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.82</td>
<td>Accuracy of estimated turbine torque (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.83</td>
<td>Measured and estimated turbine torque (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.84</td>
<td>(a) Experimental results showing speed and acceleration estimation during an upshift. (b) Performance of the Kalman filter showing improved acceleration estimation (Hebbale and Ghoneim, 1991)</td>
</tr>
<tr>
<td>2.85</td>
<td>Integrated control procedure (Shinohara et al., 1989)</td>
</tr>
<tr>
<td>2.86</td>
<td>1&lt;sup&gt;st&lt;/sup&gt; - 2&lt;sup&gt;nd&lt;/sup&gt; Upshift with engine torque control (Kondo et al., 1990)</td>
</tr>
<tr>
<td>2.87</td>
<td>Detailed control block diagram of throttle valve control with estimated (transmission output shaft) torque feedback (Minowa et al., 1994)</td>
</tr>
<tr>
<td>2.88</td>
<td>Transmission output shaft torque characteristics: (a) using conventional method, (b) using throttle control only, (c) using throttle and line pressure control</td>
</tr>
<tr>
<td>2.89</td>
<td>Comparison of longitudinal acceleration using proposed and conventional methods (Minowa et al., 1994)</td>
</tr>
<tr>
<td>2.90</td>
<td>Block diagram for upshift control with torque estimation</td>
</tr>
<tr>
<td>2.91</td>
<td>Algorithm for upshift control (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.92</td>
<td>Algorithm for downshift control (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.93</td>
<td>Experimental results for 1&lt;sup&gt;st&lt;/sup&gt; - 2&lt;sup&gt;nd&lt;/sup&gt; full throttle upshift</td>
</tr>
<tr>
<td>2.94</td>
<td>Experimental results for 4&lt;sup&gt;th&lt;/sup&gt; - 1&lt;sup&gt;st&lt;/sup&gt; downshift (Ibamoto et al., 1995)</td>
</tr>
<tr>
<td>2.95</td>
<td>Time chart of proposed control algorithm (Minowa et al., 1996)</td>
</tr>
<tr>
<td>2.96</td>
<td>Comparison of longitudinal acceleration during 3&lt;sup&gt;rd&lt;/sup&gt; - 1&lt;sup&gt;st&lt;/sup&gt; and 4&lt;sup&gt;th&lt;/sup&gt; - 1&lt;sup&gt;st&lt;/sup&gt;</td>
</tr>
<tr>
<td>2.97</td>
<td>Layout of electric vehicle (Patil and Davis, 1990)</td>
</tr>
<tr>
<td>2.98</td>
<td>Experimental results for integrated motor-transmission control</td>
</tr>
<tr>
<td>2.99</td>
<td>Federal urban driving schedule</td>
</tr>
<tr>
<td>2.100</td>
<td>Neutral idle control system (Brown and Kraska, 1993)</td>
</tr>
<tr>
<td>Page</td>
<td></td>
</tr>
<tr>
<td>------</td>
<td></td>
</tr>
<tr>
<td>2.101 Adaptive idling (Leising et al., 1990a)</td>
<td>111</td>
</tr>
<tr>
<td>2.102 Time chart of neutral control (Hayabuchi et al., 1996)</td>
<td>113</td>
</tr>
<tr>
<td>2.103 Control method in phase 2 (Hayabuchi et al., 1996)</td>
<td>114</td>
</tr>
<tr>
<td>2.104 Heat generation vs. C-1 clutch application time lag by calculation in phase 3 (Hayabuchi et al., 1996)</td>
<td>115</td>
</tr>
<tr>
<td>2.105 Simulation model of neutral control (Hayabuchi et al., 1996)</td>
<td>115</td>
</tr>
<tr>
<td>2.106 Control method of c-1 clutch pressure during idling in phase 3</td>
<td>116</td>
</tr>
<tr>
<td>2.107 Fuel injected quantity at the moment of neutral control</td>
<td>116</td>
</tr>
<tr>
<td>2.108 Fuel economy improvement vs. vehicle mean speed</td>
<td>116</td>
</tr>
<tr>
<td>3.1 Powertrain model (with emphasis on the transmission)</td>
<td>120</td>
</tr>
<tr>
<td>3.2 Overview of the Simulink diagram of the powertrain model</td>
<td>121</td>
</tr>
<tr>
<td>3.3 Stick diagram for automatic transmission</td>
<td>128</td>
</tr>
<tr>
<td>3.4 Stick diagram of gear train including gear inertia</td>
<td>129</td>
</tr>
<tr>
<td>3.5 Freebody diagrams of the transmission subassemblies in the first gear</td>
<td>133</td>
</tr>
<tr>
<td>3.6 Band clutch engagement modes (Fanella, 1994)</td>
<td>138</td>
</tr>
<tr>
<td>3.7 Freebody diagram of automatic transmission during second – third gear upshift</td>
<td>139</td>
</tr>
<tr>
<td>3.8 Torque patterns for the second – third gear upshift</td>
<td>140</td>
</tr>
<tr>
<td>3.9 Freebody diagrams for the transmission during third-to-fourth gear upshift and in fourth gear</td>
<td>143</td>
</tr>
<tr>
<td>3.10 Overview Simulink block diagram of the transmission mechanical system</td>
<td>148</td>
</tr>
<tr>
<td>3.11 Hydraulic network (Tugcu et al., 1986)</td>
<td>151</td>
</tr>
<tr>
<td>3.12 2nd clutch-accumulator assembly</td>
<td>152</td>
</tr>
<tr>
<td>3.13 Gross vehicle dynamics model</td>
<td>156</td>
</tr>
<tr>
<td>3.14 Freebody diagrams for vehicle dynamics model (Cho, 1987)</td>
<td>157</td>
</tr>
<tr>
<td>3.15 Turbine and subassembly speeds during 1st-4th gear upshifts</td>
<td>163</td>
</tr>
<tr>
<td>3.16 Transmission output shaft torque $T_2$</td>
<td>165</td>
</tr>
<tr>
<td>3.17 Torque converter turbine and pump torques $T_t$ and $T_p$</td>
<td>166</td>
</tr>
<tr>
<td>3.18 Vehicle velocity $V$ (kmph)</td>
<td>166</td>
</tr>
<tr>
<td>3.19 Torque transfer during 1st-2nd gear upshift</td>
<td>167</td>
</tr>
<tr>
<td>3.20 Torque transfer during 2nd-3rd gear upshift</td>
<td>167</td>
</tr>
<tr>
<td>3.21 Torque patterns during 2nd-3rd gear upshift</td>
<td>168</td>
</tr>
<tr>
<td>3.22 Torque transfer during 3rd-4th gear upshift</td>
<td>169</td>
</tr>
<tr>
<td>3.23 TypeA and TypeB friction characteristics</td>
<td>170</td>
</tr>
<tr>
<td>3.24 Engine, turbine, input sungear and reaction ring gear speeds with Type A Friction, $P_{a2} = 100kPa$</td>
<td>172</td>
</tr>
<tr>
<td>3.25 Shaft torque with Type A Friction and $P_{a2} = 100kPa$</td>
<td>173</td>
</tr>
<tr>
<td>3.26 Clutch pressure with Type A Friction and $P_{a2} = 100kPa$</td>
<td>173</td>
</tr>
<tr>
<td>3.27 Engine, turbine, input sungear and reaction ring gear speeds with Type A friction characteristics and $P_{a2} = 180kPa$</td>
<td>174</td>
</tr>
<tr>
<td>3.28 Output shaft torque with Type A friction characteristics and $P_{a2} = 180kPa$</td>
<td>174</td>
</tr>
<tr>
<td>3.29 Clutch pressure with Type A friction characteristics and $P_{a2} = 180kPa$</td>
<td>175</td>
</tr>
</tbody>
</table>
3.30 Engine, turbine, input sun gear and reaction ring gear speeds with TypeB Friction and $P_{a2} = 100kPa$ .......................................................... 175
3.31 Shaft torque with TypeB friction characteristics and $P_{a2} = 100kPa$ .................................................. 176
3.32 Clutch pressure with TypeB friction and $P_{a2} = 100kPa$ .................................................. 176
3.33 Clutch pressure traces at different bulk modulus values ........................................ 178
3.34 Shift duration traces at different bulk modulus values ........................................ 178
3.35 Magnified view of shaft torque with Type A friction characteristics and different bulk modulus values ........................................ 179
3.36 Clutch pressure traces under different temperatures ........................................ 181
3.37 Shift duration under different temperatures .................................................. 181
3.38 Shaft torque under different temperatures .................................................. 182
3.39 Clutch plate friction coefficient under different temperatures ........................................ 182
3.40 Simulation results for compliant shaft model (without tire-road interaction):
   speeds during 1-2 power-on upshift .................................................. 184
3.41 Simulation results for compliant shaft model (without tire-road interaction):
   torques during 1-2 power-on upshift .................................................. 185
3.42 Simulation results for compliant shaft model with tire-road interaction: speeds and tire slip during 1-2 power-on upshift .................................................. 186
3.43 Simulation results for compliant shaft model with tire-road interaction: torques during 1-2 power-on upshift .................................................. 187
3.44 Comparison of output shaft torque and turbine speed traces during 1-2 power-on upshift: stiff shaft (SS), compliant shaft without tire-road interaction (CSNT), and compliant shaft with tire-road interaction (CST) .................................................. 188
3.45 Comparison of load transfer torque traces during 1-2 power-on upshift: stiff shaft (SS), compliant shaft without tire-road interaction (CSNT), and compliant shaft with tire-road interaction (CST) .................................................. 189
4.1 Frequency response $\omega_{1}/P_{a2} = 100$ kPa, 20%, 50% and 90% into inertia phase .................................................. 196
4.2 Frequency response $\omega_{1}/P_{a2} = 20$ kPa, 100kPa and 180 kPa (50% into inertia phase) .................................................. 196
4.3 Frequency response $\omega_{1}/P_{a2} = 100$ kPa, 90% into inertia phase, Type A and B friction characteristics .................................................. 200
4.4 Frequency response of the 7th order model .................................................. 201
4.5 Frequency response of the 4th order model .................................................. 202
4.6 Frequency response of the 4th order model and solenoid .................................................. 204
4.7 Control system block diagram .................................................. 204
4.8 Frequency response of the controller .................................................. 206
4.9 Frequency response of the compensated system .................................................. 206
4.10 Simulation results for $-100$ rad/sec$^2$ slope of $\omega_{t}$: turbine speed .................................................. 208
4.11 Simulation results for $-100$ rad/sec$^2$ slope of $\omega_{t}$: clutch pressure and the accumulator back pressure .................................................. 209
4.12 Simulation results for $-100$ rad/sec$^2$ slope of $\omega_{t}$: shaft torque .................................................. 209
4.13 Simulation results for $-200$ rad/sec$^2$ slope of $\omega_{t}$: turbine speed .................................................. 210
4.14 Simulation results for $-200$ rad/sec$^2$ slope of $\omega_r$: clutch pressure and the accumulator back pressure ............................................................. 210
4.15 Simulation results for $-200$ rad/sec$^2$ slope of $\omega_r$ ......................................................... 211
4.16 Comparison of the frequency response of the system with original and modified accumulator parameters (50% into the inertia phase) ................................................................. 213
4.17 Comparison of the frequency response of the compensated systems with original and modified accumulator parameters ........................................ 213
4.18 Pressure versus time with modified accumulator, $P_{a2} = 100kPa$ ........................................ 214
4.19 Shaft torque versus time with modified accumulator, $P_{a2} = 100kPa$ ........................................ 214
4.20 Simulation results for $-200$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: turbine speed .......... 215
4.21 Simulation results for $-200$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: clutch pressure and the accumulator backpressure ...................................................... 215
4.22 Simulation results for $-200$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: shaft torque ................. 216
4.23 Simulation results for $-300$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: turbine speed ................. 216
4.24 Simulation results for $-300$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: clutch pressure and accumulator backpressure .............................................................. 217
4.25 Simulation results for $-300$ rad/sec$^2$ slope $-\frac{1}{4} F_{a20}, \frac{1}{2} A_{a2}$: shaft torque ................. 217
4.26 Output shaft torque during an upshift ................................................................................. 220
4.27 Comparison of output shaft torque during the torque phase for 40 degree throttle opening, 12 degree spark and 12 $\rightarrow$ 30 degree spark ................................................................. 220
4.28 Comparison of output shaft torque during the torque phase for 90 degree throttle opening, 12 degree spark and 12 $\rightarrow$ 30 degree spark ................................................................. 222
4.29 Effect of combined throttle and spark control ................................................................. 223
4.30 Open loop control of torque and inertia phase shaft torque ........................................ 227
4.31 Comparison of throttle angle response and 2$^{nd}$ clutch pressure using two open loop control schemes ................................................................. 227
5.1 Photograph of test setup ................................................................................................. 232
5.2 Schematic of test setup ................................................................................................. 233
5.3 Physical schematic of the power-stage of the pressure control valve ......................... 234
5.4 Schematic of hydraulic system .................................................................................... 237
5.5 Data set 110322: Pressure and force data versus solenoid input voltage ....................... 244
5.6 Data set 110327: Pressure and force data versus solenoid input voltage ....................... 244
5.7 Data set 130328: Pressure and force data versus solenoid input voltage ....................... 245
5.8 Comparison of measured leakage flow rates at different feeding chamber pressures ................................................................. 246
5.9 Comparison of computed leakage resistance at different feeding chamber pressures ................................................................. 247
5.10 Comparison of simulation and experimental results for feeding chamber pressure: check ball at top location ................................................................. 250
5.11 Comparison of simulation and experimental results for outer chamber pressure: check ball at top location ................................................................. 250
5.12 Comparison of simulation and experimental results for spool valve back chamber pressure: check ball at top location ................................................................. 251
5.13 Measured pressures at different locations: check ball at top location ................................................................. 251
5.14 Simulation results for spool valve opening for check ball at top location ................................................................. 252

xviii
### Table of Contents

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.15</td>
<td>Comparison of simulation and experimental results for feeding chamber pressure: check ball at bottom location</td>
<td>252</td>
</tr>
<tr>
<td>5.16</td>
<td>Comparison of simulation and experimental results for clutch outer chamber pressure: check ball at bottom location</td>
<td>253</td>
</tr>
<tr>
<td>5.17</td>
<td>Comparison of simulation and experimental results for spool valve back chamber pressure: check ball at bottom location</td>
<td>253</td>
</tr>
<tr>
<td>5.18</td>
<td>Measured pressures at different locations: check ball at bottom location</td>
<td>254</td>
</tr>
<tr>
<td>5.19</td>
<td>Simulation results for spool valve opening: check ball at bottom location</td>
<td>254</td>
</tr>
<tr>
<td>5.20</td>
<td>Measured pilot pressure, clutch outer chamber pressure and back chamber pressure: checkball at the top location</td>
<td>258</td>
</tr>
<tr>
<td>5.21</td>
<td>Measured pilot pressure, clutch outer chamber pressure and back chamber pressure: checkball at the bottom location</td>
<td>258</td>
</tr>
<tr>
<td>5.22</td>
<td>Comparison of identified and measured frequency response data</td>
<td>259</td>
</tr>
<tr>
<td>5.23</td>
<td>Step response comparison between simulation and experimental data</td>
<td>261</td>
</tr>
<tr>
<td>5.24</td>
<td>Step response comparison between simulation and experimental data</td>
<td>262</td>
</tr>
<tr>
<td>5.25</td>
<td>Step response comparison between simulation and experimental data</td>
<td>263</td>
</tr>
<tr>
<td>5.26</td>
<td>Step response comparison between simulation and experimental data</td>
<td>264</td>
</tr>
<tr>
<td>5.27</td>
<td>Operating conditions</td>
<td>270</td>
</tr>
<tr>
<td>5.28</td>
<td>Bode plots of $\frac{P_{co}}{P_p}(s)$ (solid line) and $\frac{P_b}{P_p}(s)$ (dashed line)</td>
<td>273</td>
</tr>
<tr>
<td>5.29</td>
<td>Bode plots of $\frac{P_{co}}{P_p}(s)$: full order (solid line) and reduced order (dashed line)</td>
<td>274</td>
</tr>
<tr>
<td>5.30</td>
<td>Step response of power stage to step change in pilot pressure</td>
<td>275</td>
</tr>
<tr>
<td>5.31</td>
<td>Two degree-of-freedom controller structure</td>
<td>276</td>
</tr>
<tr>
<td>5.32</td>
<td>Relationship of neutral-idle valve pressures to solenoid input</td>
<td>279</td>
</tr>
<tr>
<td>5.33</td>
<td>Bode plot of $G_{pi}(s)$</td>
<td>280</td>
</tr>
<tr>
<td>5.34</td>
<td>Bode plot of $G_{e-id}(s) \cdot G_p(s)$</td>
<td>281</td>
</tr>
<tr>
<td>5.35</td>
<td>Bode plot of $G_{e-pid}(s) \cdot G_p(s)$</td>
<td>283</td>
</tr>
<tr>
<td>5.36</td>
<td>Bode plot of the designed closed loop system with “Lead+integration” (solid line) and “PID” (dashed line) controllers</td>
<td>283</td>
</tr>
<tr>
<td>5.37</td>
<td>Pressure trace</td>
<td>284</td>
</tr>
<tr>
<td>5.38</td>
<td>Comparison of back chamber pressure under open loop and closed-loop control results for trajectory A: (a) Overview, (b) Magnified view</td>
<td>288</td>
</tr>
<tr>
<td>5.39</td>
<td>Comparison of control effort: solenoid input voltages under different control algorithms for trajectory A</td>
<td>289</td>
</tr>
<tr>
<td>5.40</td>
<td>Comparison of error under different control algorithms for trajectory A</td>
<td>289</td>
</tr>
<tr>
<td>5.41</td>
<td>Pressures under feedforward control for trajectory A</td>
<td>290</td>
</tr>
<tr>
<td>5.42</td>
<td>Pressures under “Feedforward+Integral+Lead” control for trajectory A</td>
<td>291</td>
</tr>
<tr>
<td>5.43</td>
<td>Pressures under “Feedforward+PID” control for trajectory A</td>
<td>292</td>
</tr>
<tr>
<td>5.44</td>
<td>Comparison of back chamber pressure under different control algorithms for trajectory B: (a) Overview, (b) Magnified view</td>
<td>293</td>
</tr>
<tr>
<td>5.45</td>
<td>Comparison of control effort: solenoid input voltage under different control algorithms for trajectory B</td>
<td>294</td>
</tr>
</tbody>
</table>
5.46 Comparison of error under different control algorithms for trajectory B .......... 294
5.47 Pressures under “Feedforward” control for trajectory B .................................. 295
5.48 Pressures under “Feedforward+Integral+Lead” control for trajectory B .......... 296
5.49 Pressures under “Feedforward+PID” control for trajectory B ...................... 297
5.50 Comparison of back chamber pressure under different control algorithms for trajectory C ................................................................................................................. 298
5.51 Comparison of control effort: solenoid input voltage under different control algorithms for trajectory C ........................................................................................ 299
5.52 Comparison of error under different control algorithms for trajectory C .......... 299
5.53 Pressures under “Feedforward” control for trajectory C .................................. 300
5.54 Pressures under “Feedforward+Integral+Lead” control for trajectory C .......... 301
5.55 Pressures under “Feedforward+PID” control for trajectory C ...................... 302
5.56 Comparison of back chamber pressure under different control algorithms for trajectory D ...................................................... 303
5.57 Comparison of control effort: solenoid input voltage under different control algorithms for trajectory D ................................................................. 304
5.58 Comparison of error under different control algorithms for trajectory D .......... 304
5.59 Pressures under “Feedforward” control for trajectory D .................................. 305
5.60 Pressures under “Feedforward+Integral+Lead” control for trajectory D .......... 306
5.61 Pressures under “Feedforward+PID” control for trajectory D ...................... 307
5.62 Sensitivity function of the system with PID and “Lead+Integration” controllers ...................................................................................................................................... 310
5.63 Stability robustness measure of PID and “Lead+Integrator” controller when temperature is changed to 0°C ................................................................. 311
5.64 Stability robustness measure of PID and “Lead+Integrator” controllers when the temperature changes to 150°C ......................................................... 311
5.65 Stability robustness measure of PID and “Lead+Integrator” controllers with solenoid model parameter change ................................................................. 313
5.66 Variation of $K_{bpool}$ at different operating points ........................................ 313
5.67 Stability robustness measure of PID and “Lead+Integrator” controllers with solenoid model parameter change and $K_{bpool}$ change ................................................................. 314
6.1 Torque ratio versus speed ratio for the torque converter .................................. 324
6.2 $K$-factor (rad/sec/$\sqrt{Nm}$) versus speed ratio for the torque converter ....... 324
6.3 Magnitude frequency responses of reduced order system ($F_r; G_r; H_r; J_r$) at the seven different operating conditions ................................................................. 348
6.4 Explicit Model Following structure .................................................................. 352
6.5 Shift quality versus rate of turbine speed change during forward clutch application (Hayabuchi et al., 1996) ................................................................. 356
6.6 Speed trajectories at different damping ratios, step input: 680 rpm, $\omega_n = 20$ rad/sec ................................................................. 357
6.7 Rate of speed change at different damping ratios, step input: 680 rpm, $\omega_n = 20$ rad/sec ....................... 358
6.8 Human response sensitivity to frequency of vibration .................................. 358
6.9 Fast tracking: Engine, turbine and vehicle speed trajectories ................................363
6.10 Fast tracking: Vehicle acceleration, output shaft torque and jerk ..................364
6.11 Fast tracking: Throttle angle and forward clutch pressure .............................365
6.12 Engine, turbine, slip trajectory and vehicle speed for $\omega_{ne} = 30\text{ rad/sec}$ 368
6.13 Vehicle acceleration, output shaft torque and jerk for $\omega_{ne} = 30\text{ rad/sec}$ ..369
6.14 Throttle angle and forward clutch pressure response $\omega_{ne} = 30\text{ rad/sec}$ ...370
6.15 Comparison of engine and turbine speed traces with different $\omega_{ne}, \omega_{er} = 500\text{rpm}$ .................................371
6.16 Output shaft torque and jerk with different $\omega_{ne}, \omega_{er} = 500\text{rpm}$ ........372
6.17 Throttle and clutch pressure with different $\omega_{ne}, \omega_{er} = 500\text{rpm}$ ........372
6.18 Engine and turbine speed trajectories: $\omega_{er} = 100\text{rpm}, \omega_{ne} = 30\text{ rad/sec}$ 373
6.19 Comparison of output shaft torque and jerk with different engine speed steps ..373
6.20 Comparison of throttle angle and clutch pressure with different engine speed steps .................................................................374
6.21 Singular values of return ratio for the two designs ........................................376
6.22 Singular values of $S(j\omega)$ for the two designs ........................................377
6.23 Singular values of $T(j\omega)$ for the two designs ........................................377
6.24 Comparison of engine and turbine speeds with and without AFR disturbance: $\omega_{ne} = 30\text{ rad/sec}$ ..........................................................378
6.25 Comparison of output shaft torque and jerk with and without AFR disturbance: $\omega_{ne} = 30\text{ rad/sec}$ ..........................................................379
6.26 Slow response case: Engine, turbine speed and vehicle speed trajectories ....383
6.27 Slow response case: Vehicle acceleration, output shaft torque and jerk ........384
6.28 Slow response case: Throttle angle and forward clutch pressure .................385
NOMENCLATURE

A transformation matrices defined in Table 3.2
contact area between the spool and valve sleeve
$A_{a2}$ 1-2 accumulator piston area
$A_{ai}$ $i^{th}$ accumulator piston area
$A_c$ Clutch Area
$A_{c1}$ $C_1$ clutch piston area
$A_{c2}$ $C_2$ clutch piston area
$A_{ci} \cdot A_{bj}$ $C_i$ clutch plate / $B_j$ band piston area
$A_{d1}$ discharge orifice cross-sectional area on the $C_1$ back pressure side
$A_{d2}$ discharge orifice cross-sectional area for the 1-2 accumulator
$A_{e2}$ 2nd exhaust valve area
$A_{i1}$ $C_1$ input valve orifice cross-sectional area
$A_{i2}$ 1-2 shift valve area
$A_{th}$ throttle plate open area
$AFI$ normalized air fuel influence
$AFR$ air/fuel ratio
$A_v$ frontal area of vehicle
$a_{ci}$ area of the supply orifice to the inner clutch chamber
$a_{co}$ restriction orifice between the inner and outer clutch chambers
$a_f$ feedback orifice area connecting the valve back chamber with the feeding chamber
$a_{ij}$ the $i^{th}$ row, $j^{th}$ column element of matrix A
$a_{kvy}$ cumulative flow area associated with leakage paths between the feeding chamber and the exhaust
$a_{kx}$ leak through the valve mounting surface with an equivalent orifice area
$a_{ks}$ Another leakage path is the shaft end leakage path characterized by leakage area
$a_{sp}$ right side of the spool at area
$a_{sp2}$ secondary spool cross sectional area
\( a_s \) spool valve opening area
\( a_t \) orifice with flow area
\( B \) transformation matrices defined in Table 3.2
\( B_{si} \) viscous damping coefficient for the \( i^{th} \) accumulator piston
\( B_{Ci}, B_{ci} \) viscous damping coefficient for the \( Ci \) clutch piston
\( B_{c2} \) viscous damping coefficient for the clutch piston motion
\( B_j \) band \( B_{12} \) or \( B_R \),
\( B_{12} \) 1st - 2nd shift band
\( B_R \) reverse band
\( B_{vsp} \) friction coefficient between the valve spool and sleeve
\( b_{ij} \) the \( i^{th} \) row, \( j^{th} \) column element of \( B \) matrix
\( C \) transformation matrices defined in Table 3.2
\( C_i \) clutches \( C_1,C_2,C_3 \) or \( C_4 \)
\( C_a \) aerodynamic drag coefficient
\( C_{co} \) lumped parameter representing the fluid compliance
\( C_{D,th} \) throttle discharge coefficient
\( C_d \) the orifice discharge coefficient, orifice flow discharge coefficient
\( c_{ij} \) the \( i^{th} \) row, \( j^{th} \) column element of \( C \) matrix
\( c_T \) torque constant
\( D \) transformation matrices defined in Table 3.2
\( D_{th} \) throttle bore diameter
\( d_{ij} \) the \( i^{th} \) row, \( j^{th} \) column element of \( D \) matrix
\( d_{sp2} \) spool diameter at the spring end
\( d_{sp1} \) spool diameter at the pilot end
\( E \) transformation matrices defined in Table 3.2
\( e_{ij} \) the \( i^{th} \) row, \( j^{th} \) column element of the \( E \) matrix in Table 3.2
\( flag \) shifting status signal defined in Table 3.4
\( F \) transformation matrices defined in Table 3.2
\( F_a \) aerodynamic drag force
\( F_{a2o} \) accumulator spring preload
\( F_c \) clutch release spring force
\( F_{c2} \) force acting on the clutch piston by the clutch pack
\( F_f \) instantaneous friction force
\( F_g \) gradient resistance = \( M_g \cdot g \cdot \sin(\theta) \)
\( F_{k0} \) spring preload
\( F_s \) spring force as a function of wave spring compression
\( F_{gf} \) combined front tire force
\( F_{tr} \) combined rear tire force
\( F_v \) viscous friction force
$f_{ij}$: $i^{th}$ row, $j^{th}$ column element of F matrix

$G(s)$: true plant transfer function

$G_c$: controller transfer function

$G_{c_{-f}}(s)$: feedforward controller transfer function

$G_{c_{-p}}(s)$: cascade controller transfer function

$G_n(s)$: nominal plant transfer function

$G_p(s)$: linearized plant model

$G_s$: solenoid dynamics transfer function

$g$: acceleration due to gravity

$h_f$: static ground-to-axle height of the front wheel

$h_r$: static ground-to-axle height of the rear wheel

$i, i_d$: tire slip/skid

$i_f, i_r$: front and rear tire slip (skid)

$I_e$: effective inertia of engine and torque converter pump

$I_{Cr}$: inertia of the reaction carrier - input ring gear assembly

$I_p$: pump inertia

$I_{Rr}$: inertia of the reaction ring gear - input carrier assembly

$I_{Si}$: inertia of the input sun gear assembly

$I_{Sr}$: inertia of the reaction sun gear assembly

$I_{sol}$: input current to the linear solenoid

$I_t$: lumped inertia of the torque converter turbine assembly

$I_v$: equivalent moment of inertia of vehicle

$I_{wF}$: combined inertia of right and left front wheels

$I_{wr}$: combined inertia of right and left rear wheels

$J$: cost function

$k, K_s$: output shaft compliance (spring rate)

$k_t$: tangential stiffness of the static contact

$K_{a2}$: 1-2 accumulator spring stiffness, assumed to be constant

$K_c$: controller parameter

$K_{c2}$: clutch release spring stiffness, assumed to be constant

$K_{ci}$: $C_i$ clutch release spring stiffness

$K_{34i}$: inertia matrix

$K_e$: effective spring constant

$K_{eng1} - K_{eng4}$: engine model constants

$k_{ERC}$: gain of the throttle position control loop

$K_o$: steady-state gain

$K_z$: mechanical spring constant of the preload spring

$k_{scale}$: scaling factor

xxiv
$K_{s2}$ the solenoid gain

$K_{wal}$ linear spring rate of the wave spring

$K_{wanl}$ a constant in the nonlinear wave spring relationship

$M,m$ vehicle mass

$m_a$ intake manifold air mass

$m_{a,cyl}$ mass flow rate of air into each cylinder

$m_{ai}$ total mass flow rate of air entering the intake manifold

$m_{a,m}$ mass of air in the intake manifold

$m_{a0}$ mass flow rate of air entering the combustion chamber

$m_{ai,th}$ mass flow rate of air entering the intake manifold through the throttle plate

$MAX$ maximum flow rate at wide-open-throttle

$MBT$ maximum brake torque

$m_{fc}$ commanded fuel rate

$m_{fi}$ actual fuel rate entering the combustion chamber

$m_{vap}$ spool mass

$n_c$ number of clutch plates

$n_{cyl}$ number of cylinders

$N_{Si}$ number of teeth of the input sun gear

$N_{Sr}$ number of teeth of the reaction sun gear

$N_{Ri}$ number of teeth of the input ring gear

$N_{Rr}$ number of teeth of the reaction ring gear

$\Delta N_e$ increase in engine speed

$P_{a2}$ 1-2 accumulator back pressure

$P_{ao2}$ discharge orifice back pressure

$P_{atm}$ atmospheric pressure

$P_{Bj}$ band hydraulic pressure, $B_j = B_{12}$ or $B_R$

$P_{B0}$ band hydraulic pressure level

$P_b$ spool valve back chamber pressure

$P_{b\text{-desired}}$ desired back chamber pressure trajectory

$P_{b\text{-IC}}$ initial condition of the desired trajectory

$P_{b\text{-measured}}$ measured back chamber pressure

$P_c$ clutch hydraulic pressure

$P_{cf}$ feeding chamber pressure

$P_{ci}$ clutch hydraulic pressure, clutch inner chamber pressure

$P_{co}$ clutch outer chamber pressure

xxv
$P_{clo}$  preload (output from the linear solenoid) on the back pressure
$P_{clo}$  back pressure of the C1 clutch
$P_f, P_{fo}$  feeding chamber pressure
$P_l$  main hydraulic line pressure
$P_m$  intake manifold pressure
$P_o$  actual ambient pressure (stagnation)
$P_{o,cal}$  ambient pressure at calibration
$P_p$  pilot pressure
$P_x$  pressure at an appropriate intermediate point between the feeding chamber and the inner clutch chamber

$PRI$  normalized pressure influence
$Q$  the torus volumetric flow rate
$Q_{cm}$  volumetric flow rate to clutch outer chamber
$Q_{1in}$  inflow rate into the C1 clutch cavity
$Q_{1out}$  outflow rate from the 1st exhaust valve
$Q_{2in}$  inflow rate into the 2nd clutch cavity
$Q_{2out}$  outflow rate from the 2nd exhaust valve
$Q_{a2}$  flow rate to the 1-2 accumulator
$Q_{c1}$  flow rate from the C1 clutch
$Q_{c2}$  flow rate to the 2nd clutch
$Q_{ci}$  volumetric flow rate to clutch inner chamber

$Q_{leak}$  total leakage
$Q_{dv}$  volumetric flow rate through flow area $A_{dv}$
$Q_{dbm}$  leakage volumetric flow rate at valve mount
$Q_{lqe}$  leakage volumetric flow rate at shaft end
$Q_{i}$  volumetric flow rate through orifice area $a_i$
$Q_{l}$  volumetric flow rate out of feeding chamber, through orifice area $a_i$

$Q_2$  volumetric flow rate from valve back chamber to feeding chamber
$r$  tire (free-rolling) radius
$r_c$  clutch Radius
$r_e$  effective rolling radius ($v/\omega$)
$R$  ideal gas constant
$R_{ci}, R_{Bj}$  Ci clutch plate / Bj band effective radius
$R_D$  final drive gear ratio

$R_{Si}, R_{Sp}, R_{Rl}, R_{Rr}$  kinematic constants defined in Table 3.2
$RT_{sp1}$  reaction torque of the first clutch
$RT_{sp3}$  reaction torque of the third clutch
$RT_{B12}$  reaction torque of the B12 band
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>RT_{C2}</td>
<td>reaction torque of C2 clutch</td>
</tr>
<tr>
<td>RT_{C4}</td>
<td>reaction torque of the C4 clutch</td>
</tr>
<tr>
<td>R_{xf}, R_{xr}</td>
<td>reaction force on the front/rear wheel</td>
</tr>
<tr>
<td>s</td>
<td>Laplace operator</td>
</tr>
<tr>
<td>SA</td>
<td>spark advance (degrees before TDC)</td>
</tr>
<tr>
<td>S_e</td>
<td>lumped parameter representing the stiffness</td>
</tr>
<tr>
<td>SI</td>
<td>normalized spark influence</td>
</tr>
<tr>
<td>T</td>
<td>controller parameter</td>
</tr>
<tr>
<td>T_{acc}, T_a</td>
<td>engine accessory load</td>
</tr>
<tr>
<td>T_{Bj}</td>
<td>band torque capacity, bj=B_{12} or B_R</td>
</tr>
<tr>
<td>T_{brake}</td>
<td>minimum brake torque holding vehicle</td>
</tr>
<tr>
<td>T_c</td>
<td>clutch torque</td>
</tr>
<tr>
<td>T_{ci}</td>
<td>i^{th} clutch torque capacity, i = 1,2,3,4</td>
</tr>
<tr>
<td>T_{sid}</td>
<td>dynamic torque acting on the inertia I_{si}</td>
</tr>
<tr>
<td>T_{TC}</td>
<td>normalized throttle characteristic</td>
</tr>
<tr>
<td>T_e (T_{net})</td>
<td>net engine torque</td>
</tr>
<tr>
<td>T_f</td>
<td>engine friction torque</td>
</tr>
<tr>
<td>T_{fl}</td>
<td>kinematic torque acting on the inertia I_{Rr}</td>
</tr>
<tr>
<td>T_{flid}</td>
<td>dynamic torque acting on the inertia I_{Rr}</td>
</tr>
<tr>
<td>T_{p2}</td>
<td>kinematic torque acting on the inertia I_{Sr}</td>
</tr>
<tr>
<td>T_{pd}</td>
<td>dynamic torque acting on the inertia I_{Sr}</td>
</tr>
<tr>
<td>T_i</td>
<td>indicated engine torque</td>
</tr>
<tr>
<td>T_{Li}</td>
<td>road load torque due to rolling resistance and aerodynamic drag</td>
</tr>
<tr>
<td>T_m</td>
<td>intake manifold temperature</td>
</tr>
<tr>
<td>T_{net}</td>
<td>net engine torque</td>
</tr>
<tr>
<td>T_o</td>
<td>kinematic torque acting on the inertia I_{cr}</td>
</tr>
<tr>
<td>T_{o,cal}</td>
<td>actual ambient temperature (absolute)</td>
</tr>
<tr>
<td>T_{od}</td>
<td>ambient absolute temperature during calibration, =298 K</td>
</tr>
<tr>
<td>T_p</td>
<td>torque converter pump torque</td>
</tr>
<tr>
<td>T_{rf}</td>
<td>combined front tire rolling resistance torque</td>
</tr>
<tr>
<td>T_{rr}</td>
<td>combined rear tire rolling resistance torque</td>
</tr>
<tr>
<td>T_S</td>
<td>output shaft torque of the transmission gear train</td>
</tr>
<tr>
<td>T_{Si}</td>
<td>kinematic torque acting on the inertia I_{si}</td>
</tr>
<tr>
<td>T_{sid}</td>
<td>kinematic torque</td>
</tr>
<tr>
<td>T_{S_{torque}}</td>
<td>output shaft torque during torque phase</td>
</tr>
<tr>
<td>T_{S_{inertia}}</td>
<td>output shaft torque during inertia phase</td>
</tr>
<tr>
<td>T_t</td>
<td>turbine torque</td>
</tr>
<tr>
<td>t_2</td>
<td>dwell time, time at zero velocity</td>
</tr>
<tr>
<td>t_3</td>
<td>the time instant to depressurize the band B_{12}</td>
</tr>
<tr>
<td>V</td>
<td>vehicle speed</td>
</tr>
<tr>
<td>V_e, V_d</td>
<td>engine displacement</td>
</tr>
</tbody>
</table>
$V_{c1}$: volume at pressure $P_{c1}$

$V_{clo}$: initial volume of the C1 clutch cavity

$V_{c2}$: volume at pressure $P_{c2}$

$V_f$: feeding chamber volume

$V_{fin}$: the volume of fluid at pressure $P_{c2}$

$V_{in}$: solenoid input voltage

$V_{kph}$: vehicle velocity in kilometers per hour.

$V_m$: intake manifold volume

$V_s$: Solenoid voltage

$V_{sol}$: command solenoid voltage

$V_{sol_{-}IC}$: initial solenoid voltage

$V_v$: the vehicle speed

$V_{s2}$: the input voltage to the solenoid

$x_{a2}$: 1-2 accumulator piston motion

$x_c$: wave spring compression

$x_{cl}$: C1 clutch piston motion

$x_{clm}$: maximum deflection of the clutch spring

$x_{cl\text{max}}$: maximum clutch piston displacement

$x_{cl\text{min}}$: minimum clutch piston displacement

$x_{cm}$: maximum compression of the wave plate

$x_{c2}$: 2nd clutch piston motion

$x_{vsp}$: valve spool displacement from initial (preloaded) position

$\alpha$: throttle angle (deg), controller parameter

$\alpha_{com}$: commanded throttle input

$\beta$: desired air-to-fuel ratio, fluid bulk modulus

$\beta_e$: effective bulk modulus

$\delta_{SR}$: stability robustness measure

$\Delta P_b$: tracking error

$\Delta r$: radial clearance between the spool and valve sleeve

$\Delta t_{it}$: intake to torque production delay

$\Delta t_{sf}$: spark to torque production delay

$\Lambda$: Relative Gain Array (RGA)

$\mu$: friction coefficient, fluid viscosity

$\eta_{vol}$, $\eta_r$: engine volumetric efficiency

$\lambda_{ij}$: relative gain

$\rho$: mass density of transmission fluid

$\rho_a$: ambient air density

$\theta$: road grade
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\theta_{Bj}$</td>
<td>band wrap angle</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>ratio of specific heat, for air = 1.4</td>
</tr>
<tr>
<td>$\tau$</td>
<td>time constant of the pressure profile</td>
</tr>
<tr>
<td>$\tau_{co}$</td>
<td>computed time constant</td>
</tr>
<tr>
<td>$\tau_{ETC}$</td>
<td>time constant of the throttle position control loop</td>
</tr>
<tr>
<td>$\tau_f$</td>
<td>effective fueling time constant</td>
</tr>
<tr>
<td>$\omega$</td>
<td>angular velocity, with the first subscript $S, R$ or $C$ denoting sun gear, ring gear or carrier, the second subscript $i$ or $r$ denoting input gearset or reaction gearset</td>
</tr>
<tr>
<td>$\omega_{Cr}$</td>
<td>reaction carrier assembly angular velocity</td>
</tr>
<tr>
<td>$\omega_e$</td>
<td>engine angular velocity</td>
</tr>
<tr>
<td>$\omega_{sr}$</td>
<td>desired engine speed step input</td>
</tr>
<tr>
<td>$\omega_{ne}$</td>
<td>natural frequency of desired engine speed trajectory</td>
</tr>
<tr>
<td>$\omega_{sr}$</td>
<td>natural frequency of desired turbine speed trajectory</td>
</tr>
<tr>
<td>$\omega_o$</td>
<td>output shaft speed of the transmission gear train</td>
</tr>
<tr>
<td>$\omega_p$</td>
<td>torque converter pump angular velocity</td>
</tr>
<tr>
<td>$\omega_{rr}$</td>
<td>reaction ring gear assembly angular velocity</td>
</tr>
<tr>
<td>$\omega_{si}$</td>
<td>input sun gear assembly angular velocity</td>
</tr>
<tr>
<td>$\omega_{slip}$</td>
<td>slip speed</td>
</tr>
<tr>
<td>$\omega_{sr}$</td>
<td>reaction sun gear assembly angular velocity</td>
</tr>
<tr>
<td>$\omega_{s2}$</td>
<td>solenoid natural frequency</td>
</tr>
<tr>
<td>$\omega_i$</td>
<td>torque converter turbine angular velocity</td>
</tr>
<tr>
<td>$\omega_{ct}$</td>
<td>torque converter turbine reference angular velocity</td>
</tr>
<tr>
<td>$\omega_v$</td>
<td>wheel angular velocity</td>
</tr>
<tr>
<td>$\Delta\omega_{ci,Bj}$</td>
<td>$C_i$ clutch/$B_j$ band slip speed</td>
</tr>
<tr>
<td>$\zeta_i$</td>
<td>damping ratio, $i = e$: engine, $t$: turbine</td>
</tr>
</tbody>
</table>
CHAPTER 1

INTRODUCTION

Electronic control of vehicle functions has grown dramatically in the last two decades as automobile manufacturers have tried to improve vehicle performance and fuel economy while keeping costs down. Among vehicle functions, automotive powertrain control has long been an active research area for both automobile manufacturers and academia. A representative powertrain is shown in Figure 1.1. The two major subsystems in a powertrain are the engine and transmission. Depending on the coupling means between the engine and transmission, the transmission can be of the manual or automatic type. The transmission depicted in Figure 1.1 is a stepped automatic transmission.

Automatic transmissions were first developed in 1934 (MacDuff et al., 1969), and have attracted consumers because of the resulting ease of driving, especially in urban driving. Since then, more and more automatic transmissions have been installed in passenger vehicles. By 1993, over 70% of mid and large size passenger cars in the Japanese market were equipped with automatic transmissions, demonstrating that they have become the predominant transmissions (Yasushi, 1991; Naruse et al., 1993).

1.1 Motivation for proposed research

With the extension of electronic control to transmission functions, automatic transmissions have ceased to be self-contained systems. Powertrain performance can be
optimized by closer coordination of engine and transmission functions, which in turn requires integration of engine and transmission control.

Figure 1.1 A representative powertrain

The research in this dissertation is directed at the modeling and control of stepped automotive powertrains, that is, the engine-transmission-drivetrain system, with special emphasis on automatic transmissions. Despite the fact that electronic control of transmission functions offers great potential for improved efficiency, convenience, and smoothness of gearshifts, it is less well-developed than electronic control of engine or chassis functions (Hrovat and Powers, 1988; Yu and Moskwa, 1994). Electronic control has been applied routinely to transmission functions such as gearshift scheduling which require implementation of relatively simple control strategies. Far greater benefits can be realized, however, from closed loop control of transmission functions based on sensed feedback. Such closed loop control of transmission functions, when it has been implemented in practice, has relied on heuristic approaches and considerable
experimentation. Development of model-based approaches to transmission controller
design offers the potential for addressing more demanding control applications and
significantly greater gains. Furthermore, coordination of engine and transmission
functions requires integrated engine-transmission control. Improved prediction of
powertrain system behavior is essential to controller design.

The specific powertrain control problems being studied in this research are gear
shift control for a stepped automatic transmission, and neutral-idle control, which is
motivated by fuel economy considerations. Consistency of shift quality can be achieved
by controlling the duration of the inertia phase (speed adjustment phase) of a shift.
Hence, the first control problem studied in this research is a closed-loop controller design
for the inertia phase of a gear shift. Following this, a torque phase control strategy is
considered. In stepped automatic transmissions, there is a torque drop during the torque
phase (load transfer phase) of a gearshift. Analysis of the torque phase reveals that this
torque drop can potentially be reduced or totally avoided if integrated engine and
transmission control is employed. An open-loop control strategy is presented to
demonstrate this concept.

Powertrain neutral idle control requires fully or partially disengaging the input
clutch to the transmission when the vehicle is stopped, or is moving at a very low speed,
so as to put the transmission automatically in neutral. Doing so improves fuel economy in
urban driving and, furthermore, minimizes the engine vibrations transmitted to the
driveline. When the driver removes his foot from the brake pedal and actuates the
accelerator pedal, the input clutch should be automatically engaged smoothly but rapidly,
in order to avoid a sharp rise of the engine speed and a subsequent harsh clutch lockup.
The vehicle is then in first gear.

Neutral-idle control poses several challenging problems. First, the application
phase of the input clutch requires fast and smooth engagement in coordination with
engine control functions. This requires the forward clutch pressure to be directly
controlled. Understanding of the clutch pressure control system is essential to the control
strategy of neutral-idle process, as well as other shift control applications such as clutch-
to-clutch shift control. Thus, theoretical and experimental study of a clutch pressure control system is carried out.

Furthermore, the engine functions have to be controlled in conjunction with the forward clutch engagement process in order to ensure a smooth and rapid engagement. If the forward clutch is applied too fast without engine speed control, the engine may stall. On the other hand, if the forward clutch is applied too slowly, engine speed may race which will cause harsh clutch lockup. Therefore, MIMO (Multi-Input Multi-Output) control strategy for coordinated engine and transmission control is necessary for a successful neutral-idle application.

The challenging nature of the neutral-idle control problem requires that the control strategy be based explicitly on improved quantitative models of powertrain dynamic responses. While there have been some such models reported in the literature (Cho and Hedrick, 1989; Tugcu et al. 1986), a number of research issues are posed by the need for models suitable for closed loop control, and the development and evaluation of such control algorithms.

1.2 Objectives of the research

The primary objectives in the research involve development of improved dynamic models of transmission components for a representative stepped automatic transmission, and use of these models, together with an available engine dynamic model, to devise model-based control strategies for gear shifts and neutral-idle shifts. The research consists of analysis, simulation studies, component testing and implementation of clutch pressure control strategies in a test setup.

Transmission response to control action during gearshifts is usually nonlinear. There are many sources of nonlinearity. Quantitative models of transmission response during a gearshift exhibit marked differences between the two phases which occur during any shift, the load transfer phase involving little speed change in the transmission input speed, and the speed adjustment phase involving significant speed change. In addition, the clutch friction force is normally not a linear function of slip velocity at the clutch.
Development of transmission dynamic models of varying degrees of complexity, and yet suitable for controller design, is the first objective of this research. The second objective of this research is to develop model-based shift control strategies for conventional automatic transmissions. The third objective is to analyze and characterize a solenoid controlled clutch pressure control system via simulation studies and experiments. The final objective is to design an integrated engine-transmission control strategy for the neutral-idle clutch application phase using a MIMO control technique.

1.3 Organization of the dissertation

An extensive literature survey of the current state-of-the art in automotive powertrain modeling and control, with emphasis on automatic transmissions, is provided in Chapter 2. A review of the modeling of powertrain components as well as the system is presented first, followed by a detailed survey of automatic transmission related control problems and control schemes. Limitations of current research on modeling and control of automatic transmissions are discussed.

In Chapter 3, dynamic models of varying complexity of automatic transmission shift responses for a specific transmission are presented. A first – fourth gear upshift dynamic simulation is first implemented using Matlab/Simulink, followed by a complex transmission model incorporating a hydraulic network simulation for first-second shift. Simulation results are presented.

In Chapter 4, control strategies for the first-second gear upshift are developed. Based on the powertrain model developed in Chapter 3, linearized models of the inertia phase of the first-second gearshift are derived. Closed-loop model-based controller design for the inertia phase is carried out, and simulation results are presented. In addition, an open-loop control strategy for the first-second gearshift is presented.

In Chapter 5, theoretical and experimental studies of a clutch pressure control system are presented. An analytical model of a clutch pressure control system is derived and validated by experimental data. A closed-loop controller is designed based on the
validated model, and implemented in the test setup. Both simulation and experimental results are presented.

In Chapter 6, powertrain neutral-idle control is presented. The dynamic model for the powertrain neutral idle process is derived first. The Explicit Model Following (EMF) approach with linear quadratic regulator design is used to control engine and turbine speed trajectories. Simulation results for the controller performance are presented.

Finally, conclusions of the research and recommendations for future work are given in Chapter 7. Detailed Simulink model diagrams are presented in the appendices to help the reader follow the ideas presented in the various chapters.
CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

Automobile manufacturers keep seeking new technologies to improve vehicle performance and lower cost. As more electronic control techniques are applied to powertrains, improved prediction of powertrain system behavior becomes essential to controller design. Improved modeling of powertrain systems can help identify existing problems and aid controller design. Many researchers have worked on automatic transmission modeling and control related research issues. Especially in recent years, modeling and control of automatic transmissions have been active research areas, since they complement earlier efforts of a similar nature directed at engines. Improved control of powertrains requires advances in understanding of the behavior of engines and transmissions.

This literature review focuses on the state of the art of modeling and control of automotive powertrains with emphasis on automatic transmissions. The review is divided into three parts. Section 2.2 describes the structure of a representative automotive powertrain and its main components. Section 2.3 focuses on modeling issues for powertrain systems. Section 2.4 focuses on control issues for powertrain systems. The nomenclature employed by the reviewed references is retained here. No attempt is made to define all the symbols used, as the intent of the review is to present the approaches and compare them. More complete quantitative treatments are deferred until chapters 3-6.
2.2 A representative automotive powertrain

Figure 2.1 shows a representative automotive powertrain. There are two main subsystem in a powertrain – the engine and the transmission. The engine is the "power plant" of the powertrain and generates power to drive the car. The transmission, as the name indicates, is the power transfer mechanism which transmits the engine power to the driveline. Depending on the coupling means between the engine and transmission, the transmission can be either manual or automatic. The manual transmission uses a mechanical clutch controlled by the driver to transfer engine power to the transmission, while the automatic transmission employs a hydraulic coupling device called the torque converter to transmit engine power to the transmission. The transmission shown in Figure 2.1 is an automatic transmission.

A stick-diagram representation of the automatic transmission studied in this dissertation is shown in Figure 2.2. An automatic transmission consists of the torque converter, planetary gear sets, hydraulic clutches and bands.

The torque converter is a hydrodynamic device that sends power from the engine to the transmission input shaft. A three-element torque converter is shown in Figure 2.3. The three elements are the pump (impeller) which is attached to the engine, the turbine which is attached to the transmission input shaft, and the stator or reactor which is attached to the housing via a one-way clutch (Rizzoni and Srinivasan, 1998). The torque converter is filled with transmission fluid. When the impeller rotates, it imparts angular momentum to the fluid involved in torus flow or circulatory fluid flow. The stator's vanes catch the oil thrown off from the pump. This circulating fluid flow then imparts angular momentum to the turbine and results in the turbine speed increasing. The turbine rotates and drives the input shaft of the transmission.
Figure 2.1 A representative powertrain (Danno et al., 1989)

Figure 2.2 A representative automatic transmission
Clutches and bands are frictional elements. In automatic transmissions, operation of the clutches and bands is controlled by hydraulic valves. Figure 2.4 shows a typical clutch assembly and its operation. When a clutch is engaged, the input shaft and the output shaft move together. The band operates differently from a clutch. Figure 2.5 shows a typical band assembly. By providing fluid under pressure to the apply side of piston, the band holds the shaft stationary.

Clutches and bands are employed to hold certain members of the planetary gear sets together to realize a gear ratio. Figure 2.6 shows a simple planetary gear set assembly. There are four parts in a planetary gear set – the sun gear, the ring gear (internal gear), the planet pinion and the carrier. By holding one of the elements of the planetary gear set stationary, we get one gear ratio. By changing the stationary element, we can get another ratio. Additionally, by changing the drive element, we can get speed increase, decrease, and direct drive or reverse.
Figure 2.4 A typical clutch assembly (Crouse and Anglin, 1983)

Figure 2.5 A typical band assembly (Crouse and Anglin, 1983)
2.3 Modeling of automotive powertrain systems

Powertrain system modeling has long been an active research topic. There are two approaches commonly employed in powertrain modeling, depending on the primary focus of interest.

One approach involves powertrain modeling with emphasis on engine related research issues and with a relatively simple transmission-driveline model. If gearshifts are not involved, the load (from the transmission) seen by the engine does not change dramatically over a short time interval. So the transmission model used in this situation is simplified.

The second approach involves powertrain modeling with emphasis on transmission related issues and with a relatively simple engine model. While studying shift transients, the engine torque is nearly unchanged if the engine inputs are unchanged. Therefore, the engine model used in this situation can be simplified.

If engine and transmission control is to be integrated, we need to model the transient behavior of both the engine and transmission with greater fidelity. This is the perspective of this dissertation.

Figure 2.6 A simple planetary gear set assembly
This review will focus on powertrain modeling with emphasis on automatic transmission related research issues. Powertrain component modeling is discussed first below, followed by comments related to integrated powertrain modeling.

2.3.1 Engine models for transmission related research

The literature on engine dynamic modeling research is rich. Engine models vary in complexity, in some cases being no more than torque generation maps (Hrovat and Tobler, 1991; Kono et al., 1995), whereas in other cases individual cylinder events are modeled (Kao and Moskwa, 1993, 1995a, b). The level of complexity of engine modeling varies with the application.

Generally speaking, engine dynamic modeling consists of modeling individual subsystem and related phenomena, such as air intake dynamics, fueling dynamics, torque production dynamics, exhaust manifold dynamics, and crankshaft dynamics. Determination of these models requires consideration of issues related to heat transfer, fluid mechanics, thermodynamics etc. However, the focus of this dissertation is on integrated powertrain modeling and control with greater emphasis on automatic transmissions. Thus this review will concentrate on engine models important to transmission oriented research. A comprehensive review of engine models can be found in Dawson (1998).

Two types of engine models commonly used are the mean-value-engine-model and the individual-cylinder-event-model. Numerous studies resulting in both types of models can be found in the literature. An example of the latter approach is Dawson's (1998) model. Dawson developed a control-oriented engine model, which captures the many aspects of engine dynamics. One key contribution of his work is the two-zone combustion model. An overview of Dawson's model is shown in Figure 2.7.
If noise and vibration studies are important, individual-cylinder models are needed. However, for transmission control studies, they are less important. In this case, a mean-value engine model is suitable. It needs to respond to load changes during shifts, throttle change and spark change. Therefore, certain key dynamic phenomena such as air intake dynamics, fueling dynamics, torque generation and crankshaft dynamics need to be captured. Among these dynamics, the intake manifold dynamics is the slowest, with a time scale of a few seconds (Moskwa, 1988; Cook and Powell, 1988; Hendricks et al., 1996). The fuel film dynamics involves time constants of the order of 0.25 - 0.5 seconds for a port injected engine, depending mostly on engine coolant temperature (Hendricks et al., 1996; Shayler et al., 1995).

The three-state mean-value engine model described by Cho (1987) and Cho and Hedrick (1989b) is a good example of such an engine model, and is used in this dissertation. In this model, air intake dynamics, fueling dynamics, and engine rotational dynamics are included. The mass of air in the intake manifold is obtained by using the conservation of mass principle together with expressions for air inflow and outflow rates. The mass flow rate of air entering the intake manifold is modeled as a product of the
maximum flow rate, throttle characteristics, and the normalized pressure influence. The mass flow rate of air entering the combustion chamber is the product of mass of air stored in the intake manifold, the volumetric efficiency, and engine speed. The fueling dynamics are modeled by a first order differential equation, with the effective fueling time constant accounting for the lag and delay involved with any fueling method. The engine rotational dynamics involve the engine indicated torque, the engine frictional torque, the accessory load, and the torque converter pump torque, along with inertial effects. The engine torque (indicated torque) production process is a discrete event but is modeled in the continuous domain approximately by invoking several factors such as an intake-to-torque production delay and a spark-to-torque production delay. Many of the parameters in the engine model are obtained empirically, limiting the generality of the engine model. Nevertheless, the three-state engine model described here captures the key dynamics important for automatic transmission-oriented powertrain research.

In addition to modeling the key engine dynamics, actively controlling the engine inputs is also important. Electronic-throttle-control (ETC) - also referred as a “Drive-by-Wire” system has seen greater application in recent years. The ETC control loop is commonly modeled as a first order lag with a time constant ranging from 25 to 100 ms, depending on specific ETC model. Lenz and Schroeder (1997) identify an ETC model with a time constant of 30 ms. For this dissertation, a first-order ETC model is employed for integrated powertrain neutral-idle control.

2.3.2 Modeling of torque converters

A typical torque converter has three parts: a pump, a turbine and a reactor as shown in Figure 2.3. The pump is driven by the engine output shaft and pumps fluid radially outward and into the turbine inlet by centrifugal action. The turbine absorbs the energy of the fluid and forces the fluid to flow radially inward. The reactor redirects the back flow from the turbine to a forward direction, and then discharges it to the pump inlet. This circulating flow is referred as the torus flow. There are two phases of torque converter operation. The first phase involves torque multiplication in the low (turbine)
speed range. When the turbine speed increases to a certain value, the reaction torque at
the reactor becomes zero and reverses direction, at which point the reactor starts
freewheeling. This is the second phase — the fluid coupling phase. Torque multiplication
is desired for vehicle launching. In addition, the torque converter acts as a vibration
damper as shown by Eksergian (1943), Ishihara and Emori (1966), and Tsangarides and
Tobler (1985). The engine torque pulsations are damped, so engine vibrations are
attenuated before being transmitted to the driveline. On the other hand, the engine is
isolated from abrupt changes in the driveshaft torque and road load.

Research on torque converters considers several aspects. One aspect is the fluid
field study of torque converters. One example of work is the study of pump, turbine and
stator fluid velocity fields using a laser velocimeter (Gruver et al., 1994; Brun et al.,
1994; Brun and Flack, 1995a, 1995b; Bahr et al., 1990). Such a fluid field study is
important for torque converter design and performance. However, for control oriented
research, our primary interest is on torque converters’ torque amplification and filtering
effects. Therefore, the emphasis of torque converter research for control studies is on
modeling the torque delivery behavior of torque converters.

Several types of converter models are found in the literature. One approach to
converter modeling is closely related to the converter design. This can be seen from
Jandasek (1961). He discusses design issues for torque converters and gives some
practical guidance on how to build an acceptable unit. He also gives very useful
information on the physical construction of torque converters, which is very helpful in
understanding important underlying geometric issues.

An early treatment of converter models is presented by Eksergian (1943). He uses
angular momentum conservation equations to derive relationships between torques and
the torus flow rate. The torus flow rate is then solved for from the compatibility equation
that is derived from the energy losses in the system. Two types of losses are considered.
The first one is the entrance shock loss between blade systems, and the second consists of
friction losses in the blades themselves. An important aspect of his model is the
discussion of fluid coupling behavior as a vibration damper. After deriving a linearized
small perturbation model of the damping torque due to fluid coupling as a function of the slip velocity, he demonstrates the damping effect of fluid couplings in a two-mass system coupler by a fluid damper.

Ishihara and Emori (1966) also illustrate a similar “vibration damper” concept. They first derive dynamic torque converter equations including the fluid inertial effects. The damping characteristics of the torque converter are then demonstrated by a linearized small perturbation model. They further propose a spring-damper system to represent the torque converter model, and to illustrate the “damper” concept. The authors also claim that the inertial effect of the fluid is negligible for relatively slow transient phenomena. The fluid inertial effect may be neglected if the frequency of the external disturbance is below one pulse per two impeller revolutions. If external disturbances are so slow, the working fluid in the torque converter responds to the change fast enough so that it may be considered to be always in equilibrium, and steady-state characteristics can be used for the analysis of transient phenomena. An important result of this work is that it points out that the inertial effect of the working fluid could be neglected under some conditions and the steady-state characteristics could be used for the computation of transient phenomena.

A widely used torque converter model is the Kotwicki (1982) model. Kotwicki uses momentum and energy equations to derive a model of a three-element, two-phase torque converter, but does not consider fluid inertial effects. Based on experimental data, a static regression model for the torque converter is developed and has been widely used for modeling and control of automatic transmissions (Runde, 1986; Cho, 1987; Cho and Hedrick, 1989; Tugcu et al., 1986; Pan and Moskwa, 1995). This static model can predict steady-state behavior of torque converters very well. The limitation of the model is that it may not be suitable for studying transient characteristics of the torque converter when there are sharp changes in the operating conditions such as those encountered during gear shifts. Kotwicki states that for most engine control work, the flow rate dynamics can be neglected, but for shift control and converter clutch control, these dynamics may not be negligible. Kotwicki’s model has been modified by Tugcu et al. (1986) heuristically to account for the effects of fluid inertia.
Whitfield et al. (1978) abandon the uniform velocity assumption implicit in the mean flow path analysis of Eksergian (1943) and propose a linear velocity distribution, with the velocity being zero at the mean radius and increasing linearly to the outer wall of the torus. This avoids the discontinuity in velocity at the mean radius that separates the inlet and outlet flows. The radial location about which the fluid is considered to rotate is solved for here by considering the continuity of flow rate. Then, by applying the momentum equation to a differential element, the torque component is derived and integrated along the radial direction. Shock loss, friction loss, and secondary circulating loss in a plane normal to the main through flow path are considered. The third flow loss is not considered in previous analyses by Eksergian (1943), Jandesek (1961), and Kotwicki (1982). Figure 2.8 and Figure 2.9 show that the linear velocity model gives about 6% error compared with the 10% error given by the uniform velocity model. The torque ratio versus speed ratio plot shows a near-perfect fit for the linear velocity model, while the uniform velocity model shows a 5% error. Tsangarides and Tobler (1985) report on a torque converter analysis in which model parameters are adjusted slightly from geometric values to improve the comparison of the model results with experimental data. Without parameter adjustment, model predictions fall within a 10 percent deviation of the data. However, insufficient detail is provided.

Hrovat and Tobler (1985) extend the results of Ishihara and Emori (1966) and develop a dynamic torque converter model that consists of four nonlinear first order differential equations. Fluid inertial effects are included in this model. Moreover, by using the bond graph modeling method, the authors develop interesting insights into the underlying dynamic structure. The resulting equivalent mechanical structure representing the dynamic model of the torque converter is shown in Figure 2.10.

The torque converter is represented by two inertias (pump and turbine inertias), two continuously variable transmissions (CVT), a clutch and a torsional spring. Validation of the static torque converter model showed good agreement with experimental data. The complete dynamic model is used in modeling the 1st-2nd upshift of an electronically controlled transmission, and also yields good results.
Figure 2.8 Plot of input torque versus speed ratio (Whitfield et al., 1978)

Figure 2.9 Plot of efficiency and torque ratio versus speed ratio (Whitfield et al., 1978)
Tsangarides and Tobler (1985) use this model in their study of the dynamic behavior of a torque converter with a centrifugal bypass clutch. Figure 2.11 shows the effectiveness of the torque converter as a vibration damper in that, when the converter clutch is not locked up, there is almost no transmissibility of the engine firing signals to the driveshaft torque during sudden increase of engine torque. Figure 2.11 (a) - (c) indicate that the firing signals show up in the driveshaft torque when the torque converter clutch is locked up. Figure 2.11 (e) shows the damping effect of the torque converter. It behaves like a first order system with a time constant about 80 ms. Figure 2.11 also indicates that the torque converter clutch changes the torque converter response. For clutch slip control system stability studies, the torque converter clutch characteristics need to be modeled. It is also needed for vibration studies when the torque converter clutch is locked up. However, the torque converter clutch is disengaged for all shift operations (Schwab, 1984). Therefore, it is not relevant for shift transient studies.

Transient characteristics of the torque converter are also analyzed by Fujita and Inukai (1990). The authors derive a torque converter model considering the fluid inertial
effects in both the torus and annular (plane of the torque converter) flows. The model is used to study the effect of the transient characteristics of torque converters on the acceleration performance of vehicles. The authors conclude that torque converter transient characteristics are important to the acceleration performance, as are the transient characteristics of the engine. Another important conclusion is that the inertial effects of the fluid in annular flow contribute very little to the overall transient performance of the torque converter. This implies that, when modeling the dynamic characteristics of torque converters, only the fluid inertial effects associated with the torus flow need to be considered.

Based on the above review, the following model enhancements are necessary for transmission modeling and control studies. First, a dynamic torque converter model with fluid inertial effects may play an important role in effectively modeling shift transients. Second, the dynamic converter model needs to be enhanced to represent the reverse flow situation. Ultimately, experimental validation is crucial to establish appropriate torque converter dynamic models.

2.3.3 Modeling of transmission hydraulic systems

The functions of a transmission hydraulic system are to generate the hydraulic pressure needed to perform a variety of operations, for example, initiating gear shifts as functions of vehicle operating conditions, and performing gear shifts satisfactorily by appropriately pressurizing and depressurizing friction elements (Rizzoni and Srinivasan, 1998). There are primarily two components of the transmission hydraulic system: the regulated hydraulic supply including the pump and associated regulation mechanisms, and the hydraulic load including clutch assemblies. This section briefly reviews the modeling of a hydraulic supply and a clutch.
One form of the regulated hydraulic supply is a pump with a pressure regulation circuit. A variable-displacement vane pump (VDVP) with a pivoting cam as shown in Figure 2.12 is modeled by Karmel (1988a). The analysis of the eccentricity dynamics corresponds to the analysis of the pump dynamics. This is because the eccentricity of the VDVP uniquely determines its flow capacity if the leakage effect is negligible, and the flow capacity determines the other operating variables for a given load. Therefore, the
author uses a single degree-of-freedom motion model of the eccentricity dynamics governed by the torques applied to the cam about its pivot. This model is called the "exact" model by the author, the "exact" model being simplified to get two "approximate" models for control analysis purposes.

The stability of the regulation circuit is discussed elsewhere by Karmel (1988b). The dynamics of the pressure-regulation circuit shown in Figure 2.13 are modeled. Hydraulic flow models include the output flow of the VDVP, the load flow, the valve-orifice flow and the regulation-chamber flow. Pressures are determined by the flow balance and the fluid compressibility. A stability analysis of the pressure-regulation circuit is done based on a linearized model.

Figure 2.12 Schematic of a variable-displacement vane pump (Karmel, 1988a)
Karmel (1986) continues the modeling work to include the shift dynamics where the clutch-accumulator assembly is also modeled. The lumped-parameter model developed assumes a compressible fluid with negligible inertia, turbulent flow through the valve orifices, rigid hydraulic lines and constant oil viscosity. Transmission dynamics under both fixed-ratio and shift conditions are simulated to study pressure regulation characteristics, effect of engine speed changes, changes in design parameters, and load variations during the release and engagement of clutches.

This hydraulic system model has been used by Tugcu et al. (1986) to study powertrain dynamics. In this model, the behavior of the VDVP is coupled to the behavior of the hydraulic network through the pressure-flow relationships of the shift valves. The hydraulic network modeled by Tugcu et al. is shown in Figure 2.13 and Figure 2.14.

One important phenomenon shown by Karmel through such simulations is the effect of air entrainment in hydraulic fluid on system stability. By changing the effective bulk modulus of the fluid, the transition from acceptable performance to unstable operation is abrupt, i.e. rather small changes in the effective bulk modulus can result in

![Diagram of pressure-regulated circuit of a VDVP in an automatic transmission](image)

Figure 2.13 A schematic of the pressure-regulated circuit of a VDVP in an automatic transmission (Karmel, 1988b)
system instability. Especially for the rotating clutches used in most automatic transmissions, air entrainment by the hydraulic fluid can be significant as a result of the high speed rotation of clutch plates. It is estimated that air entrainment can be as high as 30% in the worst case. As demonstrated by Karmel (1986) in Figure 2.15, air entrainment in hydraulic fluids can lower bulk modulus and result in system instability. If a high percentage of air is trapped inside the fluid, the fluid bulk modulus changes by a large amount and the stiffness of the fluid varies widely as a result. This makes it difficult to design controllers to control the engagement of clutches if the air entrainment and its consequences are ignored in the modeling, which is usually the case. Furthermore, if air entrainment varies from one shift to the next, shift quality will vary as well from shift to shift. This may be one key reason for shift-to-shift variations of shift quality.

The hydraulic systems investigated by Karmel are all purely mechanical and hydraulic systems. He pointed out that transient phenomena such as load variations
during shift, pressure regulation input due to throttle change, or speed changes do not necessarily impose additional regulation constraints for a system well calibrated for steady-state calibration. However, the studies did not show whether the regulation system is stable under load changes if the system deteriorates from the calibration. More robustness analysis studies can be helpful. In addition, electronic regulation may offer some opportunities for active regulation and improved performance.

Hasunaka et al. (1989) report a study on electro-hydraulic control of clutch pressure in an automatic transmission. The system under study is shown in Figure 2.16. An on-off solenoid valve is used to control clutch pressure. The flow is modeled as a one-dimensional compressible flow, and the model is a lumped-parameter type of model. One unique feature of this work is that the authors model the nonlinear clutch spring rate as shown in Figure 2.17. This type of spring is typical in automotive clutch springs. Figure 2.18 shows fairly good comparison between simulation and experimental results for the clutch engagement at a fluid temperature of 40°C. After the transition point, the clutch pressure rises to line pressure in about 10 ms. The authors also conclude that the flow resistance in this hydraulic system is dominated by the orifice flow at high temperature, and by the fluid viscosity at low temperature.

Kono et al. (1995) and Osawa et al., (1995) develop a dynamic model for a hydraulically powered torque converter clutch. The clutch piston dynamics are shown in Figure 2.19 together with other components of the clutch control system. The model structure in Figure 2.19, though reasonable, does not lead to good agreement with experimental data. The model relating the current command to the slip speed must be linearized to be used in controller design. As the original model itself lacks accuracy, the authors develop instead an ARMAX (Auto-Regressive Moving Average Exogenous) model by using model identification techniques (Kono et al., 1995; Osawa et al., 1995). The identification results are shown in Figure 2.20 and show good comparison for small signal variations of the command.
Figure 2.15 Hydraulic-pressure regulation under fixed-ratio conditions - effect of fluid compressibility (Karmel, 1986)

Figure 2.16 Experimental circuit used for clutch pressure studies (Hasunaka et al., 1989)
Figure 2.17 Clutch model (Hasunaka et al., 1989)
(a) Clutch model (b) Clutch pressure versus clutch stroke characteristics

Figure 2.18 Simulation and experimental results clutch engagement (Hasunaka et al., 1989)
(a) Simulation results for clutch engagement
(b) Experimental results for clutch engagement
2.3.4 Clutch/band friction characteristics for different types of clutches

Clutch/band capacity is commonly represented as a function of the hydraulic pressure and clutch geometry, and involves a friction coefficient which is a nonlinear function of slip speed as shown in Figure 2.21 (Tugcu et al., 1986). Clutch capacity models also usually assume that the torque buildup in the clutches is instantaneous, i.e., the torque transmitted is proportional to the axial force across the clutch plates and to the coefficient of friction at the contact surfaces. The friction coefficient $\mu$ is given by the $\mu$-
\( \omega_{\text{slip}} \) curve, data being usually available at high slip speeds. In Figure 2.21, when the slip speed approaches zero, the friction coefficient is assumed to follow the solid curve shown in the figure instead of the dashed curve and a much higher value of the static friction coefficient. The simplification described here has been made by Tugcu et al. for the purpose of improving the stability of the computations required for simulation, but is not essential.

Friction characteristics of clutch material are closely related to the system stability as demonstrated by Tsangarides and Tobler (1985). Figure 2.22 shows the schematic of a torque converter with a centrifugal bypass clutch described by Tsangarides and Tobler (1985) and used for stability analysis. The bypass clutch torque capacity is shown in Figure 2.23. Simulation results demonstrate that self-excited vibrations may result from the frictional behavior of the clutch during clutch slip. These self-excited driveshaft oscillations are called “shudder”. According to the authors, the existence of a negative slope in the clutch torque versus slip curves as shown in Figure 2.23 and Figure 2.24 is the main cause of clutch shudder. A similar situation called “chatter” is also true for the clutches involved in gear shifts. Figure 2.25 shows the effects of different slopes of friction characteristics on shudder. With curve A, there is sustainable shudder. Curve B results in a marginally sustainable shudder. Note that curve C, with small negative slope, does not lead to sustainable shudder.

Friction characteristics are dependent on the material as well as lubricant and aging effects. Some examples of friction characteristics from the literature are shown in Figure 2.26 and Figure 2.27. Figure 2.26 shows aging effects of clutch friction characteristics. Figure 2.27 shows the friction characteristics of Borg-Warner clutches. When modeling and designing controllers for automatic transmission systems, the effect of the friction characteristics should be considered in robustness studies.
Figure 2.21 Coefficient of friction versus slip speed for wet clutches (Tugcu et al., 1986)

Figure 2.22 Schematic of a torque converter with centrifugal bypass clutch (Tsangarides and Tobler, 1985)
Figure 2.23 Typical bypass clutch capacity as a function of clutch slip and engaging pressure (Tsangarides and Tobler, 1985)

Figure 2.24 Typical bypass clutch torque capacity vs. clutch slip with negative damping (Tsangarides and Tobler, 1985)
Figure 2.25 The effect of the slope of the clutch torque capacity ($T_c$) vs. clutch slip ($W_{slip}$) on shudder (Tsangarides and Tobler, 1985)

Figure 2.26 Friction characteristics of new and aged clutch materials (Ward et al., 1994)
2.3.5 Modeling of transmission mechanical systems

Traditional methods of analyzing the dynamics of planetary gear sets include methods from classical mechanics such as use of Newton's second law of motion, Lagrange's equation etc. Such methods are widely used (Ishihara and Inui, 1970; Shindo et al., 1979; Hojo et al., 1992; Jeong and Lee, 1994; Pan and Moskwa, 1995).

A graphical tool called the "Lever Analogy" is used by Benford and Leising (1981) to represent planetary gear train kinematics. Relationships between the angular velocities of planetary gear train members are displayed with reference to a lever, as are relationships between torques transmitted by these members. Since its introduction, the lever analogy has been a commonly used in industry for analyzing the kinematic as well as dynamic behavior of transmissions (Tugcu et al., 1986; McKenny et al., 1993).
Taking the simple planetary gear set shown in Figure 2.28 as an example, the lever representation for this simple planetary gear set is shown in Figure 2.28b. Note that each lever segment length is proportional to the number of gear teeth marked on the segment. \( n_s \) represents the number of teeth in the sun gear, and \( n_r \) represents the number of teeth in the ring gear. \( K \) is a suitable scaling constant. The points labeled S, C and R on the lever are points of application of forces analogous to the sun, planet carrier and sun gear torques respectively. Figure 2.28c shows one of the possible configurations of this planetary gear set. The corresponding speeds are noted also. Figure 2.28d shows the corresponding speed lever diagram. Figure 2.28e shows the torques applied on the gear set and Figure 2.28f shows the corresponding torque lever diagram.

Another method used for graphical representation of transmissions is the “bond graph” method. This method can model dynamic relationships for complex systems with several forms of energy storage (Karnopp et al., 1990). The use of bond graphs in transmission related research is not widespread but is increasing (Cho, 1987; Runde, 1986; Hrovat and Tobler, 1985, 1991). Figure 2.29 shows a bond graph representation of the planetary gear set in Figure 2.28.

We should notice that the model of transmission mechanical systems used for powertrain control is simpler than those used for vibration studies. This is due to the fact that the primary interest in modeling mechanical systems for powertrain control is to identify low frequency aspects of system dynamic behavior.

2.3.6 Modeling of vehicle dynamics

Vehicle dynamics is a subject of research in its own right. However, for the purpose of powertrain research, a focus on longitudinal vehicle dynamics is a good starting point. The simplest type of longitudinal vehicle dynamics model is a gross vehicle dynamics model. In this case, the driveline is modeled as a torsional spring connected to a vehicle that is modeled as a lumped inertia. Such a model ignores details such as wheel inertias and tire-road interactions. The load torque due to wind and rolling resistance is modeled as a nonlinear function of vehicle speed (Heywood, 1988).
Figure 2.28 An example of lever analogy

\[ T_C = T_I + T_S \]
\[ T_S = n_S \]
\[ T_C = n_S \]

Figure 2.29 Bond graph representation of the planetary gear set shown in Figure 2.28

\[ R_S = \frac{n_S}{n_S + n_R} \]
\[ R_R = \frac{n_R}{n_S + n_R} \]
In studying longitudinal vehicle dynamics, one important aspect is that of tire-road interaction. The details of how the tire interacts with the road directly determine how the vehicle performs, since the primary control and disturbance forces applied to the vehicle, with the exception of aerodynamic forces, are generated in the tire-road contact patch (Gillespie, 1992). From a system dynamics point of view, the tire is one of the two key sources of damping in the system, the other one being the torque converter. Therefore, tire-road interaction can become important when studying dynamic phenomena such as shuffle-mode oscillations (Hrovat and Tobler, 1991). Vehicle dynamic models including tire-road interactions have been used by Tugcu et al. (1986), Cho (1987), Cho and Hedrick (1989), and Hrovat and Tobler (1991). The models used in these works can be traced back to Wong (1978), with slight differences in defining tire slip. Another general reference on this topic is Gillespie's book (1992). The following review on longitudinal tire-road interaction follows the definitions given by Wong (1978).

Figure 2.30 (a) and (b) show the behavior of a tire under a driving torque (tractive) and a braking torque respectively. When a driving torque is applied, the tread elements are compressed before entering the contact region. The distance that the tire travels is less than that in free rolling. The slip of the tire under a driving torque is defined by

\[
i = \left(1 - \frac{V}{r \omega}\right) \times 100\% = \left(1 - \frac{r_c}{r}\right) \times 100\% \tag{2.1}\]

where \(V\) is the longitudinal speed of the tire center, \(\omega\) is angular speed of the tire, \(r\) is the rolling radius of the free-rolling tire, and \(r_c\) is the effective rolling radius of the tire defined as the ratio of the longitudinal speed of the tire center over the angular speed of the tire. When a driving torque is applied, \(\omega r > V\).

A typical curve of tractive effort coefficient versus longitudinal slip for a tire is shown in Figure 2.31. The maximum tractive force of a pneumatic tire on hard surfaces is usually reached between 15 - 20% of slip. Further increase of slip beyond that results in an unstable condition, for example, tire spin. The tractive effort coefficient, i.e. the ratio
of the tractive force to the vertical load of a tire, falls from the peak value $\mu_p$ to the pure sliding value $\mu_s$.

When a braking torque is applied to the tire, the tread elements are stretched prior to entering the contact area as shown in Figure 2.31(b). The distance that the tire travels under a braking torque is greater than that in free rolling. The skid of the tire $i_s$ is used to measure the severity of braking,

$$i_s = \left(1 - \frac{r \omega}{V}\right) \times 100\% = \left(1 - \frac{r}{r_c}\right) \times 100\%$$  \hspace{1cm} (2.2)

The relationship between the braking effort coefficient defined as the ratio of the braking force to the normal load of the tire, and skid has similar characteristics to those between the tractive effort coefficient and slip. Variation of the braking effort coefficient with skid of a tire on various surfaces is shown in Figure 2.32.

![Figure 2.30](image)

Figure 2.30 (a) Behavior of a tire under a driving torque (b) Behavior of a tire under a braking torque (Wong, 1978)
Figure 2.31 Variation of tractive effort coefficient with longitudinal slip of a tire
(Wong, 1978)

![Tractive Effort Coefficient Graph]

Figure 2.32 Variation of braking effort coefficient with skid of a tire on various surfaces
(Wong, 1978)

![Braking Effort Coefficient Graph]

In addition to the longitudinal vehicle dynamics with tire-road interaction discussed above, vehicle dynamics models can include additional degrees of freedom of motion. Shindo et al. (1979) describe a model of shift response taking into consideration the vehicle vibration system, as shown in Figure 2.33. The vehicle dynamics model in this case includes fore-and-aft, up-and-down and rotating motions that are independent of one
Vehicle vibration effects on shift dynamics are studied. The authors indicate that vehicle vibrations significantly affect shift characteristics. Proper selections of the following stiffness and damping coefficients are especially important for the shift characteristics: (1) the torsional stiffness of the rear axle in a rear-wheel drive vehicle \( (k_e) \) in Figure 2.33; (2) the fore-and-aft stiffness \( (k_{10}) \) and coefficient of viscous damping \( (C_{10}) \) of engine mounts, and (3) the torsional stiffness \( (k_w) \) and coefficient of viscous damping \( (C_w) \) for the rear axle case. However, for a front-wheel-drive vehicle, the dominant factors may change. In addition, the engine dynamics are not considered in this model. It is hard to extrapolate from this study what the dominant factors would be when the whole system, including key engine dynamics, is considered.

This study suggests that vehicle dynamics with more than just longitudinal dynamics should be considered when doing a complete study on shift quality. The effect of clutch shudder/chatter on driveline vibrations can also be considered within the same framework.

Figure 2.33 Automatic transmission dynamic model (Shindo et al., 1979)
2.3.7 Integrated powertrain modeling

In the previous sections, the modeling of key individual components of powertrain systems is discussed. In this section, the focus is on integrated powertrain modeling. Integrated powertrain modeling is important in studies of overall powertrain dynamic behavior and in the design of powertrain control strategies. It is essential in studying problems such as shift transient dynamics, torque converter clutch control, shuffle mode oscillations, integrated engine and transmission control, etc.

There are three types of integrated powertrain models in the literature in terms of the details of engine dynamics included. The first type focuses on automatic transmission modeling, and assumes engine torque as input without including details of engine modeling except the engine crankshaft inertial effect. Examples of this type are Ishihara and Inui (1970), Shindo et al. (1979), Jeong and Lee (1994). These models are used to study shift transients, the objectives being identification of the sources of shift shock disturbances and solutions for suppressing these shocks. Ishihara and Inui (1970) derive shift transient dynamic equations and examine the effect of the variations of different parameters and inputs on shift characteristics. The authors quantify shift characteristics in terms of the duration of the inertia phase of a shift, the time rate of change of the output shaft torque, and the difference between maximum and minimum output shaft torques. The model is validated by bench tests. However, there is no consideration of tire-road interaction in the model. Shindo et al.’s work (1979) is briefly discussed in the previous section. One unique feature of their work is the multi-degree of freedom vehicle dynamics model shown in Figure 2.33. Comparison of the transmission output torque from simulation and vehicle tests is shown in Figure 2.34. The unique feature of Jeong and Lee’s work (1994) is the incorporation of dynamic torque converter model in the powertrain model. However, the modeling results do not match vehicle test results.
The second type of powertrain model incorporates engine torque - engine speed maps with transmission models. This approach takes into account the effect of changing engine operating conditions and is used in control approaches reported in production vehicles, as in Kono et al. (1995) and Hrovat and Tobler (1991). In addition, Kono et al. (1995) use a static torque converter model, but include some details of hydraulic control dynamics and clutch piston dynamics. Hrovat and Tobler (1991) focus on investigating "shuffle mode" oscillations, which occur in gearshifts at low gears in manual transmissions, and in automatic transmissions with the torque converter clutch in the lock-up mode. Their work is fairly complete in modeling automotive powertrains. Using the bond graph modeling approach, they include the shaft compliance effect, tire-road interaction, and an electronic-controlled clutch dynamic model. Tsangarides and Tobler's work (1985) on torque converter clutch control uses a similar approach in incorporating an engine torque map. The integrated powertrain model used by Tugcu et al. (1986) is another fairly complete model which includes an engine model in the form of an engine map, the dynamics of a shift hydraulic system with the VDVP pump, pressure regulation circuit dynamics, hydraulic network model, simplified clutch and accumulator dynamics, a torque converter model, transmission mechanical system model, and a vehicle dynamics model including details of tire-road interaction. This model is useful in studying shift transients taking the entire powertrain model into account.
The third type of powertrain model integrates engine dynamics with transmission dynamics. Taniguchi et al. (1991) model engine torque control as a first order lag, which approximately takes account of the engine manifold dynamics. However, the authors do not give adequate detail. The transmission model does not include tire road interaction. The powertrain model is used to design a feedback shift controller. However, the reference does not give adequate detail. Pan and Moskwa (1995) report a detailed transmission dynamics model. They mention that an engine model is used in the simulation, again without any details.

The powertrain model used by Cho and Hedrick (1989) is another example of a complete powertrain model with a three-state engine model, a static torque converter model, a transmission mechanical system model, and a vehicle dynamics model with tire-road interaction. This model is also useful in studying shift transients. The engine model, in particular, captures the key dynamics of the engine and can play an important role in helping develop integrated engine-transmission control. However, the shift hydraulic system dynamics are not modeled. The clutch pressure trace is represented by a predetermined exponential curve that is unrealistic for a study of shift transients. The delays encountered in the filling of clutch cavities are important for an overall dynamic performance study of the powertrain and are omitted.

In summary, the integrated powertrains models seen in the literature incorporate varying levels of description of relevant phenomena. Given our interest in integrated engine and transmission modeling and control, we need to model both engine and transmission/vehicle dynamics with greater fidelity.

2.4 Review of control issues for automatic transmissions

The introduction of automatic transmissions has made driving easier and safer for customers in the last few decades. However, this type of transmission has also caused some problems. For example, reduction in fuel economy as compared to manual transmissions and additional control problems involved in automating shifts are a few of the problems encountered.
The objectives of automatic transmission control are to perform various transmission related control functions in an optimal way, so as to achieve satisfactory gear shifts with reduced shift shock, and to improve the efficiency of automatic transmissions and the overall fuel economy. Electronically controlled transmissions, which have become more common in recent years, enable integration of the control of the engine and the transmission to achieve optimal performance of the powertrain (Schwab, 1984; Hiramatsu et al., 1986; Shinohara et al., 1989; Yasushi, 1991; Usuki et al., 1996).

2.4.1 Shift control

Shift shock is a phenomenon associated with shifting of manual and automatic transmissions. A good driver shifts smoothly by coordinating the actuation of the acceleration pedal and clutch when driving a manual transmission equipped vehicle. However, this requires experience and is inconvenient for many drivers in urban driving. Frequent gearshifts in such cases involve frequent engaging and disengaging of the clutch. While an automatic transmission equipped vehicle will handle shifts automatically, the design and control of a transmission that shifts smoothly is not an easy task.

Due to the discrete gear ratios of automatic transmissions, engine speed and engine and driveshaft torques undergo large changes during shifts, and hence ensuring smooth transitions is a demanding task. Two phases can be identified in each shift – the torque phase and the inertia phase. In the torque phase, torque transfer between the two friction elements involved in the shift takes place. For example, in the automatic transmission shown in Figure 2.2, clutches C1 and C2 are the two friction elements involved in the 1st - 2nd gearshift. During the torque phase in a 1st - 2nd gear upshift, the oncoming clutch C2 takes up more and more of the torque from the offgoing clutch C1, C1 still being in the locked-up condition. The engine speed changes little during this phase, the primary action being one of torque transfer from the offgoing friction element C1 to the oncoming friction element C2. During a downshift, the torque phase follows the inertia phase.
The inertia phase is the phase during which the engine speed adjustment takes place. The inertia phase follows the torque phase during an upshift. Referring to the same example of the 1st-2nd gear upshift described above, during this phase, torque at the oncoming friction element C2 reduces the slip speed of the C2 clutch until it is reduced to zero and lockup occurs. During a downshift, the inertia phase precedes the torque phase.

The harshness of the lockup depends on the change in the torque transmitted by the oncoming friction element. The larger is this change, the more abrupt is the transition. The magnitude of this change depends on the clutch/band pressure and friction characteristics at low slip and the torque requirement in the oncoming gear. Lower clutch/band pressures would reduce the magnitude of the change in torque at lockup, but would increase the inertia phase duration.

A number of control techniques have been used to improve shift quality. These approaches are in the form of either open loop or closed-loop control. One feature of the shift control problem is that the control objective is normally only indirectly related to the controlled variable. However, regardless of the approach, the manipulated inputs are engine variables such as throttle angle, spark advance and transmission variables such as hydraulic pressure. If engine variables are manipulated, we can characterize the shift control approach as an integrated powertrain control approach, discussion of this approach being deferred to a later section titled "integrated powertrain control during shifts". Due to the important role that hydraulic pressure plays in transmission control, we will first take a look at how hydraulic pressure (line pressure, clutch/band pressure) is manipulated. Following that, we will review other techniques used in shift control.

2.4.1.1 Pressure manipulation

Hydraulic pressure provides the primary "power" that drives the engaging and disengaging of friction elements like clutches and bands. The most important hydraulic pressure is line pressure, which is the primary hydraulic pressure in an automatic transmission and influences other pressures such as clutch/band pressures. It can play an important role in suppressing shift shock. Almost all automatic transmissions produced
nowadays incorporate a form of line pressure control, either mechanical or electronic (Hiramatsu et al., 1986; Shinohara et al., 1989; Yasushi, 1991; Hojo et al., 1992). Experimental and theoretical studies have been reported showing that adjusting the line pressure during a shift can reduce shift shock. For example, at the end of a gearshift, the \( \mu - \omega_{\text{slip}} \) characteristics of the oncoming clutch often give rise to a torque peak. Such torque peaks can be reduced by lowering the line pressure at the initiation and conclusion of gearshifts as shown in Figure 2.35 (Shinohara et al., 1989).

Shinohara et al. (1989) describes line pressure control in a Nissan automatic transmission. Yoshimura and Takuji (1992) from Mazda Motor Corporation also reports similar approach to adjusting line pressure. The only difference with the latter is that the frequency of the control signal to the variable duty solenoid valve is changed based on the throttle sensor and turbine speed sensor signals, so that the line pressure maintains a desired value. We will review Shinohara et al.'s approach (1989) as an example.

In Shinohara et al.'s work (1989), a variable capacity vane pump is used, and a variable duty solenoid is employed to achieve electronic control over the line pressure. The control computer sends a signal to actuate a hydraulic solenoid driven by pulse width modulation and causes a pressure related to throttle opening, the throttle pressure, to be generated. The throttle pressure serves as the signal pressure that actuates the regulator valve which modulates the line pressure as shown in Figure 2.36. The computer has a precalibrated duty ratio map so that the regulator valve adjusts the line pressure level according to the throttle opening of the engine, as well as the optimum line pressure for different road gradients. A map indicating the optimum line pressure relative to each throttle opening and shift is shown in Figure 2.37. Figure 2.38 shows the simulated acceleration behavior of the vehicle for a power-on upshift, and indicates the tradeoff between shift shock and shift duration as line pressure is changed at a given throttle setting.

In this approach, pressure control input, i.e. the solenoid duty ratio is dependent on calibration. The mechanical structure of the pressure regulation valve determines the dynamics of the pressure regulation. However, there is no electronic feedback to adjust
the control input. In addition, there is no quantitative analysis relating the solenoid duty ratio to the line pressure output or any information on the time constants achievable. It is hard to evaluate the applicability of this approach to other transmissions. Furthermore, the nature of this pressure regulation is open loop, which has generic drawbacks such as lack of robustness to disturbances caused by change of operating conditions. For example, shift quality changes when the torque characteristics of the engine and the friction characteristics of clutches/bands vary with time due to normal wear or due to build-to-build variations. Shift quality can also change due to changes in engine torque caused by air density differences with change of temperature and/or altitude. Therefore, closed loop control of hydraulic pressure is more desirable. Shinohara et al. (1989) employ adaptive learning control of hydraulic line pressure to overcome some of these problems.

Figure 2.35 Line pressure control during 1\textsuperscript{st} to 2\textsuperscript{nd} shift (Shinohara et al., 1989)
Figure 2.36 Line pressure control system (Shinohara et al., 1989)

Figure 2.37 Line pressure for each shift (Shinohara et al., 1989)
In addition to controlling line pressure in the overall system, clutch pressure control is done by combining electronic line pressure control together with accumulator back pressure control. A Variable duty solenoid valve is used to vary the back pressure of the accumulator, which makes it possible to modulate the pressure applied to the friction element during shifting. An example of this approach is reported by Yasushi (1991). However, with an accumulator in the system, the system can not respond to commands fast enough. In applications requiring direct manipulation of the clutch pressure such as the clutch-to-clutch control to be discussed later, this configuration is not acceptable.

2.4.1.2 Inertia phase control

If there is a large torque change at the end of the inertia phase in a shift, shift quality is degraded. There are several approaches in the literature dealing with closed-loop control of the inertia phase in order to improve shift quality and consistency of shifts. To summarize, reported forms of inertia phase control include inertia phase

Figure 2.38 Acceleration map during power-on upshift (Shinohara et al., 1989)
duration control, turbine speed trajectory control, and speed ratio control. Inertia phase control is also part of clutch-to-clutch shift control, which will be discussed in the following section.

It is reported that there is a close correlation between shift quality and the duration of the inertia phase (Narita, 1991; Shinohara et al., 1989). Usually, shift shock is smaller with longer inertia phase. If shift shock is suppressed at the expense of inertia phase duration, the longer time required to complete a gearshift might deteriorate the friction elements. On the other hand, trying to shorten the shift time results in a larger shift shock that degrades shift quality.

A feedback control system is proposed by Narita (1991) to optimize shift duration and torque waveforms for Nissan’s RE4R01A automatic transmissions. First, experiments are conducted to establish the relationship between “Inertia Phase Duration” and “Line Pressure Duty Ratio” together with the torque waveforms. Experiments are also conducted at different altitudes and the results show that “Inertia Phase Duration” is a good index of shift quality. In the normal operating range, the higher is the line pressure duty ratio, the shorter is the inertia phase duration. However, shift shock becomes excessively high at higher values of the line pressure duty ratio. Based on the experiments, “Total Shift Time”, which is defined as the interval from the initiation of the shift command to the completion of the inertia phase, can help choose the correct line pressure duty ratio corresponding to the optimum inertia phase duration. The optimum inertia phase duration is also adjusted, based on throttle valve opening, to match engine torque characteristics. Closed-loop correction of inertia phase duration is done by repeated gearshifts. Though this method works well with a given transmission design, it is heavily dependent on empirical data. The controller design is also based on experimental tuning, which requires much labor and time. On the other hand, one tuned data table may not be the best for other types of transmissions. Despite these limitations of the proposed method, it does offer one way of implementing closed-loop shift control based on experimental data. It also indicates the different factors affecting shift quality based on empirical evidence.
Turbine speed closed-loop control is reported by several researchers. The objective of turbine speed control is to adjust the clutch pressure to achieve a turbine speed trajectory, which will result in a smooth shift. One example of turbine speed closed-loop control is presented by Kondo et al. (1990) and Taniguchi and Ando (1991) from Toyota Motor Corporation. The clutch pressure control system is shown in Figure 2.39. In their work, clutch pressure control has three stages. As shown in Figure 2.40, the first stage is "Basic Control" during which the optimum accumulator back pressure is set in an open loop fashion according to the throttle opening for a specified shift. By applying learning control, the initial hydraulic pressure level is adjusted from shift to shift so that shift time may be optimized. The second stage is "Feedback Control". The accumulator back pressure is continually adjusted by adjustment of the clutch pressure, to cause the actual rotational speed of the input shaft, or turbine speed, to follow the target trajectory closely. A PID controller is used for this control (Taniguchi and Ando, 1991). The third
stage is "Completion Control" which is also open loop and involves lowering the clutch pressure temporarily so that the output torque change at lockup is reduced. Also, if engine torque level is lowered during the shift, it is allowed to rise at the end of the shift. Excessive torque levels can lead to deterioration of friction materials and the automatic transmission fluid (ATF). The first and third stages above are open-loop control schemes and are based on many calibration experiments. Only the second stage uses closed-loop control. Such a turbine speed closed-loop control scheme is used in Toyota's A350E transmissions (Hojo et al., 1992).

An example of closed-loop control of speed ratio during the inertia phase is described by Hrovat and Powers (1988) from Ford Motor Company. A powertrain model including an engine, torque converter, clutch hydraulic system and driveline system is derived. The torque converter model has four states: pump speed, turbine speed, reactor
speed, and torus flow. The turbine torque is assumed to be a static, linear function of clutch pressure only. The clutch actuator transfer function between the clutch pressure and duty cycle is determined experimentally using a spectral analyzer and results in a second-order dynamic model. The torque converter model is the only nonlinear subsystem model in the system, and it is linearized. The closed-loop controller design is based on the linearized model and is subsequently verified by nonlinear simulation of the powertrain and by experiments. The speed ratio is the control target. In the inertia phase, the oncoming clutch pressure is controlled via duty-cycle variations of a PWM solenoid. The shift controller block diagram is shown in Figure 2.41. \( G_{c1} \) is a PID controller and \( G_{c2} \) is a lead-lag compensator used for fine-tuning of the closed-loop control system. The filter \( G_{f1} \) reduces measurement signal noise, and the filter \( G_{f2} \) shapes the commanded signal. The inner loop is for hydraulic pressure estimation and pole-placement control. For the “classical” controller, this loop is not used and is bypassed by setting the gain \( K_u = 0 \). The experimental and simulation results for a 1st - 2nd power-on upshift are shown in Figure 2.42. However, no information on the bandwidth of the speed control loop is given.

Jeong and Lee (1994) report a shift controller design by using MIMO LQG/LTR theory. The powertrain model derived includes a dynamic torque converter model, engine crankshaft dynamics, gear train dynamics and gross vehicle dynamics. The model is then linearized. In the inertia phase, a desired turbine speed trajectory is given. However, the inputs to the system are torques such as engine torque, clutch and band torques. This assumption simplifies the problem to a great extent and ignores the dynamics involved with engine torque generation, the nonlinear effect of clutch/band pressure manipulation, and actuator dynamics. These ignored effects may be dominant in the system response, and simply ignoring them without detailed examination of the dynamics involved greatly limit the feasibility of the proposed approach.
Figure 2.41 Shift controller block diagram (Hrovat and Powers, 1988)

Figure 2.42 Experimental and simulation results for 1st-2nd power-on upshift (Hrovat and Powers, 1988)
2.4.1.3 Clutch-to-clutch shift control

Clutch-to-clutch shift control involves active control of two clutches to achieve smooth load transfer during torque phase. The inertia phase control involved in clutch-to-clutch shift control is essentially the same as that described above, i.e. to control the oncoming clutch pressure in a certain way to achieve the desired control objective.

A one-way clutch to clutch shift makes the shift control problem easier. The load transfer between the one-way clutch and clutch occurs smoothly due to the nature of operation of the one-way clutch. In this process, only the oncoming clutch needs to be controlled. On the other hand, clutch-to-clutch shift control requires good coordination between the offgoing and the oncoming clutches, which is not an easy task. Timing relationships between the disengaging and engaging clutch pressures are critical in clutch-to-clutch shift systems, since any mismatch in capacity between the releasing and applying friction elements will result in either runaway (excessive engine speeds) or increased torque loss (Leising et al., 1989). One advantage of a clutch-to-clutch shift is that the mechanically complex one-way clutch can be eliminated. Usually lighter weight, simpler, and less costly gear trains can be achieved with this type of configuration as compared to transmissions with one-way clutches (Shindo et al., 1979; Leising et al., 1990b).

Early implementations of clutch-to-clutch controlled timing relationships between hydraulic pressures by hydraulic timing valves before electronically controlled transmissions were built (Numazawa et al., 1978; Shindo et al., 1979). After electronically controlled transmissions were introduced, solenoids were used to control the clutch pressures and the timing control was done electronically.

Chrysler's Ultradrive Automatic Transaxle with all clutch-to-clutch shifts has been reported in several papers (Martin and Nogle, 1989; Leising et al., 1989; Leising et al., 1990b; Martin and Redinger, 1993). The schematic of this automatic transmission is shown in Figure 2.43. There are two friction elements involved in each shift. For example, during 1\textsuperscript{st} - 2\textsuperscript{nd} gearshift, clutches "LR" and "24" are involved.
The authors explained the development history of their clutch-to-clutch control scheme. During the torque phase, the release time of the off-going clutch and the fill time of the on-coming clutch are learned values. A bang-bang type of control is used for early implementations of shift control systems (Leising et al., 1990b). This method is in fact used in inertia phase control, not torque phase control. Bang-bang control is used as follows. The turbine speed acceleration obtained from speed measurements is compared with the desired value. Depending on the sign of the acceleration error, the clutch is controlled to apply or vent. Bang-bang control has no fixed frequency. Its operating frequency depends primarily on the accuracy and rate of speed measurement and on the solenoid response time.

In the initial development phase, accumulators were used to reduce the amplitude of torque variations during shifts and to reduce the bang-bang control frequency. The authors do mention that accumulators limit the system ability to respond to large torque changes, and they decided to resolve the tradeoff as well as they could. There is no theoretical analysis reported in this study. In later applications, they encountered a vibration problem when applying the adaptive controls developed on one powertrain to another powertrain. To overcome the vibration problems, proportional control is used for
closed loop control of turbine acceleration in the inertia phase, but that is not enough. They point out that the use of accumulators makes the response slower since the proportional control results in a change in the duty cycle of the solenoid and in the flow rate, but not the clutch pressures. There is a lag before the desired clutch pressure is reached. High gain control would increase the response speed but tends to cause instability in the system. To compensate for this, a brief duration of high-gain control is used together with low gain control. In the whole design process, the controller gain is determined mainly by calibration. There is no theoretical analysis of the dynamic behavior of the system, or on how the controller design should be carried out. This may explain why this approach has not been widely accepted.

Figure 2.44 shows the simulation results for a $1^{\text{st}} - 2^{\text{nd}}$ upshift and a $4^{\text{th}} - 3^{\text{rd}}$ kickdown. During the $1^{\text{st}} - 2^{\text{nd}}$ upshift, the offgoing element is allowed to have a slight negative slip to ensure that it does not have excess capacity like an overrunning clutch. This feature is also used in the GM clutch-to-clutch shift approach to be discussed later. The oncoming element is filled at a controlled rate until the speed change starts. The authors indicate that the release time of the offgoing clutch and the fill time of the oncoming clutch are learned values to guarantee that the release and apply of the two clutches occur simultaneously. The rate of application of the oncoming clutch is also a learned value to accommodate the variations in solenoid performance and hydraulic circuit leakage. When the oncoming clutch develops sufficient capacity to take over the torque from the offgoing clutch, the inertia phase starts. The pressure in the oncoming element is controlled to maintain the desired rate of speed change until the shift is complete. The desired rate is reduced near the completion of the shift to reduce the shock at clutch lockup. During the $4^{\text{th}} - 3^{\text{rd}}$ kickdown shown in the same figure, there are three control features. The first is to "vent the offgoing clutch until slip occurs". The second is to control the rate of the input speed change by controlling the offgoing clutch pressure. When the input speed reaches the target speed, the offgoing clutch is reapplied to maintain that speed until the oncoming clutch is filled, which is called "hold speed" control. Though the proposed approach uses the adaptive control concept, it is heavily
dependent on experimental data. In addition, the existence of an accumulator in the system is disadvantage. Furthermore, no theoretical analysis on the system dynamics and controller design is given. It is not a systematic approach that could be applied to any type of clutch-to-clutch shift controls.

Another clutch-to-clutch control technique has been proposed and incorporated in some General Motors transmissions (Butts and Hebbale, 1992; Butts et al., 1991; Hebbale and Kao, 1995). Figure 2.45 and Figure 2.46 show the power-on upshift and down shift base control strategies. As shown in Figure 2.45, the upshift starts with the fill command (between t₀ and t₁) for the oncoming clutch. The fill time and fill pressure are

Figure 2.44 Computer simulated adaptive control (Leising et al., 1990b)
predetermined as a function of clutch speed and transmission oil pressure. The offgoing clutch starts depressurizing from $P_{ofig}$ to $P_{ofig}$. On completion of the fill phase, the oncoming clutch pressure command is reduced from fill pressure to a start pressure $P_{oncs}$. Then the oncoming clutch pressure is ramped up to a target pressure $P_{ontc}$ (the pressure at which the oncoming clutch should be able to carry all of the input torque) in a predetermined interval. At time $t_2$, the offgoing clutch pressure command is stepped down to an intermediate value and then ramped down even further at a predetermined rate. At time $t_3$, the output acceleration, which is estimated from shaft speed measurements using a Kalman filter (Hebbale and Ghoneim, 1991), approaches the target value which is the ideal value of the acceleration at the bottom of the torque phase. The offgoing clutch is fully released at this point. As the oncoming clutch pressure continues to increase, a negative slip occurs at the offgoing clutch shortly. At this point, time $t_4$, closed-loop control of the oncoming clutch is initiated. The oncoming clutch slip is closed-loop controlled to follow a desired time-based slip trajectory. When the oncoming clutch slip is less than a predetermined small amount, the oncoming clutch pressure is ramped to line pressure ($t_5$) and the shift is complete (Hebbale and Kao, 1995). Power-on down shift base control is shown in Figure 2.46 and can be explained using similar reasoning.

To compensate for the effects of life cycle and build-to-build variations on transmission shift quality, the base shift controls are augmented with adaptive controls. Hebbale and Kao (1995) propose a model reference adaptive control approach with parameter adjustments being made using the gradient method. The proposed method relies on shaft acceleration information and timing of the application of the clutch pressures which can be handled by the base control strategies. The adaptive control for the upshift adapts the oncoming clutch fill time and fill pressure. For the downshift, only the oncoming clutch pressure is adapted. Vehicle test results of these adaptations are shown in Figure 2.47 and Figure 2.48. In upshift case, the adaptation brings the system to desired (predetermined) shift quality in 6 shifts. In the downshift case, the adaptation succeeds in 3 shifts. These results show that the adaptive control system is effective in maintaining the upshift and downshift quality at an acceptable level.
Comparing the Chrysler and GM approaches on clutch-to-clutch shift control, we notice many similarities. For example, one common feature is the identification of specific events such as onset of slip in the off-going clutch. Other common features are the learning of fill time and fill pressure, and the use of adaptation to accommodate life cycle and build-to-build variations. There are several differences between these two approaches. The first one is that no accumulator is present in GM approach, which is more reasonable. The second one is that the Chrysler approach uses turbine speed information to determine the adaptation and the timing of the clutch pressure application.
The GM approach uses both input and output shaft acceleration information and the base control scheme to determine adaptation and the timing of the clutch pressure application. The third difference is how the inertia phase is controlled. The Chrysler approach uses bang-bang and proportional control to control the on-coming clutch pressure so that the turbine acceleration follows a predetermined value. The GM approach feedback controls the on-coming clutch pressure to make the on-coming clutch slip follow a desired trajectory. Finally, the GM approach uses a model reference adaptation technique, while the Chrysler approach uses bang-bang and proportional control.

Figure 2.46 Power-on downshift base control strategy (Hebbale and Kao, 1995)
Figure 2.47 Vehicle test results showing power-on upshift adaptation (Hebbale and Kao, 1995)
The nonlinear sliding mode control technique is used by Cho and Hedrick (1987, 1989) for the control of clutch-to-clutch shifts. The oncoming and offgoing clutch torques are chosen as manipulated variables so that the designed controller is generic and applicable to any clutch-to-clutch shift control systems. However, the results may be unrealistic, since some important and probably dominant dynamic effects such as the dynamics related to hydraulic pressure manipulation have been ignored and can drastically change the total system behavior.
Sliding mode control is one type of robust control that can deal with nonlinearities in the system. The objective of sliding mode control is to design a control law to effectively account for parameter uncertainty and unmodeled dynamics, and quantify the resulting modeling/performance trade-offs (Slotine and Li, 1991). In Cho's work, sliding mode control is used to achieve closed loop control of the turbine speed. He considers two cases. One case involves design of a sliding model shift controller for the transmission alone, the simulation results being shown in Figure 2.49 and Figure 2.50. The peak-to-peak acceleration is reduced by 66% and the peak-to-peak jerk (derivative of acceleration) is reduced by 82%. The required clutch torques and pressure profiles are depicted in Figure 2.51 and Figure 2.52, and are unrealistic. The authors also point out the pressure profiles are impossible to realize in practice. They then design an "integrated engine transmission control strategy" using the "nonlinear dynamic inversion" technique with sliding mode control, which uses the spark advance as a manipulated input instead of using the offgoing clutch pressure. In this case, the oncoming clutch is the manipulated transmission variable. The spark advance/retard is solved for by inverting the engine dynamic equations. Control smoothing approximations are used to avoid exciting unmodeled dynamics. The acceleration and jerk trajectories from the simulation are shown in Figure 2.53 and Figure 2.54. 65% and 64% reductions in the peak-to-peak acceleration and jerk are achieved. The clutch pressures and spark advance/retard commands and the engine torque reduction required to provide the shift characteristics are depicted in Figure 2.55 - Figure 2.57, and are still unrealistic. All the results given so far are simulation results.
Figure 2.49 Longitudinal acceleration: open-loop vs. transmission-only closed-loop (Cho, 1987)

Figure 2.50 Longitudinal jerk: open-loop vs. transmission-only closed-loop (Cho, 1987)

Figure 2.51 Required clutch torques: transmission-only closed-loop (Cho, 1987)
Figure 2.52 Required clutch pressures: transmission-only closed-loop (Cho, 1987).

Figure 2.53 Longitudinal acceleration: open-loop vs. integrated engine and transmission closed-loop (Cho, 1987)

Figure 2.54 Longitudinal jerk: open-loop vs. integrated engine and transmission closed-loop (Cho, 1987)
Figure 2.55 Clutch pressures: integrated engine and transmission closed-loop (Cho, 1987)

Figure 2.56 Spark command from mbt (negative=retard): integrated engine and transmission closed-loop (Cho, 1987)

Figure 2.57 Percent reduction in engine torque required for integrated engine and transmission closed-loop control (Cho, 1987)
The variation of the clutch friction coefficient is a critical factor that affects shift quality. Continuing on Cho and Hedrick’s (1987) work, Glitzenstein and Hedrick (1990) propose a sliding model adaptation scheme to compensate for the variation of clutch friction coefficient during a clutch-to-clutch shift. Three adaptive algorithms based on Lyapunov stability criteria are proposed to compensate for these variations, which are the “direct adaptive approach” based on a sliding control law to adjust parameters in the inertia phase, the “indirect least squares estimator with the direct law” forming a composite adaptive approach in the inertia phase, and a third approach which extends the composite principle from the inertia phase of the shift to the torque phase. In the torque phase, the objective is to control the off-going clutch slip to zero. In the inertia phase, the objective is to maintain a desired vehicle acceleration trajectory. As in Cho’s work, the clutch torques are used as inputs assuming perfect actuator dynamics.

The simulation results of these three adaptive approaches with the same amount of parameter error are shown in Figure 2.58 - Figure 2.60. When the first algorithm -- "direct adaptive approach" is applied in the inertia phase of the shift, parameter $\mu_2$ (friction coefficient of the oncoming clutch) and shift performance converge in 20-25 shifts. When the second algorithm -- "composite adaptive approach" is applied in the inertia phase, the convergence occurs in 7 or 8 shift. Applying the third algorithm -- "composite adaptive approach" in both the torque and inertia phases, the convergence of parameters $\mu_1$ (friction coefficient of the offgoing clutch) and $\mu_2$ and performance occurs in 4 to 5 shifts. As the results show, the third adaptive algorithm gives the best results in terms of parameter and performance convergence speed.
Figure 2.58 (a) Vehicle acceleration, 25 Shifts, Direct inertia adaptation, 25% $\mu_2$ Error
(b) Normalized $\mu_2$ behavior, 25% error (Glitzenstein and Hedrick, 1990)

Figure 2.59 (a) Vehicle acceleration, 8 Shifts Composite inertia adaptation, 25% $\mu_2$ Error
(b) Normalized $\mu_2$ behavior, 25% error (Glitzenstein and Hedrick, 1990)

Figure 2.60 (a) Vehicle acceleration, 5 Shifts Composite torque and inertia phase adaptation, 25% $\mu_2$ error (b) Normalized $\mu_2$ behavior, 25% error (Glitzenstein and Hedrick, 1990)
A unique approach for clutch-to-clutch load transfer is proposed by Brown and Hrovat (1988) of Ford Motor Company. Contrary to the conventional belief that the output torque drop during the torque phase (load transfer phase) is unavoidable, they propose a clutch-to-clutch load transfer strategy in the torque phase which will eliminate the torque phase drop of the output shaft torque as shown in point F in Figure 2.61. In the invention, no torque converter is present in the system. The engine is coupled to two clutches, one being the off-going clutch and the other one the oncoming clutch. The engine torque is held constant while the load is transferred from the first clutch to the second clutch, and the vehicle speed is assumed to be constant during the load transfer.

At the beginning of the torque phase, engine inertia or the equivalent rotating inertia of other components of the driveline are used to avoid as much of the output torque drop as necessary to complete the power-on shift so that the output torque changes smoothly without the abrupt transients associated with the conventional method. In the torque phase, the offgoing clutch pressure is controlled to decrease so that the clutch slip will increase until it reaches a prescribed value corresponding to the engine speed increase $\Delta N_e$. This amount of clutch slip is maintained by closed-loop control of the offgoing clutch pressure. As shown in Figure 2.61, the torque phase where the clutch-to-clutch load transfer occurs lies between the time instants $t_D$ and $t_E$. In the inertia phase (after time $t_E$), the speed ratio is closed-loop controlled by controlling the oncoming clutch pressure. The output torque change during the gear ratio change is very smooth. The method, however, does require pressure transducers to achieve the desired pressure control. In the absence of pressure transducers, this control can be approximated by closed-loop speed ratio control. The idea is unique, but it requires a detailed and validated powertrain model to predict the relationship between engine speed increase and the pressure in the off-going clutch. In addition, the proposed scheme requires control of engine speed increase by controlling the off-going clutch pressure, and at the same time, the oncoming clutch pressure needs to be controlled to achieve load transfer. The complexity involved in the pressure control system is not discussed, and may pose a limit to the applicability of this approach. For systems with torque converters, the nonlinear
coupling between the engine and transmission adds more difficulty in applying the proposed scheme.

The difficulty in (disk) clutch-to- (disk) clutch control is that two friction elements have to be actively controlled simultaneously, the load transfer between the two elements during torque phase being crucial. Other types of shift control involving the coordination of two actively controlled friction elements are band clutch-to-clutch and band clutch-to-band clutch shift control. Due to the unique nature of the band clutch which has different torque capacities in the two slip directions, the coordination between the off-going friction element and the on-coming friction element is simpler than for disk clutch-to-disk clutch control. To differentiate the band clutch from the disk clutch, we will use the term "band" to represent band clutch.

![Figure 2.61 Clutch-to-clutch load transfer (Brown and Hrovat, 1988)](image-url)
A transmission requiring a band-to-clutch shift control is seen in the 2nd-to-3rd gear upshift in Figure 2.2. Other such transmissions are reported by Japanese automotive researchers. Usuki et al. (1996) and Hiramatsu and Naruse (1986) from Mitsubishi Motor Corporation describe electronically controlled automatic transaxles. All bands and clutches except the reverse clutch are independently controlled by pulse-width modulated solenoid valves. The band and clutch pressures are indirectly controlled by spool valves powered by hydraulic pressure from solenoid valves. As in clutch-to-clutch control, band-to-clutch shifts also require timing the pressure supply and release between the two friction elements. The desired timing is determined by learning control to cope with build-to-build variations and aging of the powertrain components. In the inertia phase, the rate of turbine speed change is feedback controlled so that the actual rate of the input speed change follows the predetermined target value, which is similar to the Chrysler approach.

Hojo et al. (1992) describe a band-to-band load transfer in the Toyota A350E automatic transmission. Figure 2.62 shows the layout of the transmission. Table 2.1 lists the engagement schedule. A 2nd-3rd gearshift involves shifting of two gear sets synchronously in opposite directions in several phases. As shown in Figure 2.63, the first phase is the load transfer from a one-way clutch to a clutch. Phase 2 is when the rear gear unit shifts. Phase 3 represents the band-to-band shift region. Phase 4 represents the region where the front gear unit completes the shift, while the rear gear unit continues to shift independently.

As we can see from Figure 2.63, phase 3 of the shift involves a band-to-band shift (B2 to B0). Figure 2.64a and b show the shift characteristics for the cases where the completion of the shift by the front gear unit lagged and led the rear gear unit respectively. Large variations of the output torque are observed in both cases. The band-to-band cooperative control system is shown in Figure 2.65. A state space model is developed and a state feedback controller designed. The control inputs are the electric
Two-speed unit  Three-speed unit-Simpson type planetary gear set
(Front gear unit)  (Rear gear unit)

Input shaft  Output shaft

Figure 2.62 Schematic drawing of the A350E transmission (Hojo et al., 1992)

<table>
<thead>
<tr>
<th>Gears</th>
<th>S1</th>
<th>S2</th>
<th>S3</th>
<th>C0</th>
<th>C1</th>
<th>C2</th>
<th>B0</th>
<th>B1</th>
<th>B2</th>
<th>B3</th>
<th>F0</th>
<th>F1</th>
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<td>O</td>
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<tr>
<td>3</td>
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<tr>
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<tr>
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<td>O</td>
</tr>
</tbody>
</table>

S1, S2, S3 : Shift solenoid No.1, No.2, No.3, O: ON, X: OFF
●: Operating, ▲: Operating only in the selector position for engine brake

Table 2.1 Operation conditions of the A350E transmission (Hojo et al., 1992)

Figure 2.63 Transient characteristics, shifting from 2nd to 3rd gear (Hojo et al., 1992)
current outputs to the linear solenoids, which control the B2 and B0 pressures. The outputs are B2 and B0 speeds, which are measured by speed sensors. The target speed is calculated from the transmission output speed and engine throttle position. The state feedback gain matrix K is obtained by solving a discrete type Riccati equation to minimize a quadratic performance function. Another feature of this work is the use of compensators for delays (including response delays of the electrohydraulic systems as well as dead times) to suppress speed and output shaft torque fluctuations. The effect is shown in Figure 2.66. However, the authors give no details on the magnitudes of the delays and the approaches used to compensate the delays. Effects of variation in engine torque, μ-values of friction materials, hydraulic pressures etc, on the synchronization timing gap and the total shifting time are evaluated by Monte Carlo simulation. The authors claim that their technique is the industry's first modern control theory aided shifting technique (Hojo et al., 1992).

![Figure 2.64](image)

Figure 2.64  (a) Shift characteristics: the completion of shift by the front gear unit lagged
(b) Shift characteristics: the completion of shift by the rear gear unit lagged
(Hojo et al., 1992)
Figure 2.65 Clutch-to-clutch cooperative control system (Hojo et al., 1992)

Figure 2.66 Effectiveness of compensation for delays (Hojo et al., 1992)
2.4.1.4 Shift schedule control

All automatic transmissions include shift schedule control. Based on vehicle operating conditions – throttle opening, vehicle speed etc., the appropriate preprogrammed schedule is selected.

Nissan's R01A model transmission (Shinohara et al., 1989) employs a throttle sensor and a vehicle speed sensor to detect the throttle opening and the vehicle velocity. This information is used to actuate the shift solenoids in the automatic transmission according to a preprogrammed schedule. It is possible to select the desired shift schedule from several choices, for example "normal", "power", "hold", and "snow" etc. Fixed shift schedules work well when the transmission is in a relatively new condition. After some time, the characteristics of the engine and friction elements will change, and fixed schedules no longer guarantee good shifts. To design shift schedules that will work for a wide range of operating conditions (for example, including road gradient changes), some researchers have proposed using advanced control techniques like fuzzy logic (Sakai et al., 1990; Yamaguchi, 1993), neural networks (Usuki et al., 1996) and adaptive control methods (Leising et al., 1990b).

One example of shift schedule control is presented by Sakai et al. of Honda (1990) using fuzzy logic. A survey is conducted to obtain the information about shift scheduling based on personal experience. Qualitative knowledge of control based on human experience is coded with many IF-THEN fuzzy production rules. The output from each "IF THEN" statement represents a shift variable value. Road conditions can be incorporated into shift schedule design, and driving safety is enhanced.

Another similar example is presented by Yamaguchi et al. of Nissan (1993). This paper focuses on solving shift-hunting problems encountered when a vehicle is climbing up a hill or driving on a winding road. Usually, shift schedules are designed to provide an optimum balance of fuel economy and driveability. Such fixed shift schedules present no problems when a vehicle is operated under ordinary driving conditions, such as on a flat road at sea level without any steep gradients and under normal passenger and cargo loads. However, fixed shift schedules can result in hunting in the form of excessive gearshifts.
when there is large road resistance as in uphill driving. To overcome such hunting problems, Yamaguchi et al. present a variable shift schedule control system by applying fuzzy logic to derive the relationship between the driver’s actions and the road gradient using the throttle valve opening and vehicle speed as the information inputs. The driver is considered to be a comprehensive sensor that can react to change in driving conditions. Statistical analysis of driving behavior data is compiled based on a large number of drivers operating under a variety of road conditions. This method offers an alternative way to incorporate human driving behavior into shift schedule design.

Another solution to the hunting problem is proposed by Leising et al. of Chrysler using adaptive shift scheduling (1990b). In the example considered by them, a hunting problem occurs during the 3rd - 4th shift when the vehicle is climbing up a moderate grade slope. The approach used to resolve the hunting problem is to make the 3rd - 4th shift schedule adaptable to current vehicle operating conditions. The 3rd gear and the 4th gear torques are calculated from the torque converter characteristic map (when the torque converter is not locked up) or from the engine torque and gear ratio. In the 3rd gear, the controller calculates the level of acceleration that the vehicle would be capable of maintaining in the 4th gear, and, if the level is too low, the controller inhibits the 3rd - 4th shift until the climbing is done.

Usuki et al. (1996) propose a “neural network” based shift schedule which determines the need for engine braking with high accuracy on descending roads and prevents frequent shifting on flat or ascending roads. In this scheme, a learning control algorithm is used for determining shift timing and to modify the original shift timing depending on the driving habits of individual drivers. Also, optimum control of engine brake timing is implemented by using neural networks based on factors such as the road gradient, vehicle speed, brake force, and steering angle (Usuki et al., 1996).

Shift schedule control is different from shift quality control in that the latter involves control of continuous variables, whereas the former involves switching between discrete variables. The primary interest in this dissertation is on shift quality control.
2.4.2 Torque converter clutch control

There are two types of torque converter clutch control. One is the torque converter clutch lock-up control which is a feature seen in many automatic transmissions (Shinohara et al., 1989; Schwab, 1984; Taga et al., 1982; Kawata and Yoshida, 1988; Hojo et al., 1992; Tsangarides and Tobler, 1985). The objective of lock-up control is to lock up the impeller and the turbine of the torque converter when the vehicle speed reaches a certain level, to avoid the slip losses caused by these two members rotating at different speeds. The other type of control is torque converter clutch slip control (Hiramatsu et al., 1985; Shinohara et al., 1989; Kondo et al., 1990; Kono et al., 1995; Osawa et al., 1995; Hrovat and Colvin, 1996). The objective of slip control is to reduce transmission of the torque variation generated by the engine, and to realize low fuel consumption by extension of operation of the clutch to lower engine speeds. In highway driving situations, noise and vibration problems are less severe as compared to urban driving situations and the torque converter clutch may be allowed to lock up under these conditions. Clutch slip control systems maintain a specified low nonzero slip in the torque converter so that engine noise and vibration are not transmitted to the drivetrain. If clutch slip control is extended to the low engine speed region, it will improve transmission efficiency and hence fuel economy.

Shinohara et al. (1989) describe the torque converter clutch control system in Nissan’s model R01A automatic transmission. Constant slip control is performed in the low speed range and when the throttle opening is small. Smooth lock-up control is carried out in the medium and high speed ranges and when the throttle opening is medium to large. Feedback control is implemented so that the slip speed is kept at a specified level. This is accomplished by manipulating the hydraulic pressure in the torque converter clutch so that the lock-up clutch slips continuously without engaging. Smooth lock-up control is performed by manipulating the hydraulic pressure as part of a closed loop before lockup and manipulating the pressure open loop when lockup occurs, to avoid a harsh transition. However, no quantitative information and details of controller design are provided.
Hiramatsu et al. (1985) from Mitsubishi Motors Corporation present a minimal slip control system for the torque converter damper clutch, which is shown in Figure 2.67. Oil pressure applied to the clutch is electronically closed-loop controlled to keep clutch slip to a minimum amount for reducing the transmitted engine torque variation and improving fuel economy. The stability of the feedback system is important, as is knowledge of the factors that affect the stability of the system. Among these factors, impairment of the $\mu-\omega_{\text{slip}}$ characteristics due to the failure of the friction materials or the automatic transmission fluid is the most severe. If the friction coefficient becomes negative, the system could become unstable and hunting may result. This is shown in Figure 2.68 and Figure 2.69.

The authors propose a control method that adjusts the oil pressure in proportion to slip deviation from a target value and to its integral. The block diagram of the damper clutch control system is shown in Figure 2.70. Slip detection and solenoid valve dynamics are approximated by first order lag models. A theoretical analysis of system stability is accomplished based on static characteristics of the engine and torque converter and the system equations are linearized. By selecting the controller parameters in the stable range, an originally unstable system is stabilized as a result. If the clutch facing materials are impaired too much, clutch shuddering occurs at a resonant frequency (about 20Hz) unique to the drive line system. Using the model described above, the authors predict the slip hunting frequency to be 1Hz which agrees well with the hunting frequency of the actual powertrain.

Clutch torque - slip speed characteristics vary with driving conditions and aging. These variations make it difficult to design a feedback control system that is stable under all conditions. Kono et al. (1995) and Osawa et al. (1995) from Toyota address this problem by using $H_\infty$ robust control design theory. They develop a torque converter clutch slip control system and implement it in the model A541E automatic transmission. The block diagram of this clutch control system is shown in Figure 2.18. The model is explained in section 2.3.3.
Figure 2.67 Damper clutch control system (Hiramatsu et al., 1985)

Figure 2.68 $\mu$-\omega characteristic of clutch facing material (Hiramatsu et al., 1985)
If the friction coefficient versus slip speed characteristic ($\mu$ - $\omega_{\text{slip}}$ characteristic) has a positive gradient ($\frac{d\mu}{d\omega_{\text{slip}}} > 0$), the clutch operates smoothly. On the other hand, if the gradient is negative, it may cause clutch shudder. So, the selection of the automatic transmission fluid and clutch lining must ensure a positive gradient of the $\mu$ - $\omega_{\text{slip}}$ characteristic.
characteristic to prevent the clutch from shuddering. Kono et al. (1995) use a linear solenoid valve with a feedback circuit to control solenoid current and claim that the resulting fluid pressure is affected less by changes in fluid viscosity or air entrainment than the more conventional variable duty cycle solenoid valve.

The feedback control system is shown in Figure 2.71. $G_0$ is the ARMAX model at the nominal operating point, the input being solenoid current, and the output being torque converter slip speed. $\Delta$ is the characteristic variation from the nominal model, which takes into account the variations of operating conditions such as engine load change, turbine speed change, and the ATF aging effect. $H_\infty$ control is used to optimize the stability and control performance. The frequency response is shaped by choosing two weighting functions to minimize the sensitivity function (transfer function relating the slip speed deviation to the target slip speed) and the complementary sensitivity function (which is inversely related to the robustness to unstructured multiplicative modeling error). The latter weighting function targets the frequency range with large variations in the plant transfer characteristic, in this case the high frequency region, and the former weighting function targets the low frequency region where higher accuracy is desired. Compared with the conventional PID controller, the $H_\infty$ controller is seen to be superior in stability. The simulation results are shown in Figure 2.72. Vehicle testing results show that fuel efficiency is improved by 5% - 7% and noise and vibration are at an acceptable level, the latter being shown in Figure 2.73.

Kono et al.'s (1995) approach is very similar to that of Hiramatsu et al. (1985). They use similar models of the torque converter. The manipulated inputs in both cases are solenoid current, and the output is torque converter slip speed. Kono et al. use first order dynamics and delay to represent the solenoid response and clutch piston dynamics, and later use an identified ARMAX model and $H_\infty$ control to design the slip control system. Though Hiramatsu et al. (1985) model the solenoid valve as a first order system, no clutch dynamics are modeled and the linearized torque converter characteristics are used. PI control is used to design the controller in the latter case. One other difference is that Hiramatsu et al. include a detection delay in their system.
Figure 2.71 Feedback control system (Kono et al., 1995)

Figure 2.72 Comparison of PID and H∞ controllers (Kono et al., 1995)
(a) Step response simulations (nominal point)
(b) Step response simulations (DC-gain variation)
(c) Step response simulation (ATF deterioration)
Another example of torque converter slip control is seen in the work reported by Martin and Redinger (1993) of Chrysler. Feedback control by the transmission controller establishes a duty cycle for the converter clutch control valve to provide a pressure level, which maintains the desired slip at the torque converter. The minimum solenoid operating duty cycle to keep the converter clutch slipping is learned so that the control can quickly respond to light throttle (low torque) vehicle operation and avoid converter clutch lockup at low vehicle speeds. To achieve full engagement of the torque converter clutch from a fully released condition, the transmission controller first uses “slip acceleration control”. Once slip is brought into the desired range, the solenoid duty cycle is incremented until the solenoid is fully on to achieve full engagement of the torque converter clutch. This scheme is similar to previous approaches, but there is no quantitative information provided in the reference.

Hrovat and Colvin (1996) from Ford describe torque converter bypass clutch slip control using nonlinear inverse dynamics. The control system is shown in Figure 2.74 – Figure 2.77. The slip of the torque converter bypass clutch is controlled by a feedforward and feedback control system on the basis of throttle angle, engine speed and turbine speed. Clutch hydraulic pressure is controlled by the solenoid valve which gets the
control signal from the above control system. A second order nonlinear dynamics model of the type

\[ \begin{align*}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= -f(x_1, x_2) + K_u(x_1, x_2)U_{dc}
\end{align*} \]  

(2.3)

is used to characterize the bypass clutch dynamics relating the solenoid duty cycle to the bypass clutch control pressure. Input-output feedback linearization is used to obtain the effective duty cycle as follows,

\[ U_{dc} = \frac{1}{K_u(x_1, x_2)} + \left( P_d - C_2 \dot{e} - C_1 e \right) \]  

(2.4)

where \( x_1 \) is the bypass clutch pressure, \( x_2 \) is the time derivative of \( x_1 \), \( e \) and \( \dot{e} \) are errors, \( C_1 \) and \( C_2 \) are controller gains, \( P_d, \dot{P}_d, \ddot{P}_d \) are the desired clutch pressure and its time derivatives. \( K_u \) is a function of the state variables and its time derivatives, and \( U_{dc} \) is the effective duty cycle. Substituting equation (2.4) into equation (2.3), the system is linearized. The solenoid inverse model can be a simple static inverse, or it could include the solenoid dynamics. However, the authors do not give details. The effective delay \( T_{ef} \) includes engine torque production delay, solenoid delay, computer calculation delay etc. The inverse duty cycle \( U_{dc} \) in Figure 2.77 is an effective preview of the incoming torque disturbance at the bypass clutch.

In addition to the two main types of torque converter clutch control, Taga et al. (1982) of Toyota Motor discuss coordination of the lockup clutch engagement with the shift schedule to achieve a smooth shift. This is done by disengaging the lockup clutch only momentarily during shifting to maximize fuel economy while retaining the damping action of the hydrodynamic unit to soften shift shock. However, poor control of the timing of the releasing and engaging of the lock-up clutch would lead to engine speed fluctuations which will result in degraded shift quality.
Figure 2.74 Torque converter bypass clutch slip control system: overview
(Hrovat and Colvin, 1996)

Figure 2.75 Torque converter bypass clutch slip control system: subsystem 1
(Hrovat and Colvin, 1996)
Figure 2.76 Torque converter bypass clutch slip control system: subsystem 2
(Hrovat and Colvin, 1996)

Figure 2.77 Torque converter bypass clutch slip control system: subsystem 3
(Hrovat and Colvin, 1996)
2.4.3 Estimation techniques

Torque and acceleration information are critical for better control of shift transients. But installing torque and acceleration sensors can be costly, besides raising reliability issues. Hence, many researchers have developed estimation techniques to acquire torque and acceleration information for use in shift control.

Minowa et al. (1994) propose a smooth gearshift control system using estimated torque. Turbine torque estimation is done by using torque converter steady-state characteristics in the form of the pump capacity coefficient (pump torque divided by the engine speed squared) and the torque ratio (turbine torque divided by pump torque) as functions of the speed ratio (turbine speed divided by pump speed). Typical steady-state torque converter characteristics are shown in Figure 2.78. Turbine torque is calculated as the product of pump capacity, torque ratio and engine speed squared. Simulation results for the estimated turbine torque are shown in Figure 2.79. The detection of the start of the inertia phase is critical. Conventional shift start detection is done by the transmission speed ratio (wheel angular velocity divided by turbine speed) which does not provide good detection. The authors propose using the “transmitted torque ratio” which is obtained by dividing the driven shaft (transmission output shaft) torque by the estimated turbine torque. The transmitted torque ratio changes abruptly at the point of ideal shift start (inertia phase start) and is nearly equal to the 2nd gear ratio for a 1st-2nd upshift. During the inertia phase, the transmitted torque ratio is constant (the 2nd gear ratio). This is then used to estimate the transmission output shaft torque. The estimated and measured transmission output shaft torque are shown in Figure 2.80. The estimated transmission output shaft torque is a delayed version of the measured torque near the end of the inertia phase. The application of this scheme is discussed in the section on integrated powertrain control.
Figure 2.78 Hydraulic torque converter characteristics (Minowa et al., 1994)

Figure 2.79 Simulated results for estimated turbine torque (Minowa et al., 1994)
In order to reduce or totally eliminate the tuning required above, Ibamoto et al. (1995) present an alternative method for output torque estimation which is relatively "tuning free". It is applied to a car without an electronically controlled throttle valve. So, engine torque control is achieved by ignition timing retardation. The turbine torque is estimated in two ways as shown in Figure 2.81. One involves use of the engine map together with torque ratio characteristics to estimate the torque converter output shaft torque. The other involves use of the torque converter characteristics in the form of the capacity coefficient and torque ratio as functions of the speed ratio. These two methods are carried out simultaneously and the one with higher accuracy is chosen. This is shown in Figure 2.82. However, in order to determine which method to use for a certain torque region (high or low), experiments have to be done ahead of time and cannot be done online. Therefore, this method also relies on predetermined data for the estimation technique. Figure 2.83 shows the estimated turbine torque during a shift compared with
the measured data. The torque wave forms are similar except in phase. The control application of this estimation scheme is deferred to the next section.

Figure 2.81 Two methods of torque estimation (Ibamoto et al., 1995)
Figure 2.82 Accuracy of estimated turbine torque (Ibamoto et al., 1995)

Figure 2.83 Measured and estimated turbine torque (Ibamoto et al., 1995)
Another aspect of estimation related research concerns speed and acceleration estimation. Since speed measurement is quite noisy, differentiating speed measurements to obtain acceleration signals is not very practical. Therefore, the application of stochastic estimation algorithms is necessary to filter the noise from the speed measurements and to estimated the measured speed and its derivative. Hebbale and Ghoneim (1991) of General Motors propose a speed and acceleration estimation technique based on the Kalman filter approach. This technique needs only speed measurements. The filter gives an optimum estimate for both speed and acceleration which minimizes the mean square estimations error. The state dynamics and measurement equations are linear and time invariant. But the plant dynamics for the powertrain are actually highly nonlinear and time variant because of shifting from one gear to another. The authors point out that a dynamic Kalman filter for such a system will be very complex and not practical for real implementation. Therefore, the authors assume that noise statistics are stationary and that a steady-state discrete-time Kalman filter is adequate. The experimental results for speed and acceleration estimation during a 1st - 2nd upshift at 30% throttle opening are shown in Figure 2.84 (a). The upper part of the figure shows the raw speed signal as a function of time. The speed measurement using one set of the filter gains is shown in the middle portion of the figure. It is obvious that the estimated speed is smoother than the raw data. The bottom portion shows the acceleration estimation using the same filter gains, which is noisy and not suitable for feedback control purposes. Using another set of filter gains, the acceleration estimation is much better but it has a lag of about 50 ms as shown in Figure 2.84 (b). Comparing Figure 2.84 (a) and (b), there is a trade-off in the design of the filter between the lag in speed and acceleration estimation and the noise level in acceleration estimation. A filter giving a smooth acceleration signal will estimate the speed with a larger lag-while-good speed estimation results in a noisy acceleration signal. Therefore, these two filters are implemented in parallel for different control purposes.
Figure 2.84 (a) Experimental results showing speed and acceleration estimation during an upshift. (b) Performance of the Kalman filter showing improved acceleration estimation (Hebbale and Ghoneim, 1991)
Integrated powertrain control during shifts

In response to the growing need to improve shift quality and optimize powertrain control, researchers have attempted to integrate engine and transmission control to achieve benefits that can not otherwise be obtained. In performing engine control, better estimation of transmission states and coordination with transmission control functions is extremely helpful. In performing transmission control, engine torque manipulation gives transmission control another degree of freedom. From a systems perspective, the engine and transmission are not stand-alone units. They have to be better coordinated to make the vehicle perform well. In certain control problems such as shift control and neutral-idle control, integrated engine—transmission control is inevitable for optimal powertrain control. In this section, we will focus on integrated powertrain control during shifts.

Shift shock is suppressed usually by controlling the line pressure and decreasing the engine torque during shifts. The objective of engine torque control when shifting up under load is to reduce the energy dissipated in the friction elements during shifting (Schwab, 1984). This can increase the life of the friction elements by shortening the clutch slip time. The objective of engine torque control when shifting down is to suppress the jerk which occurs when the overrunning clutch or friction element locks up at the end of the speed phase, and hence improve shift quality.

Engine torque control together with clutch pressure control (Schwab, 1984; Shinohara et al., 1989) can improve shift quality and is commonly used in transmission shift control today. Engine torque can be controlled by ignition timing retardation (Schwab, 1984; Tugcu et al., 1986; Shinohara et al., 1989; Danno et al., 1989; McKenny et al., 1993; Noguchi et al., 1993; Ibamoto et al., 1995; Usuki et al., 1996), throttle control (Tugcu et al., 1986; Danno et al., 1989; Minowa et al., 1994) and fuel flow control (Martin and Redinger, 1993). Ignition timing retardation is the most commonly used method because there is nearly no time delay involved, but it has a limitation because it is limited by knock occurrence at the upper limits of engine torque. On the other hand, electronically controlled throttle valves can provide a better way to increase or decrease engine torque (Minowa et al., 1994). But the problem with throttle control is
that there are actuator and manifold delays. Spark control acts much faster than throttle control (Hrovat and Powers, 1988). Fuel flow control also has fuel injector delays, and it is a difficult task to coordinate shift control together with air-to-fuel ratio control which is a complex control problem in itself. MIMO control of air and fuel flow rates to perform simultaneous A/F ratio control and shift quality control has not been reported.

As mentioned above, manipulating line pressure during shifts can reduce shift shocks. At the same time, manipulation of engine torque during shift transients in coordination with line pressure control can improve shift quality. Figure 2.85 shows the effect of combining line pressure control with engine torque control to achieve smoother shifts.

Figure 2.85 Integrated control procedure (Shinohara et al., 1989)
Another example of integrated shift control is proposed by Kondo et al. (1990) and Taniguchi and Ando (1991) for the Toyota "ECT- i" automatic transmission model A341E, used in LEXUS LS400 passenger cars. The engine torque and clutch pressure are manipulated together during shifting so that a smooth shift can be obtained. When the engine control unit detects the start of the inertia phase, engine torque control starts. When the input shaft speed becomes close to the synchronous rotational speed, the engine control unit determines it to be the completion of the gearshift and terminates engine torque control as shown in Figure 2.86. Ignition timing retardation is used here as the engine torque control method. Figure 2.86 shows the effect of engine torque control on shift quality.

A smooth gearshift control system using estimated torque is proposed by Minowa et al. (1994). An electronically controlled throttle is required to control engine torque. Throttle valve control is done during the inertia phase to reduce the deviation of the estimated shaft torque from the target shaft torque. PID control is used and controller
gains are determined by experiment. The throttle valve control block diagram using the estimated transmission output shaft torque is shown in Figure 2.87. Figure 2.88 shows the transmission output shaft torque characteristics using throttle only and using the throttle and line pressure control together. The acceleration test results are shown in Figure 2.89. By coordinating the control of turbine torque, engine torque, and line pressure, upshift shocks are suppressed by as much as 35%. However, this method demands much effort for implementation of the torque estimation and furthermore, requires trial and error tuning of the controller.

An alternative method for shift control with output torque estimation which is relatively “tuning free” is described in Ibamoto et al. (1995). The block diagram for upshift control is shown in Figure 2.90. Figure 2.91 and Figure 2.92 show the algorithms for upshift and downshift control. Figure 2.93 and Figure 2.94 show the test results for the 1st-2nd upshift and the 4th -1st downshift respectively.

Figure 2.87 Detailed control block diagram of throttle valve control with estimated (transmission output shaft) torque feedback (Minowa et al., 1994)
Figure 2.88 transmission output shaft torque characteristics: (a) using conventional method, (b) using throttle control only, (c) using throttle and line pressure control (Minowa et al., 1994)
Figure 2.89 Comparison of longitudinal acceleration using proposed and conventional methods (Minowa et al., 1994)

Figure 2.90 Block diagram for upshift control with torque estimation (Ibamoto et al., 1995)
Figure 2.91 Algorithm for upshift control (Ibamoto et al., 1995)

Figure 2.92 Algorithm for downshift control (Ibamoto et al., 1995)
Figure 2.93 Experimental results for 1\textsuperscript{st} - 2\textsuperscript{nd} full throttle upshift (Ibamoto \textit{et al.}, 1995)

Figure 2.94 Experimental results for 4\textsuperscript{th} - 1\textsuperscript{st} downshift (Ibamoto \textit{et al.}, 1995)
In order to lower the system cost and to simplify the calibration process, Minowa et al. (1996) present a powertrain control system using only the transmission output shaft speed. Normally, the start of the engine torque control is determined using the gear ratio which needs a speed sensor for the transmission input shaft. Here, the authors propose to use only the transmission output shaft speed to detect the start of the engine torque control, which in turn can eliminate the use of the speed sensor on the input shaft of the transmission. The relationship between the transmission output shaft speed and the longitudinal acceleration data is obtained by analyzing the data. The time chart of the proposed method is shown in Figure 2.95. The torque control period and its beginning and end (Figure 2.95) and the proportional gain for the torque control loop have to be calculated. The first two quantities factors can be calibrated by analyzing the transmission output shaft speed data for every downshift. The third quantity has to be determined by vehicle testing. Some experimental results are shown in Figure 2.96. The downshift shock is eliminated by using the proposed method. Although this method works for a research vehicle, it may not be universally acceptable, since there is no accompanying theoretical analysis. But this method does provide a possible way of doing transmission control using fewer sensors.

The above review has focused on integrated engine and transmission control during shifts. In recent years, development of alternative powertrains such as electric vehicle and hybrid vehicles has gained momentum. We will not go into details of these developments, but will look at a final example of integrated electric motor and transmission control. Patil and Davis (1990) report a synchronous shifting scheme for an electric motor and transmission. The layout of the vehicle system is shown in Figure 2.97, and includes an induction motor, and a two-speed transmission with no torque converter. During the gear upshift, the load is transferred from a one-way clutch to a clutch, and during downshift, the load transfer is from clutch to one-way clutch. The powertrain model is derived. The solenoid valve plus shift hydraulic system model is represented as a third-order model, with an open loop time constant of 200 ms. The dominant plant pole is at 6.66 rad/sec, and the natural frequency of the other two poles is 56 rad/sec (8.9 Hz).
A phase-lead controller is designed to improve the valve solenoid response time to 69 ms. However, this design did not work experimentally in a bench test setup. They found out that the valve body has different frequency response than the model they used. An experimentally determined transfer function is used as the model, and a PID controller is designed. No details are given about this later design. The upshift control scheme is implemented in the vehicle, test results being shown in Figure 2.98. The motor torque is controlled together with the on-coming clutch pressure. The downshift control scheme did not work, however. This approach is conceptually similar to integrated engine and transmission control. In addition to speed sensors, the torque sensor is required for the torque feedback control implementation.

![Figure 2.95 Time chart of proposed control algorithm (Minowa et al., 1996)](image-url)
Figure 2.96 Comparison of longitudinal acceleration during 3rd-1st and 4th-1st kick-down shifting (Minowa et al., 1996)

Figure 2.97 Layout of electric vehicle (Patil and Davis, 1990)
2.4.5 Integrated powertrain control – neutral idle control

Another example of integrated engine-transmission control is “Neutral Idle Control”. It is well known that vehicles with automatic transmissions have worse fuel economy than those equipped with manual transmissions. The biggest fuel economy difference is seen when the vehicle is moving in city traffic (urban driving). For a vehicle with a manual transmission, the driver will shift the transmission into neutral when he brakes the car or when the car stops at traffic lights. For a vehicle with a conventional automatic transmission, when the car stops in city traffic, the transmission is kept in first gear and the engine is idling unless the driver intentionally pushes the lever to “N” (neutral) position. By examining the velocity profile – FUDS (Federal Urban Driving Schedule) shown in Figure 2.99, the idle condition (when vehicle velocity is zero) is seen to last about 19% of the driving time. In one such cycle of duration 1370 seconds, the “neutral-to-drive” shift happens 15 times, which means an average one “neutral-to-drive” shift every 1.5 minutes for a manual transmission equipped vehicle.

For an automatic transmission equipped vehicle under such conditions, the torque converter incurs significant slip loss since the pump is rotating at the engine crankshaft speed, while the turbine is not moving. The engine control strategy has to provide sufficient fuel to maintain a smooth idle. The resulting requirement to increase the fuel supply by advancing the throttle during idle also reduces the effective fuel economy. Further, undesirable exhaust emission levels are possible when the driveline is loaded with the engine in its idle condition (Brown and Kraska, 1993). On the other hand, the speed ratio of the torque converter is zero, which means that the torque amplification effect of the torque converter is maximum, i.e. torque is still transferred to the drivetrain. The engine vibrations produced in the idling operation are transferred to the drivetrain and felt by the passengers (Yamamoto, 1986; Brown and Kraska, 1993). The noise, vibration and harshness (NVH) of the powertrain may be severe due to the direct torque flow path between the turbine and the input shaft of the transmission mechanical system compared to the NVH characteristics for the full neutral condition when the manual shift valve is in the neutral (N) position (Brown and Kraska, 1993).
Figure 2.98 Experimental results for integrated motor-transmission control

(a) 20% Power upshift results. (b) Full power upshift results

(Patil and Davis, 1990)

107
To improve fuel economy and the NVH quality of automatic transmission equipped vehicles in urban driving, "Neutral Idle Control" is being considered by several automotive manufacturers. The key idea is to disengage the input clutch from the torque converter turbine when the vehicle brakes and the vehicle stops in city traffic with the manual shift valve at "Drive" position, and when the vehicle is coasting. If the input clutch is disengaged under such conditions, the torque converter turbine will rotate freely so that slip loss is reduced. As a result, total fuel economy could be improved. At the same time, the driveline is disengaged from the turbine, engine vibrations will not be transmitted, and so the powertrain will be quieter.

An alternative motivation for neutral idle control is the so called "anti-creep" control (Yamamoto, 1986; Kanamo and Kashihara, 1988). "Creep" means that the vehicle moves slowly even under idling operation of the engine since the engine torque is still transmitted to the drivetrain. Creeping of the vehicle indirectly leads to engine idling
vibration. When creep occurs, the driver brakes more to force the vehicle to stop. The drag of the torque converter on the engine causes more fuel consumption compared with that normally consumed during engine idling. Furthermore, the transmission fluid in the torque converter tends to become hot and can lead to overall overheating. Thus, the forward clutch is disengaged in such situations to prevent “creeping”.

New problems arise with this new concept. First, the engaging and disengaging frequency of the input clutch is increased (Figure 2.99) which calls for enhancement of the clutch material to withstand high frequency use of the input clutch. Second, engaging the input clutch at the end of the neutral idle mode is like engaging the 1st gear. This shift requires a fast response of the pressure control system. However, it is more difficult than other gear shifts since, during the idling phase, the engine idle speed control loop comes into play which makes the neutral idle control problem more difficult. Therefore, the objective of neutral idle control is to improve fuel economy and acceptability of the shift.

A survey of reported work on neutral idle control by researchers in the automotive companies is reported below.

Moan (1984) of Ford Motor Company describes a fuel saving strategy which involves forcing the control system to assume a neutral condition when the engine is idling or when the vehicle is coasting. If the forward clutch is disengaged during coasting or idling, when the driver opens the throttle to continue the normal driving mode, an undesirable delay is encountered due to delay in engagement of the clutch. During this delay period, the engine speed increases and so when the clutch finally engages, the high engine speed causes harshness in the engagement of the clutch. Moan’s strategy in overcoming this problem is to let the clutch pressure regulator valve assume a threshold pressure within the forward clutch via a mechanical linkage. This causes the clutch to become frictionally engaged with a minimum torque capacity, and allows the clutch capacity to increase rapidly as line pressure is distributed to the clutch following the throttle opening command. This scheme relies on the system hardware design, and no active control is involved.
Brown and Kraska (1993), also of Ford, focus on a closed-loop clutch engagement control system to achieve a smooth engagement of the forward clutch following a neutral idle state. The improved control system makes it possible to supply a residual or threshold pressure to the input clutch servo and to engage the input clutch at the beginning of the acceleration phase by using a closed-loop controller. This is to guarantee that the forward clutch is filled and stroked and ready to be fully applied when the throttle is advanced by the driver. The closed-loop control system can compensate for changes in oil temperature and spring force tolerances so that clutch engagement is repeatable. The slip control system is shown in Figure 2.100. Torque converter slip is monitored by measuring the engine and turbine speeds. Low-pass filters are used to acquire useful speed information from engine and turbine speed measurements. As shown in Figure 2.100, a PID controller is used to ensure low slip error. A variable-force solenoid is used to generate the needed clutch pressure.

Figure 2.100 Neutral idle control system (Brown and Kraska, 1993)
Leising et al. (1990a) of Chrysler claim a method for adaptively idling an electronic automatic transmission system, which is similar to Brown and Kraska (1993). The adaptive system provides an adaptive-idle mode of operation aiming at achieving significant gains in fuel economy. The three stages of adaptive idling are shown in Figure 2.101. The control logic determines which stage condition is true and then executes the corresponding control algorithm. If the first stage "adaptive idle entrance" condition is true, then the ON time of the input clutch solenoid valve is controlled to bring the slip speed (engine speed minus turbine speed) to within 50 rpm. The control algorithm then controls the input clutch solenoid valve to maintain the target slip in the steady state, which is the second stage. The third stage "adaptive idle exit" causes the controller to engage the input clutch. Though the basic principle of adaptive idling is described, coordination of the idle speed control loop with the transmission adaptive idle control and means for fast application of the input clutch are not mentioned.

It is easy to identify some common features of the above mentioned neutral idle control schemes. In general, neutral idle control can be divided into three phases. The first phase involves disengaging the forward clutch, the second phase is the idling phase, and the third phase is the forward clutch application phase. The key in neutral idle control is to maintain a minimum pressure threshold in the forward clutch during the second phase while maintaining satisfactory engine idle speed control, and to ensure fast and smooth engagement in the third phase. These features are clearly shown in the following implementation.
A practical implementation of the neutral idle technique in a front wheel drive 4-speed automatic transmission is reported by Hayabuchi et al. (1996). Speed sensors are used to detect the engine, turbine shaft and output shaft speeds. The input clutch C-1 control is the key to the neutral idle control system. An enhanced clutch lining surface to achieve uniform pressure distribution and high durability is used. The C-1 clutch pressure is controlled by a linear solenoid valve. A representative time chart for neutral idle control is shown in Figure 2.102. As shown in the figure, the neutral idle control has three phases. Phase 1 involves the C-1 clutch release. Phase 2 involves C-1 clutch being in neutral. Phase 3 involves C-1 clutch application.

In phase 1, C-1 clutch pressure is controlled by a linear solenoid valve. The pressure is changed to shorten the releasing time and to release the C-1 clutch gradually. When C-1 clutch slip occurs signaling the start of phase 2, the engine reduces the injected fuel. However, no details are given on whether fuel flow rate is closed loop controlled. In phase 2, the C-1 clutch piston is kept at the stroke position ("the Piston Stroke End") where the drag torque is minimal. The control method to keep the piston at the piston stroke end shown in Figure 2.103 is the most important point in the neutral control. This is also mentioned by Moan (1984), Brown and Kraska (1993) and Yamamoto (1986), and Kanamo and Kashihara (1988). Phase 3 involves application of the C-1 clutch, the objective being to achieve smooth engagement of the C-1 clutch. One important point discussed by the authors is the heat generated at the surface lining. As shown in Figure 2.104, the application time lag of the clutch is related to heat generation at the surface lining. The simulation model is shown in Figure 2.105. The simulation result shows that the heat generation has a tendency to increase sharply at higher engine speeds when the application time lag is greater than 0.2 seconds. The rate of turbine speed change is controlled. This is one of the techniques discussed in the shift control part of this review. Self-learning control is used to adjust the C-1 clutch pressure control during idling. According to the authors, the rate of turbine speed change (slope of turbine speed vs. time trajectory) could differ by about 500 rpm/sec even at the same C-1 clutch pressure. So, the correction is determined according to the average turbine speed change rate shown in
Figure 2.106. Figure 2.107 shows that fuel consumption is reduced by 20% during the neutral condition. Figure 2.108 shows fuel economy improvement versus vehicle mean speed. The authors claim that this is the world’s first example of neutral-idle control technology in the automotive market.

The variation of the fluid bulk modulus poses big problem for such a process. This can be due to the fluid temperature change and/or air entrainment inside the fluid. Air entrainment causes decrease of fluid “stiffness” and is unavoidable since the high speed rotating motion of the torque converter could cause air to be easily entrained in the fluid. One way to incorporate fluid viscosity change due to temperature is to detect the temperature and when engaging the clutch, set the line pressure accordingly to make sure the clutch will engage rapidly (Shinohara et al., 1989). In this work, the neutral to drive

![Figure 2.102 Time chart of neutral control (Hayabuchi et al., 1996)](image)

113
engagement (manual shift from neutral to drive) is done by increasing the line pressure for a specific period of time following the selector lever operation and then lowering the line pressure again so that the engagement shock is reduced. In this configuration, the line pressure is controlled in the presence of accumulators, which is very different from the "neutral idle control" idea that calls for frequent fast engagement of the input clutch with reduced shift shock at the end of the neutral-to-drive shift.

Figure 2.103 Control method in phase 2 (Hayabuchi et al., 1996)
Figure 2.104 Heat generation vs. C-1 clutch application time lag by calculation in phase 3 (Hayabuchi et al., 1996)

Figure 2.105 Simulation model of neutral control (Hayabuchi et al., 1996)
Figure 2.106 Control method of c-1 clutch pressure during idling in phase 3 (C-1 application) (Hayabuchi et al., 1996)

Figure 2.107 Fuel injected quantity at the moment of neutral control (Hayabuchi et al., 1996)

Figure 2.108 Fuel economy improvement vs. vehicle mean speed (Hayabuchi et al., 1996)
2.5 Summary

This chapter gives a review of the state of the art of powertrain related research issues with emphasis on the automatic transmission. A review of the modeling of powertrain components as well as systems is presented first, followed by a review of the control techniques used in powertrain control systems with emphasis on the control of automatic transmissions.

Powertrain models can be developed by a combination of physical principles and empirical data followed by model validations. Depending on the specific research application, powertrain models can have different levels of complexity. Satisfactory powertrain models are helpful in predicting dynamic behaviors of automotive powertrains and can be a useful tool for design engineers. Though powertrain models with emphasis on automatic transmissions cover a broad range, there are certain areas that have not been studied or modeled sufficiently. For example, validated dynamic torque converter models have not been reported in the literature and the dynamic behavior of torque converters is critical in studying certain dynamic phenomena. Hydraulic clutch modeling is also at a relatively early stage. High speed rotation of hydraulic clutches will stir up the fluid in the clutch and the fluid centrifugal effect may be important in predicting the overall clutch behavior. In addition, the frictional characteristics of clutch facing material play an important role in clutch/band characteristics and are critical to the prediction of shift quality. Moreover, many powertrain component models such as hydraulic pressure actuator models need to be refined to capture their contribution to powertrain dynamics.

Control techniques used in automatic transmission related control applications are reviewed. Many of these are open loop control techniques that demand a lot of tuning and can not respond adequately to change in operating conditions. Closed loop control is seen in some control applications, for example, inertia phase control, clutch-to-clutch shift control, torque converter clutch control etc. PID control is the most commonly used technique in these applications, controller selection being based on experimental tuning and being hardware specific. Modern linear and nonlinear control techniques are used in some of the applications. Due to the nature of the operation of automatic transmission,
the use of linear control theory is limited, since automatic transmissions are seldom at
equilibrium except when the vehicle is at idle or coasting.

Though control techniques are used more and more nowadays in automatic
transmissions, certain powertrain control problems have not been solved or studied. With
more electronically controlled transmissions being available, integrated engine-
transmission control will be the future trend in powertrain control. "Neutral-Idle" control
is one such area that calls for much attention, since it offers the potential to further
improve fuel economy and vehicle performance. "Neutral-Idle" control involves the
coordination of engine and transmission control to achieve optimal overall performance.
Theoretical analysis of "Neutral-Idle" control has not been reported in the literature.

The following chapters will discuss the details of the proposed powertrain
modeling and control research forming the subject of this thesis.
CHAPTER 3

POWERTRAIN MODEL WITH EMPHASIS ON THE AUTOMATIC TRANSMISSION

The following chapter describes a representative powertrain dynamic model with emphasis on the automatic transmission. For the purpose of studying the dynamic behavior and control of automatic transmission equipped vehicles, a time-domain powertrain model is presented here, including models of the engine, torque converter, transmission mechanical system, transmission shift hydraulic system, and simplified vehicle longitudinal dynamics. The powertrain response during gear shifts and the dependence of the response on powertrain component behavior and operating conditions is examined.

3.1 Overview of the powertrain model

The powertrain to be described here is closely related to a General Motors powertrain with a 3800cc V6 engine and a Hydramatic automatic transmission. The numerical values of model parameters have been obtained from a number of sources in the open literature – Cho (1987), Cho and Hedrick (1989a), Karmel (1986, 1988a,b), and Runde (1986). The modeled transmission is GM Hydramatic THM440 model which, with the addition of electronic control, is close to the current 4T60E model. Despite the basis of the model being a GM powertrain, it is neither intended, nor is it claimed here, that the model presented here represents the behavior of a powertrain in production.

119
A schematic of the structure of the powertrain model is shown in Figure 3.1. There are six subsystems in the model — the engine, torque converter, transmission mechanical system, shift schedule logic, shift hydraulic system (line and clutch/band pressure and torque generation) and vehicle dynamics. Symbols are defined in the nomenclature section and also following the equations below where they occur first. The equations defining the dynamic behavior of the subsystems are described in the time domain because of their nonlinearity, and because of structural changes associated with gearshifts.

The powertrain model is developed and coded using the Matlab/Simulink software package. Figure 3.2 shows the Simulink diagram overview of the powertrain model. In the following sections, the model equations for each subsystem are described in detail. Simulation results will be presented in section 3.8. Detailed Simulink block diagrams of each subsystem are attached in Appendix A.

Figure 3.1 Powertrain model (with emphasis on the transmission)
3.2 Mean-value engine model

The engine model used here is a mean-value engine model adapted from Cho (1987), Moskwa (1988), and Cho and Hedrick (1989a). There are three states in the engine model – the mass of air $m_a$ in the intake manifold, the engine speed $\omega_e$, and the fuel flow rate $m_f$. In addition, there are two transport delays in the model, which capture the discrete nature of a four-stroke engine – the intake-to-torque production delay $\Delta t_i$ and the spark-to-torque production delay $\Delta t_s$. A more complete model for the engine size of interest here is given by Jonathan Dawson (Dawson, 1998).

Considering conservation of mass in the intake manifold, intake air dynamics are described by

$$m_a = m_{ai} - m_{ao}$$

where $m_{ai}$ is the mass flow rate of air entering the intake manifold and is modeled as:

\[ m_{ai} = \]
\[ m_{\text{in}} = \text{MAX} \cdot TC \cdot PRI \]  

\( \text{MAX} \) is the maximum air flow rate at wide-open-throttle and choked flow, in this case,  
\[ \text{MAX} = 0.1843 \text{ Kg/s} \]  

\( TC \) is the normalized throttle area characteristic indicating how it changes with throttle angle \( \alpha \) (degrees), the curve-fit equation being,  
\[ TC = \begin{cases} 
1 - \cos(1.14459 \cdot \alpha - 1.06) & \alpha^\circ \leq 79.46^\circ \\
1 & \alpha^\circ > 79.46^\circ 
\end{cases} \]  

\( PRI \) is the normalized pressure influence, indicating the effect of the pressure ratio across the throttle body (ratio of the intake manifold plenum and upstream ambient pressures), the curve-fit equation being,  
\[ PRI = 1 - \exp\left(9 \cdot \left(\frac{P_m}{P_{\text{aim}}} - 1\right)\right) \]  

Under the assumption of uniform pressure distribution, the intake manifold pressure \( P_m \) and the intake manifold air mass \( m_a \) are related by the ideal gas law,  
\[ P_m \cdot V_m = m_a \cdot R \cdot T_m \]  

where  
\[ V_m \] is the intake manifold volume, in this case, \( V_m = 0.0027 m^3 \).  
\[ R \] is the gas constant, and  
\[ T_m \] is the manifold temperature.  

The mass flow rate of air entering the combustion chamber \( m_{\text{in}} \) is modeled as:  
\[ m_{\text{in}} = c_1 \cdot \eta_{\text{vol}} \cdot m_a \cdot \omega_e \]  

where \( \omega_e \) is the engine speed, \( c_1 \) is a constant given by  
\[ c_1 = \frac{V_e}{4\pi \cdot V_m} \]  

where \( V_e = 0.0038 m^3 \) is the engine displacement. \( \eta_{\text{vol}} \) is the engine volumetric efficiency, which is a parameter used to measure the effectiveness of an engine's induction process.
(Heywood, 1988). There are many factors that affect volumetric efficiency, such as fuel, engine design and engine operating variables. In this case, volumetric efficiency is given by the following empirical expression.

$$\eta_{vol} = \left(24.5 \omega_e - 3.10 \times 10^4 \right) m_e^2 + \left(-0.167 \omega_e + 222 \right) m_e + \left(8.10 \times 10^{-4} \omega_e + 0.352 \right)$$  \hspace{1cm} (3.9)

Combining equations (3.1) and (3.6)-(3.8), the intake manifold dynamics are modeled by:

$$\frac{dP_m}{dt} + \frac{\eta_{vol} V_e \omega_e}{4 \pi V_m} P_m = m_{at} \frac{RT_m}{V_m}$$ \hspace{1cm} (3.10)

The fueling dynamics are modeled by,

$$\tau_f \cdot \dot{m}_f + \dot{m}_f = \dot{m}_{fe}$$ \hspace{1cm} (3.11)

where \( \tau_f \) is the effective fueling time constant and is modeled by

$$\tau_f = 0.050 + \frac{15 \pi \cdot m_{fe} \cdot \beta}{\omega_e \cdot MAX}$$ \hspace{1cm} (3.12)

The effective fueling time constant is a complex term that includes

1. the fueling transport delay, for a 6-cylinder engine using sequential port fuel injection method, could be between \( 0^\circ \sim 120^\circ \) crankangle,
2. the time delay from the start of injection to when the intake valve closes,
3. the transport delay due to the injector firing interval, i.e., if the pulse is still occurring when the intake valve closes, then the remaining fuel will be delayed by two crankshaft revolutions (720° crankangle) or \( \frac{4 \pi}{\omega_e} \) seconds, and
4. the fuel film lag.

There are two terms in equation (3.12). The first one is a lag time constant. The second term is a transport delay expression assuming that spraying is completed before the intake valve opens, the transport delay being averaged as
where $m_{fe}$ is the commanded fuel rate

$m_f$ is the actual fuel rate entering the combustion chamber

$\beta$ is the desired air-to-fuel ratio

The rotational dynamics of engine are modeled by

$$I_e \cdot \dot{\omega}_e = T_i - T_f - T_a - T_p$$

(3.14)

where $T_i$ is the engine indicated torque

$T_f$ is the engine friction torque

$T_a$ is the accessory torque

$T_p$ is the torque converter pump torque

$I_e$ is the effective inertia of engine and pump, 0.1454 Kgm$^2$.

The engine friction torque is curve-fitted to the experimental data and modeled by

$$T_f = 0.1056 \omega_e + 15.10$$

(3.15)

Here, 15.1 Nm is a static friction term.

The engine indicated torque $T_i$ is modeled by

$$T_i = c_T \cdot \frac{m_{so}(t - \Delta t_{in})}{\omega_e(t - \Delta t_u)} \cdot AFI(t - \Delta t_u) \cdot SF(t - \Delta t_u)$$

(3.16)

where $AFI$ is the normalized air fuel influence, the curve-fit equation from the empirical data being

$$AFI = \cos(7.3834 \cdot (A/F - 135))$$

(3.17)
$SI$ is the normalized spark influence, which is a function of spark advance/retard from MBT (maximum brake-torque), the curve-fit equation from the empirical data being

$$SI = (\cos(SA - MBT))^{2.875}$$  \hspace{1cm} (3.18)

$c_T$ is the torque constant representing the maximum torque capacity of an engine for a given air mass, engine speed, $AFI = 1$, and $SI = 1$. In this case, $c_T = 498636 \text{ Nm/Kg}$.

$\Delta t_{it}$ is the intake to torque production delay, $= 5.48/\omega_e$ seconds. $\Delta t_{st}$ is the spark to torque production delay, $= 1.3/\omega_e$.

The torque production model is a steady-state model and does not contain any dynamic elements, except for the process delays associated with the four-stroke combustion process. Since the combustion dynamics are much faster than the air and fuel dynamics, this approach is reasonable. The maximum possible torque is reduced by two normalized functions. The first one is $AFI$. This function represents the decreased torque when there is not enough fuel to utilize all of the air in the cylinder, or if there is insufficient air to burn all of the fuel. The second normalized function is the spark influence function, which decreases the indicated torque as a function of how far the spark advance is from the MBT spark timing. In this case, the spark advance is always retarded from MBT to avoid the knock problems (Moskwa, 1988).

3.3 Torque converter model

The modeling of the torque converter raises issues of some complexity. The complexity arises from the three-dimensional nature of the flow inside the torque converter resulting from the complexity of the blade geometry. The torque converter model incorporated in the powertrain model described here is the static model of Kotwicki (1982). The approach combines equations for mass conservation, angular momentum conservation, and energy conservation to derive expressions for torques on the pump and turbine as functions of the pump and turbine speeds, torque converter geometry, and fluid properties. Empirical correlations are used for different energy loss
terms. Fluid inertial effects are neglected. Moreover, the three dimensional nature of fluid flow is neglected, and analysis is based on conditions existing at a mean flow path under the assumption that these conditions hold uniformly across the torque converter cross section as well. The analysis procedure can be found in Kotwicki’s paper and in Rizzoni and Srinivasan (1998).

A simpler form of Kotwicki’s torque converter model, formed by using a quadratic regression fit to the experimental data, is used here. Such a regression yields terms involving the square of the pump speed and the turbine speed, as well as a term involving the product of the pump and turbine speeds. Different regression equations are used for the torque amplification and fluid coupling modes.

Numerical values for the torque converter model are obtained from Cho (1987). The model values are determined for General Motors’ THM440 Hydramatic automatic transmission and are appropriate for the powertrain of interest. For the torque multiplication mode, i.e. \( \omega_t/\omega_p < 0.9 \):

\[
T_p = 3.4325 \times 10^{-3} \omega_p^2 + 2.2210 \times 10^{-3} \omega_p \omega_t - 4.6041 \times 10^{-3} \omega_t^2 \\
T_t = 5.7656 \times 10^{-3} \omega_p^2 + 0.3107 \times 10^{-3} \omega_p \omega_t - 5.4323 \times 10^{-3} \omega_t^2
\]  

(3.19)

where

\( T_p \) is the pump torque, Nm

\( T_t \) is the turbine torque, Nm

\( \omega_e \) is the engine speed, rad/sec, and

\( \omega_t \) is the turbine speed, rad/sec.

For the fluid coupling mode, i.e. \( \omega_t/\omega_p \geq 0.9 \):

\[
T_p = T_t = -6.7644 \times 10^{-3} \omega_p^2 + 32.0084 \times 10^{-3} \omega_p \omega_t - 25.2441 \times 10^{-3} \omega_t^2
\]  

(3.20)

As the derivation indicates, fluid inertial effects are not accounted for in the model. The moments of inertia of the pump and turbine, \( I_p \) and \( I_t \) respectively, can be
lumped with the engine inertia and the transmission input element inertia and thus their effects accounted for.

The torque converter model here does not include the overrunning mode of torque converter operation. In this mode, turbine speed exceeds the engine (pump) speed, so reversal of flow within the torque converter could be expected (Kotwicki, 1982). Modeling this phase would be necessary when handling engine braking situations that would occur with power-off downshifts. One reference for modeling the overrunning mode of torque converter operation can be found in Hrovat and Tobler (1985).

3.4 Transmission mechanical system model

The schematic diagram for the THM 440 automatic transmission is shown in Figure 3.3. \( C_1, C_2, C_3 \) and \( C_4 \) represent plate clutches, and \( C_1 \) and \( C_3 \) are one-way clutches. \( B_R \) and \( B_{12} \) are band clutches. The transmission has an input planetary gear set, a reaction planetary gear set, and a final drive gear set. The clutch engagement schedule and the overall transmission gear ratios for different gears, including the final drive gear ratio of 2.84, are shown in Table 3.1. The same transmission has been modeled by Runde (1986) and Cho (1987). Numerical values for the transmission mechanical system model have therefore been obtained from these sources.

3.4.1 Planetary gear train model

The transmission model consists of lumped inertia to help represent dynamic effects. Thus, this dynamic model is not intended to capture gear dynamics and vibrations, but is useful only for lower frequency phenomena such as gear shifting. The planetary gear sets for the transmission of interest, with the corresponding inertias lumped outside the gear train, are shown in Figure 3.4. All speeds are assumed to be positive in the same direction, according to the sign convention shown in Figure 3.4.
Assuming perfect power transfer, the steady state speeds and torque for the elements of the planetary gear sets shown in Figure 3.4 are related by:

\[ \omega_{Ci} = R_{Si} \omega_{Si} + R_{Ri} \omega_{Ri} \]  
(3.21)

\[ \omega_{Cr} = R_{Sr} \omega_{Sr} + R_{Rr} \omega_{Rr} \]  
(3.22)

with the constraints

\[ \omega_{Ci} = \omega_{Rr} \]  
(3.23)

\[ \omega_{Ri} = \omega_{Cr} \]  
(3.24)
where

\( \omega \) is angular velocity, with the first subscript \( S, R \) or \( C \) denoting sun gear, ring gear or carrier, the second subscript \( i \) or \( r \) denoting input gearset or reaction gearset,

- \( N_{Si} \) is the number of teeth of the input sun gear,
- \( N_{Sr} \) is the number of teeth of the reaction sun gear,
- \( N_{Ri} \) is the number of teeth of the input ring gear, and
- \( N_{Rr} \) is the number of teeth of the reaction ring gear.

Defining

\[
R_{Si} = \frac{N_{Si}}{N_{Si} + N_{Ri}} \quad (3.25)
\]

\[
R_{Sr} = \frac{N_{Sr}}{N_{Sr} + N_{Rr}} \quad (3.26)
\]

\[
R_{Ri} = \frac{N_{Ri}}{N_{Si} + N_{Ri}} \quad (3.27)
\]

\[
R_{Rr} = \frac{N_{Rr}}{N_{Sr} + N_{Rr}} \quad (3.28)
\]

we can get

Figure 3.4 Stick diagram of gear train including gear inertia

129
Similarly, the steady state torque equations for the input and reaction planetary gear sets are:

\[ T_{f1} = \frac{R_{Rr}}{R_{Sr}} T_{f2} - \frac{1}{R_{Si}} T_{Si} \]  
(3.31)

\[ T_o = \frac{1}{R_{Sr}} T_{f2} - \frac{R_{Ri}}{R_{Si}} T_{Si} \]  
(3.32)

The final drive gear ratio is \( R_D \), therefore the output shaft speed and torque are obtained by:

\[ \omega_o = R_D \omega_Cr \]  
(3.33)

\[ T_S = \frac{T_o}{R_D} \]  
(3.34)

In Figure 3.4, \( T_{Sid} \), \( T_{f1d} \), \( T_{f2d} \) and \( T_{od} \) are the dynamic torques acting on the corresponding lumped inertias at the ports of the planetary gear train. The relationships between these dynamic torques and steady state values of the torques at the corresponding ports are given by applying Newton's second law of motion:

\[ T_{Sid} - T_{Si} = I_{Si} \cdot \omega_{Si} \]

\[ T_{f1d} - T_{f1} = I_{Rr} \cdot \omega_{Rr} \]

\[ T_{f2d} - T_{f2} = I_{Sr} \cdot \omega_{Sr} \]  
(3.35)

\[ T_o - T_{od} = I_{Cr} \cdot \omega_{Cr} \]

where \( I_{Rr} \) is the inertia of the reaction ring - input carrier assembly.
$I_{Cr}$ is the inertia of the reaction carrier - input ring assembly

$I_{Si}$ is the inertia of input sun gear assembly

$I_{Sr}$ is the inertia of the reaction sun gear assembly

Alternative compact forms of the steady state speed and torque relationships among the four ports of the planetary gear train are shown in Table 3.2. Any two port conditions can be solved for if the other two port conditions are given.

3.4.2 Models of transmission shift dynamics

A transmission shift transient consists of two phases: the torque phase and the inertia phase. During gear shifting, different friction elements such as plate clutches or band clutches will engage or disengage to achieve the corresponding gear ratios. Based on the clutch schedule given in Table 3.1, we can identify the mechanical configurations for each in-gear phase. However, modeling shift transients is not a trivial task. The following sections give the dynamic equations for the transmission undergoing power-on upshifts from the first gear to the fourth gear. To make the derivations easy to follow, the dynamic model equations will be arranged sequentially starting from the first gear case. Note that the inputs to the transmission are the torque converter turbine torque $T_t$, the output shaft torque $T_o$ and the clutch/band torque $T_{Ci}$ or $T_{Bi}$. The outputs from the transmission are the turbine speed $\omega_t$, the input sun gear speed $\omega_{si}$, the reaction ring gear speed $\omega_{sr}$, the reaction sun gear speed $\omega_{sr}$, the reaction carrier speed $\omega_{cr}$, and the output shaft speed $\omega_o$.

3.4.2.1 Dynamic model of the transmission in the first gear

According to Table 3.1, clutch $C_1$ and band $B_{12}$ are engaged in the first gear. The freebody diagrams are shown in Figure 3.5. The dynamic equations are as follows:

$$I_t \cdot \dot{\omega}_t = T_t - T_{C2} - RT_{xpl}$$  \hspace{1cm} (3.36)
\[
\begin{align*}
\omega_{Sr} &= A \omega_{Si} \\
\omega_{Rr} &= A \omega_{Cr} \\
\omega_{Si} &= B \omega_{Rr} \\
\omega_{Sr} &= B \omega_{Cr} \\
\omega_{Rr} &= C \omega_{Sr} \\
\omega_{Si} &= C \omega_{Cr} \\
\omega_{Rr} &= D \omega_{Si} \\
\omega_{Cr} &= D \omega_{Sr} \\
\omega_{Si} &= E \omega_{Rr} \\
\omega_{Cr} &= E \omega_{Sr} \\
\omega_{Sr} &= F \omega_{Rr} \\
\omega_{Cr} &= F \omega_{Si}
\end{align*}
\]
(a)

\[
\begin{align*}
A &= \frac{1}{R_{Si}} \begin{bmatrix}
-R_{Si}R_{Rr} & 1-R_{Ri}R_{Rr} \\
R_{Si}R_{Sr} & R_{Ri}R_{Sr}
\end{bmatrix} \\
B &= \frac{1}{R_{Si}R_{Sr}} \begin{bmatrix}
R_{Sr} & -R_{Ri}R_{Sr} \\
-R_{Si}R_{Rr} & R_{Si}
\end{bmatrix} \\
C &= \frac{1}{R_{Si}R_{Rr}} \begin{bmatrix}
-R_{Si}R_{Sr} & R_{Si} \\
-R_{Sr} & 1-R_{Ri}R_{Rr}
\end{bmatrix} \\
D &= \frac{1}{1-R_{Ri}R_{Rr}} \begin{bmatrix}
R_{Si} & R_{Ri}R_{Sr} \\
R_{Sr} & R_{Si}R_{Rr}
\end{bmatrix} \\
E &= \frac{1}{R_{Si}} \begin{bmatrix}
1-R_{Ri}R_{Rr} & -R_{Ri}R_{Sr} \\
R_{Si}R_{Sr} & R_{Sr}
\end{bmatrix} \\
F &= \frac{1}{R_{Si}R_{Sr}} \begin{bmatrix}
1-R_{Ri}R_{Rr} & -R_{Si} \\
R_{Sr} & -R_{Si}R_{Sr}
\end{bmatrix}
\]
(b)

Table 3.2 Alternative representations of steady state speed and torque relationships for the planetary gear train

\[R_{Si} = \frac{N_{Si}}{N_{Ri} + N_{Si}}\]
\[R_{Ri} = \frac{N_{Ri}}{N_{Ri} + N_{Si}}\]
\[R_{Sr} = \frac{N_{Sr}}{N_{Ri} + N_{Sr}}\]
\[R_{Rr} = \frac{N_{Rr}}{N_{Ri} + N_{Sr}}\]

\[R_{Si} + R_{Ri} = 1\]
\[R_{Sr} + R_{Rr} = 1\]
Figure 3.5 Freebody diagrams of the transmission subassemblies in the first gear

\[ I_{Rr} \cdot \dot{\omega}_{Rr} = T_{C2} - T_{f1} \]  
\[ I_{Si} \cdot \dot{\omega}_{Si} = RT_{sp1} - T_{Si} \]  
\[ T_{f2} = RT_{B12} \]  
\[ I_{Cr} \cdot \dot{\omega}_{Cr} = T_o - T_s \cdot R_D \]  

Since clutch \( C_1 \) and band \( B_{12} \) are engaged, the constraints are:
\[ \omega_{Si} = \omega_i, \omega_{Sr} = 0 \]  

Clutch \( C_2 \) is not engaged, so \( T_{C2} \) (the second clutch torque capacity) is zero. From equations (3.36)-(3.40), we get:
\[ (I_i + I_{Si}) \cdot \dot{\omega}_i = T_i - T_{Si} \]  

From Table 3.2, we have:
The torque phase of the first - second gear upshift starts when the shift schedule logic module issues the upshift command and the 1-2 shift valve opens, filling clutch C₁. It ends when $R_T^{r_{pl}}$ (the reaction torque on clutch C₁) goes to zero. Referring to Figure 3.5, $T_{C2}$ and $R_T^{r_{pl}}$ are in the same direction. Increase in $T_{C2}$ causes a corresponding decrease in $R_T^{r_{pl}}$ indicating a load transfer from one clutch to the other. The dynamic equations are the same as those in the first gear, except for the additional input torque $T_{C2}$.

Therefore, the dynamic equation for $\omega_i$ is given by
\[ [(I_i + I_s) + I_R d_{ii}^2 + I_C d_{21}^2] \dot{\omega}_i = T_t + (d_{ii} - 1) T_{c2} - d_{21} R_d \cdot T_s \]  

(3.50)

\( T_{c2} \) is the \( C_2 \) clutch torque capacity calculated from the following formula (Deutschman, 1975; Runde, 1986):

\[ T_{ci} = P_{ci} \cdot \mu(\Delta \omega_{ci}) \cdot A_{ci} \cdot R_{ci} \cdot \text{sgn}(\Delta \omega_{ci}) \]  

(3.51)

where \( T_{ci} \) is the \( i^{th} \) clutch torque capacity, \( C_i = C_1, C_2, C_3 \) or \( C_4 \),

\( P_{ci} \) is the \( i^{th} \) clutch hydraulic pressure,

\( \mu(\Delta \omega) \) is the coefficient of friction, and is given by

\[ \mu(\Delta \omega) = \begin{cases} 
0.1545 & \Delta \omega = 0 \quad \text{(nonslipping)} \\
0.0631 + 0.0504e^{-0.033|\Delta \omega|} & \Delta \omega \neq 0 \quad \text{(slipping)} 
\end{cases} \]  

(3.52)

\( \Delta \omega_{ci,b} \) is the clutch/band slip speed

\[ \Delta \omega_{c1} = \omega_i - \omega_{si} \]  

(3.53)

\[ \Delta \omega_{c2} = \omega_i - \omega_{sr} \]  

(3.54)

\[ \Delta \omega_{c3} = \omega_i - \omega_{si} \]  

(3.55)

\[ \Delta \omega_{c4} = \omega_{si} \]  

(3.56)

\[ \Delta \omega_{b12} = \omega_{sr} \]  

(3.57)

\[ \Delta \omega_{br} = \omega_{sr} \]  

(3.58)

\( A_{ci} \) is the clutch plate or band piston area,

\( R_{ci} \) is the clutch plate or band effective radius, and

\( \text{sgn} \) is the sign of the clutch or band slip speed.

The directions of the torques and speeds are shown in Figure 3.5. Turbine torque \( T_t \) and turbine speed \( \omega_t \) have the same direction, as shown in the figure. This indicates a positive power flow to the turbine if both are positive. The direction of the output shaft torque \( T_s \) is opposite to that of \( T_t \) and the output shaft speed is assumed positive in the direction shown, indicating a power flow out of the transmission gear train to the
driveline. Clutch torque is calculated by equation (3.51). There are several underlying assumptions of this equation, namely, that the clutch pressure is uniform inside the clutch and over the clutch piston area, and the clutch torque varies instantaneously with clutch pressure indicating that inertia effects are assumed negligible. Also, the friction coefficient given by equation (3.52) is just one type of many possible friction models. Other types of friction models are surveyed by Armstrong-Helouvry (1994). When $RT_{sp1}$ (the reaction torque on the one-way clutch $C_1$) goes to zero, the torque phase ends and the inertia phase starts. $RT_{sp1}$ is calculated by

$$RT_{sp1} = T_t - T_{C2} - I_t \cdot \omega_t$$  \hspace{1cm} (3.59)

(2) Inertia Phase

When $RT_{sp1}$ goes to zero, the clutch $C_1$ starts overrunning, indicating zero torque transmission except for frictional effects, and the inertia phase starts. In the inertia phase, clutch $C_2$ slips at the beginning and locks up at the end. The dynamic equations in the inertia phase are as follows:

$$I_t \cdot \omega_t = T_t - T_{C2}$$  \hspace{1cm} (3.60)

$$I_{Rr} \cdot \omega_{Rr} = T_{C2} - T_{f1}$$  \hspace{1cm} (3.61)

$$I_{Si} \cdot \omega_{Si} = -T_{Si}$$  \hspace{1cm} (3.62)

$$T_{f2} = RT_{p12}$$  \hspace{1cm} (3.63)

$$I_{Cr} \cdot \omega_{Cr} = T_o - T_s \cdot R_D$$  \hspace{1cm} (3.64)

Combining the equations, we get:

$$\left(I_{Rr} + I_{Si} \cdot e_{11}^2 + I_{Cr} \cdot e_{21}^2\right) \omega_{Rr} = T_{C2} - e_{21} \cdot R_D \cdot T_s$$  \hspace{1cm} (3.65)

$$I_t \cdot \omega_t = T_t - T_{C2}$$  \hspace{1cm} (3.66)
where $e_{ij}$ is the $i^{th}$ row, $j^{th}$ column element of the $E$ matrix in Table 3.2. During the inertia phase, $\omega_{ri}$ increases and $\omega_{i}$ decreases. When $\omega_{ri}$ equals $\omega_{i}$, the $C_2$ clutch slip speed is zero. If, at the same time, $T_{C2} > RT_{C2}$, clutch $C_2$ locks up and the transmission is in the second gear.

3.4.2.3 Dynamic model of the transmission in the second gear

In the second gear, clutch $C_1$ is overrunning, whereas clutch $C_2$ and band $B_{12}$ are locked up. Therefore, $\omega_{ri} = \omega_{i}$ and $\omega_{si} = 0$. The dynamic model equations are:

\[
(I_t + I_{Rr}) \cdot \ddot{\omega}_i = T_t - T_{f1} \quad (3.67)
\]

\[
I_{si} \cdot \dot{\omega}_{si} = -T_{si} \quad (3.68)
\]

\[
T_{f2} = RT_{B12} \quad (3.69)
\]

\[
I_{cr} \cdot \dot{\omega}_{cr} = T_o - T_s \cdot R_D \quad (3.70)
\]

Combining the equations, we get:

\[
(I_t + I_{Rr} + I_{si} \cdot e_{i1}^2 + I_{cr} \cdot e_{21}^2) \cdot \ddot{\omega}_i = T_t - e_{21} \cdot Rd \cdot T_s \quad (3.71)
\]

\[
RT_{C2} = T_t - I_t \cdot \dot{\omega}_i \quad (3.72)
\]

3.4.2.4 Dynamic model of second – third gear upshift

During the second – third gear upshift, clutch $C_1$ remains locked up. Clutch $C_3$ and band $B_{12}$ are involved in the shift. Unlike shift situations where a one-way clutch is involved in the load transfer, the second – third gearshift involves two friction elements, the $C_2$ clutch and the $B_{12}$ band. The $C_3$ clutch starts pressurizing and the $B_{12}$ band starts depressurizing, i.e. $C_3$ is the oncoming clutch and $B_{12}$ is the offgoing clutch.
A band clutch's torque capacity is a function of the direction of rotation of the planetary gear element relative to the direction of force application on the band (Fanella, 1994). This is shown in Figure 3.6. The band clutch torque capacity is indicated by equations (3.73) and (3.74). In the energized mode,

$$ T_{Bj} = P_{Bj} \cdot A_{Bj} \cdot R_{Bj} \cdot \left( e^{\mu(\Delta \omega_{Bj}) \theta_{Bj}} - 1 \right) \cdot \text{sgn}(\Delta \omega_{Bj}) $$

(3.73)

In the de-energized mode,

$$ T_{Bj} = P_{Bj} \cdot A_{Bj} \cdot R_{Bj} \cdot \left( 1 - \frac{1}{e^{\mu(\Delta \omega_{Bj}) \theta_{Bj}}} \right) \cdot \text{sgn}(\Delta \omega_{Bj}) $$

(3.74)

where

- $T_{Bj}$ is the band torque capacity, $B_j = B_{12}$ or $B_R$,
- $A_{Bj}$ is the band piston area,
- $R_{Bj}$ is the band effective radius,
- $\theta_{Bj}$ is the band wrap angle,
- $\Delta \omega_{Bj}$ is the band slip speed.

The freebody diagram of the transmission during second – third gear upshift is shown in Figure 3.7. The dynamic equations are derived as follows:

Figure 3.6 Band clutch engagement modes (Fanella, 1994)
\[(I_t + I_{re}) \cdot \dot{\omega}_t = T_t - T_{f1} - RT_{sp3}\]  \hspace{1cm} (3.75)

\[I_{sl} \cdot \dot{\omega}_{sl} = RT_{sp3} - T_{sl}\]  \hspace{1cm} (3.76)

\[I_{sr} \cdot \dot{\omega}_{sr} = RT_{B12} - T_{f2}\]  \hspace{1cm} (3.77)

\[I_{Cr} \cdot \dot{\omega}_{Cr} = T_o - R_D \cdot T_s\]  \hspace{1cm} (3.78)

(1) **The torque phase**

In the torque phase, \(T_{B12} > RT_{B12}\), and the reaction sun gear assembly is still locked up. Therefore,

\[\dot{\omega}_{sr} = 0\]

\[RT_{B12} = T_{f2}\]  \hspace{1cm} (3.79)

Eliminating \(T_{sl}, T_{f2}, T_{f1}, T_o\), we get,

![Freebody diagram of automatic transmission during second – third gear upshift](image)

Figure 3.7 Freebody diagram of automatic transmission during second – third gear upshift
\[
(I_t + I_{kr} + I_{si} \cdot e_{11}^2 + I_{cr} \cdot e_{21}^2) \cdot \dot{\omega}_t = T_t + (e_{11} - 1)RT_{sp3} - e_{21} \cdot R_D \cdot T_s \tag{3.80}
\]

where

\[
RT_{sp3} = T_{C3} \tag{3.81}
\]

\[
RT_{B12} = T_{f2} = -e_{12} \cdot T_{C3} + (e_{11} \cdot e_{12} \cdot I_{si} + e_{21} \cdot e_{22} \cdot I_{cr}) \dot{\omega}_t + e_{22} \cdot R_D \cdot T_s \tag{3.82}
\]

\[
RT_{C2} = T_t - RT_{sp3} - I_t \cdot \dot{\omega}_t \tag{3.83}
\]

(2) The inertia phase

The inertia phase of the second – third gear upshift starts when \(T_{B12} = RT_{B12},\) i.e., the reaction torque on the \(B_{12}\) band is equal to the \(B_{12}\) band torque capacity. This is shown in Figure 3.8. Note that \(RT_{B12}\) and \(T_{B12}\) can go negative. This is because the band clutch torque capacity is a function of the direction of the band slip speed. Referring to Figure 3.8, when clutch \(C_3\) builds up sufficient capacity, \(RT_{B12}\) goes to zero (at \(t_1\)). When \(T_{C3}\) is further increased, \(T_{B12}\) changes direction, the band changes the operation from the energized mode to the deenergized mode. When the inertia phase starts, the band \(B_{12}\) starts slipping and we have the following conditions,

\[\text{Shift Starts}\]

\[\text{Torque Phase}\]

\[\text{Inertia Phase}\]

Figure 3.8 Torque patterns for the second – third gear upshift

140
\[ \omega_{Sr} \neq 0, \quad \Delta \omega_{B12} = \omega_{Sr} \quad (3.84) \]
\[ \omega_t = \omega_{Cr} \quad (3.85) \]
\[ RT_{B12} = T_{B12} \quad (3.86) \]
\[ RT_{sp3} = T_{C3} \quad (3.87) \]

The dynamic equations are given by equations (3.75) – (3.78), which are solved as follows:

\[
\begin{pmatrix}
\dot{\omega}_{Si} \\
\dot{\omega}_{Sr}
\end{pmatrix} = B
\begin{pmatrix}
\omega_{Cr} \\
\omega_{Cr}
\end{pmatrix} = B
\begin{pmatrix}
\dot{\omega}_{i} \\
\dot{\omega}_{Cr}
\end{pmatrix}
\quad (3.88)
\]

\[
\begin{pmatrix}
T_{f1} \\
-T_o
\end{pmatrix} = -B^\top
\begin{pmatrix}
T_{si} \\
T_{f2}
\end{pmatrix}
\quad (3.89)
\]

\[
\begin{pmatrix}
T_{si} \\
T_{f2}
\end{pmatrix} = -
\begin{pmatrix}
I_{si} & 0 \\
0 & I_{sr}
\end{pmatrix}
\begin{pmatrix}
\dot{\omega}_{Si} \\
\omega_{Sr}
\end{pmatrix} +
\begin{pmatrix}
RT_{sp3} \\
RT_{B12}
\end{pmatrix}
\quad (3.90)
\]

\[
\begin{pmatrix}
T_{f1} \\
-T_o
\end{pmatrix} = -
\begin{pmatrix}
I_t + I_{Cr} & 0 \\
0 & I_{Cr}
\end{pmatrix}
\begin{pmatrix}
\dot{\omega}_{Cr} \\
\omega_{Cr}
\end{pmatrix} +
\begin{pmatrix}
T_r - RT_{sp3} \\
-R_d \cdot T_s
\end{pmatrix}
\quad (3.91)
\]

Defining

\[
I_{23} =
\begin{pmatrix}
i_{2311} & i_{2312} \\
i_{2321} & i_{2322}
\end{pmatrix}
= \begin{pmatrix}
I_t + I_{Cr} & 0 \\
0 & I_{Cr}
\end{pmatrix} + B^\top
\begin{pmatrix}
I_{si} & 0 \\
0 & I_{sr}
\end{pmatrix} B
\quad (3.92)
\]

Then,

\[
\begin{pmatrix}
\dot{\omega}_{Cr} \\
\omega_{Cr}
\end{pmatrix} = (I_{23})^{-1}
\begin{pmatrix}
T_r + (b_{11} - 1)RT_{sp3} + b_{21} \cdot RT_{B12} \\
b_{12} \cdot RT_{sp3} + b_{22} \cdot RT_{B12} - R_d \cdot T_s
\end{pmatrix}
\quad (3.93)
\]

In equation (3.93), \( T_r, RT_{sp3} = T_{C3}, RT_{B12} = T_{B12} \) and \( T_s \) are the inputs to the system.

\( \omega_{Cr} (= \omega_t) \) and \( \omega_{Cr} \) can be solved for.
3.4.2.5 Dynamic model of the transmission in the third gear

In the third gear, clutches $C_2$ and $C_3$ are locked up. This is the direct drive gear for this transmission, i.e., the input power is directly transmitted to the output with a speed ratio of 1:1. Therefore,

$$\omega_i = \omega_{si} = \omega_{rr} \Rightarrow \omega_{cr} = \omega_{rr} \quad (3.94)$$

The freebody diagrams are the same as those in Figure 3.7 except that $T_{b12} = 0$ in the third gear. The dynamic equations are:

$$\begin{align*}
(I_t + I_{si} + I_{rr}) \cdot \dot{\omega}_i &= T_t - T_{f1} - T_{si} \\
I_{sr} \cdot \dot{\omega}_{sr} &= -T_{f2} \\
I_{cr} \cdot \dot{\omega}_{cr} &= T_o - R_D \cdot T_s
\end{align*} \quad (3.95)$$

Solving these equations, we get,

$$\begin{align*}
(I_t + I_{rr} + I_{si} + (f_{11} + f_{12})^2 \cdot I_{sr} + (f_{21} + f_{22})^2 \cdot I_{cr}) \cdot \dot{\omega}_i &= T_t - (f_{21} + f_{22}) \cdot R_D \cdot T_s
\end{align*} \quad (3.98)$$

where

$$\begin{align*}
f_{11} + f_{12} &= 1 \\
f_{21} + f_{22} &= 1
\end{align*} \quad (3.99)$$

Defining

$$\begin{align*}
I_{t3} &= I_t + I_{si} + I_{rr} + I_{sr} + I_{cr} \\
I_{t4} &= I_{rr} + f_{11} \cdot I_{sr} + f_{21} \cdot I_{cr}
\end{align*} \quad (3.100)$$

Then,

$$\begin{align*}
I_{t3} \cdot \dot{\omega}_i &= T_t - R_D \cdot T_s \\
RT_{C2} &= I_{t4} \cdot \dot{\omega}_i + f_{21} \cdot Rd \cdot T_s \quad (3.103)\\
RT_{sp3} &= T_t - I \cdot \dot{\omega}_i - RT_{C2}
\end{align*} \quad (3.104)$$
3.4.2.6 Dynamic model for third-to-fourth gear upshift

Clutch C2 is locked up from the second gear through the fourth gear. In the third gear, the one-way clutch C3 is locked up. In the fourth gear, C3 is overrunning and C4 is locked up. Load transfer during the third – fourth gear upshift occurs between clutch C3 and clutch C4. When the shift schedule logic module initiates the third – fourth gear upshift, clutch C4 is pressurized. When the C4 clutch takes over the load carried by the C3 clutch, C3 starts overrunning. This is when the inertia phase starts.

The freebody diagrams of the transmission for the third – fourth gear upshift and in the fourth gear are shown in Figure 3.9. The dynamic equations are:

\[
(I_t + I_{Rt}) \cdot \dot{\omega}_r = T_r - T_{f1} - RT_{sp3} \tag{3.105}
\]

\[
I_{sl} \cdot \dot{\omega}_{sl} = RT_{sp3} - RT_{C4} - T_{sl} \tag{3.106}
\]

![Freebody diagrams for the transmission during third-to-fourth gear upshift and in fourth gear](image)
\[ I_{s_r} \cdot \omega_{s_r} = -T_{f2} \]  
\[ I_{c_r} \cdot \omega_{c_r} = T_o - R_D \cdot T_s \]  

(3.107)  
(3.108)

(1) The torque phase

During the torque phase of the third–fourth gear upshift, load is transferred from clutch C_{3} to clutch C_{4}. Therefore, we have the following conditions,

\[ \omega_i = \omega_{r_r} = \omega_{s_l} = \omega_{s_r} = \omega_{c_r} \]  
\[ RT_{c_4} = T_{c_4} \]  

(3.109)  
(3.110)

Solving the dynamic equations, we get,

\[ \left( I_i + I_{s_l} + I_{r_r} \right) \cdot \dot{\omega}_i = T_i - T_{f_1} - T_{s_l} - RT_{c_4} \]  
\[ \begin{pmatrix} T_{f_1} \\ T_{s_l} \end{pmatrix} = -F^{T} \begin{pmatrix} T_{f_2} \\ -T_o \end{pmatrix} = \begin{pmatrix} f_{11} & f_{21} \\ f_{12} & f_{22} \end{pmatrix} \begin{pmatrix} -I_{s_r} \cdot \omega_{s_r} \\ -I_{c_r} \cdot \omega_{c_r} - R_D \cdot T_s \end{pmatrix} \]  

(3.111)  
(3.112)

\[ T_{f_1} = (f_{11} \cdot I_{s_r} + f_{21} \cdot I_{c_r}) \dot{\omega}_i + f_{21} \cdot R_D \cdot T_s \]  
\[ T_{s_l} = (f_{12} \cdot I_{s_r} + f_{22} \cdot I_{c_r}) \dot{\omega}_i + f_{22} \cdot R_D \cdot T_s \]  
\[ I_{r_3} \cdot \dot{\omega}_i = T_i - RT_{c_4} - R_D \cdot T_s \]  
\[ RT_{\text{sp}_3} = (I_{s_r} + f_{12} \cdot I_{s_r} + f_{22} \cdot I_{c_r}) \dot{\omega}_i + f_{22} \cdot R_D \cdot T_s + RT_{c_4} \]  
\[ RT_{c_2} = (f_{11} \cdot I_{s_r} + f_{21} \cdot I_{c_r} + I_{r_r}) \dot{\omega}_i + f_{21} \cdot R_D \cdot T_s \]  

(3.113)  
(3.114)  
(3.115)  
(3.116)  
(3.117)

(2) The inertia phase

When the load transfer between the C_{3} clutch and the C_{4} clutch is complete, i.e., \( RT_{\text{sp}_3} = 0 \), the torque converter turbine overruns the input sun gear assembly and the C_{3} clutch starts overrunning. Therefore, we have the following conditions,

\[ RT_{\text{sp}_3} = 0 \]  
\[ \omega_i = \omega_{r_r} \]  

(3.118)  
(3.119)

144
Solving these dynamic equations, we have the following results.

\[
\begin{bmatrix}
T_{Si} \\
-T_o
\end{bmatrix} = -A^T \begin{bmatrix}
T_{f2} \\
T_{f1}
\end{bmatrix} = \begin{bmatrix}
-a_{11} & a_{21} \\
a_{12} & a_{22}
\end{bmatrix} \begin{bmatrix}
T_{f2} \\
T_{f1}
\end{bmatrix}
\] \hspace{1cm} (3.121)

\[
\begin{bmatrix}
T_{Si} \\
-T_o
\end{bmatrix} = \begin{bmatrix}
I_{Sr} & 0 \\
0 & I_{Cr}
\end{bmatrix} \begin{bmatrix}
\dot{\omega}_{Si} \\
\dot{\omega}_{Cr}
\end{bmatrix} + \begin{bmatrix}
-RT_{C4} \\
-T_s \cdot R_D
\end{bmatrix}
\] \hspace{1cm} (3.122)

\[
\begin{bmatrix}
T_{f2} \\
T_{f1}
\end{bmatrix} = \begin{bmatrix}
I_{Sr} & 0 \\
0 & I_{Cr}
\end{bmatrix} \begin{bmatrix}
\dot{\omega}_{Sr} \\
\dot{\omega}_{Cr}
\end{bmatrix} + \begin{bmatrix}
t^{(T)}_t \\
0
\end{bmatrix} = \begin{bmatrix}
I_{Sr} & 0 \\
0 & I_{Cr} + I_{Rr}
\end{bmatrix} A \begin{bmatrix}
\dot{\omega}_{Si} \\
\dot{\omega}_{Cr}
\end{bmatrix} + \begin{bmatrix}
t^{(T)}_t
\end{bmatrix}
\] \hspace{1cm} (3.123)

From (3.121) – (3.123), we get,

\[
K_{34i} = \begin{bmatrix}
\dot{\omega}_{Si} \\
\dot{\omega}_{Cr}
\end{bmatrix} = \begin{bmatrix}
a_{21} \cdot T_t - RT_{C4} \\
a_{22} \cdot T_t - T_s \cdot R_D
\end{bmatrix}
\] \hspace{1cm} (3.124)

where

\[
K_{34i} = \begin{bmatrix}
a_{11} \cdot I_{Sr} + a_{21} \cdot (I_t + I_{Rr}) & a_{21} \cdot a_{22} \cdot (I_t + I_{Rr}) + a_{11} \cdot a_{12} \cdot I_{Sr} \\
a_{11} \cdot a_{12} \cdot I_{Sr} + a_{21} \cdot a_{22} \cdot (I_t + I_{Rr}) & a_{22} \cdot (I_t + I_{Rr}) + I_{Cr} + a_{12} \cdot I_{Sr}
\end{bmatrix}
\] \hspace{1cm} (3.125)

Let

\[
K_{34i}^{-1} = \begin{bmatrix}
k_{11} & k_{12} \\
k_{21} & k_{22}
\end{bmatrix}
\] \hspace{1cm} (3.126)

Then

\[
\begin{bmatrix}
\dot{\omega}_{Si} \\
\dot{\omega}_{Cr}
\end{bmatrix} = \begin{bmatrix}
k_{11} & k_{12} \\
k_{21} & k_{22}
\end{bmatrix} \begin{bmatrix}
a_{21} \cdot T_t - RT_{C4} \\
a_{22} \cdot T_t - T_s \cdot R_D
\end{bmatrix}
\] \hspace{1cm} (3.127)

\[
RT_{C2} = T_t - I_t \left( a_{21} \cdot \dot{\omega}_{Si} + a_{22} \cdot \dot{\omega}_{Cr} \right)
\] \hspace{1cm} (3.128)

\[
T_{Si} = \left( a_{11} \cdot I_{Sr} + a_{21} \cdot (I_t + I_{Rr}) \right) \cdot \dot{\omega}_{Si} + \left( a_{11} \cdot a_{12} \cdot I_{Sr} + a_{21} \cdot a_{22} \cdot (I_t + I_{Rr}) \right) \cdot \dot{\omega}_{Cr} - a_{21} \cdot T_t
\] \hspace{1cm} (3.129)
\[ RT_{C4} = -\left( I_{Si} \cdot \omega_{Si} + T_{Si} \right) \] (3.130)

When \( \omega_{Si} = 0 \) and \( |T_{C4}| > |RT_{C4}| \), the C4 clutch is locked up and the transmission is in the fourth gear.

3.4.2.7 Dynamic model of the transmission in the fourth gear

When the transmission is in fourth gear, C2 and C4 are locked up. Clutch C3 is overrunning. We can use the same freebody diagrams as those in Figure 3.9 with the following conditions,

\[ \omega_{Si} = 0 \] (3.131)
\[ RT_{sp3} = 0 \] (3.132)

The dynamic equations are:

\[ (I_{t} + I_{Rr}) \cdot \dot{\omega}_{t} = T_{t} - T_{f1} \] (3.133)
\[ -RT_{C4} = T_{Si} \] (3.134)
\[ I_{Sr} \cdot \dot{\omega}_{Sr} = -T_{f2} \] (3.135)
\[ I_{Cr} \cdot \dot{\omega}_{Cr} = T_{o} - R_{D} \cdot T_{s} \] (3.136)

Solving these equations:

\[ \begin{pmatrix} \omega_{Sr} \\ \omega_{Cr} \end{pmatrix} = F \begin{pmatrix} \omega_{Rr} \\ \omega_{Si} \end{pmatrix} \] (3.137)
\[ \begin{pmatrix} T_{f1} \\ T_{Si} \end{pmatrix} = -F^{T} \begin{pmatrix} T_{f2} \\ -T_{o} \end{pmatrix} \] (3.138)

\[ T_{f1} = \left( f_{11}^{2} \cdot I_{Sr} + f_{21}^{2} \cdot I_{Cr} \right) \cdot \dot{\omega}_{t} + f_{21} \cdot R_{D} \cdot T_{s} \] (3.139)
\[ (I_{t} + I_{Rr} + f_{11}^{2} \cdot I_{Sr} + f_{21}^{2} \cdot I_{Cr}) \cdot \dot{\omega}_{t} = T_{t} - f_{21} \cdot R_{D} \cdot T_{s} \] (3.140)

Defining

\[ I_{t5} = I_{t} + I_{Rr} + f_{11}^{2} \cdot I_{Sr} + f_{21}^{2} \cdot I_{Cr} \] (3.141)
\[ I_{16} = f_{11} \cdot f_{12} \cdot I_{S_1} + f_{21}^2 \cdot I_{C_2} \]  

Then

\[ RT_{C_4} = -I_{16} \cdot \omega_i + f_1 \cdot R_D \cdot T_3 \]  
\[ RT_{C_2} = T_i - I_i \cdot \omega_i \]

3.4.2.8 Transmission mechanical system model overview

Figure 3.10 shows the overview of the Simulink block diagram of the transmission mechanical system. The simulator described here includes power-on first-to-fourth gear upshifts. The downshifts are not implemented.

The downshift sequences are different from those for upshifts. There are two phases involved for each down shift – the inertia phase occurs first followed by the torque phase. During the inertia phase, the speed change occurs and during the torque phase, the load transfer occurs. There are two types of load transfer during downshifts for the transmission of interest. The first type is the clutch-to-one way clutch load transfer as in the fourth → third and second → first gear power-on downshifts. The second type is the clutch-to-band clutch load transfer as in the third → second gear power-on downshift.

Using the second → first gear power-on downshift as an example, the two clutches involved are \( C_1 \) and \( C_2 \). When the shift schedule logic module initiates a downshift, the \( C_2 \) clutch is depressurized and \( T_{C_2} \) starts decreasing. When \( T_{C_2} \) can no longer hold the \( C_2 \) clutch, the \( C_2 \) clutch starts slipping and the inertia phase starts. During the inertia phase of the second → first downshift, the reaction ring assembly is decelerated, i.e., \( \omega_{r_r} \) decreases, and the turbine speed \( \omega_i \) increases. During the inertia phase, the clutch \( C_1 \) is pressurized. When \( \omega_i \) equals \( \omega_{s_i} \) (the input sun assembly speed), the torque phase starts. During the torque phase of the second → first gear downshift, the load carried by the \( C_2 \) clutch is transmitted to the \( C_1 \) clutch. When \( C_1 \) clutch develops sufficient capacity, \( T_{C_1} \) will become greater than \( RT_{sp1} \), then clutch \( C_1 \) is locked up, and the transmission is in first gear.
Figure 3.10 Overview Simulink block diagram of the transmission mechanical system
3.5 Transmission shift hydraulic system model

The hydraulic system of a transmission can be separated into two parts: the regulated hydraulic supply and the hydraulic load. The modeling of the shift hydraulic system in this chapter focuses on modeling the clutch-accumulator assembly dynamic response. More detailed modeling of the clutch filling process controlled by a solenoid valve is carried out as part of the study of neutral-idle shifts described subsequently in the thesis. The information about the transmission shift hydraulic system modeled here is obtained from several sources, including Tugcu et al. (1986), Karmel (1986; 1988a; 1988b), and communication with GM.

Section 3.5.1 concerns the clutch and band pressure profiles used here to test the powertrain simulation without including explicit simulation of shift hydraulic system dynamics. Section 3.5.2 describes the details of clutch-accumulator assembly modeling for a specific hydraulic network.

3.5.1 Clutch and band pressure profiles

The clutch pressure profiles used in the simulation here are adapted from Cho and Hedrick (1989) and were originally curve-fitted to empirical data. The pressure profile during pressurizing is given by

\[ P_{\text{ci}} = 1000 \cdot (1 - e^{-t/0.045}) \text{ KPa} \]  \hspace{1cm} (3.145)

which corresponds to a line pressure of 1000 kPa. The band pressure profile is given by a similar formula with an exponential decay of band pressure to zero. The level of the band pressure used is adjusted in the simulation due to the lack of reported data in the literature. The depressurizing band pressure profile is given by:

\[ P_{b12} = \begin{cases} P_{b0} & t \leq t_3 \\ P_{b0} \cdot e^{-(t-t_3)/\tau} & t \geq t_3 \end{cases} \]  \hspace{1cm} (3.146)

where \( P_{b0} \) is the band pressure level, 200 kPa
\[ t \] is the time (sec)
\[ t_3 \] is the time instant corresponding to depressurizing of the band \( B_{12} \), and its value is determined by shift logic.

\[ \tau \] is the time constant of the pressure profile, 0.004 ms.

3.5.2 Clutch/accumulator assembly model

Some important aspects of clutch pressure response are missing from the pressure profiles mentioned above. For example, filling of the clutch cavity results in a time delay in the clutch pressure response. In addition, the nonlinearities associated with the clutch spring rate and finite limits of the clutch and accumulator piston strokes are not accounted for. In this section, the detailed modeling of a clutch-accumulator assembly in a specific transmission shift hydraulics network is given.

A schematic representation of a specific transmission shift hydraulic pressure network is shown in Figure 3.11 (Tugcu et al., 1986). Accumulators are used in the hydraulic network. They have two important functions. First, accumulators allow for a more gradual rise in clutch cavity pressure after the clutch release spring is compressed, by allowing the accumulator cavity to continue filling. Without accumulators, the clutch cavity pressure will rise sharply to the line pressure once the clutch spring is fully compressed. Such a sharp pressure transient would result in the clutch engaging abruptly and causing a harsh shift. Second, the backpressure on the accumulator depends on a measure of engine torque and allows the clutch pressure dependent on the operating conditions (Rizzoni and Srinivasan, 1998).

The transient behavior of clutch hydraulic pressures during shifts is a function of the design and operation of the shift hydraulic system. A schematic representation of the second clutch-accumulator assembly is shown in Figure 3.12, and represents a modified version of the circuit described by Tugcu et al. (1986). As shown in the figure, the 1-2 accumulator is connected with the 2\(^{nd}\) clutch in parallel. The back pressure \( P_{a2} \) is controlled independently by solenoid actuation, instead of being mechanically manipulated as indicated in Figure 3.11.
Figure 3.11 Hydraulic network (Tugcu et al., 1986)
We describe below the derivation of the dynamic equations for the C_2 clutch and the accumulator involved in the 1-2 shift. Neglecting the clutch plate mass and applying Newton's Second Law, we get,

\[ B_{c2} \cdot x_{c2} + F_{cs} + F_{e2} = P_{c2} \cdot A_{c2} \]  \hspace{1cm} (3.147)

where

- \( B_{c2} \) is the viscous damping coefficient for the C_2 clutch piston motion, 2000 \( \frac{N}{m/s} \);
- \( A_{c2} \) is the clutch piston area, \( 9.5411 \times 10^{-3} \) m\(^2\);
- \( F_{cs} \) is the clutch release spring force,

\[ F_{cs} = F_{c2o} + K_{c2} \cdot x_{c2} \]  \hspace{1cm} (3.148)

- \( F_{c2o} \) is the spring preload at installation, 972.5 N,
- \( K_{c2} \) is the clutch return spring rate, \( 111.4 \times 10^3 \) N/m,
- \( x_{c2} \) is the C_2 clutch spring deflection from installed height. When

\[ P_{c2} \cdot A_{c2} \leq F_{c2o} , \]

\[ x_{c2} = 0 \]  \hspace{1cm} (3.149)
\[ x_{c2\text{max}} = \max(x_{c2}) = 4.61\, \text{mm} \tag{3.150} \]

\( F_{c2} \) is the force exerted on the clutch piston by the clutch pack, when \( x_{c2} \leq x_{cp2} \),

\[ F_{c2} = \begin{cases} 0 & \text{if } 0 < x_{c2} \leq x_{cp2} \\ K_{w2} \cdot (x_{c2} - x_{cp2}) & \text{if } x_{cp2} < x_{c2} \leq x_{c2\text{max}} \end{cases} \tag{3.151} \]

\( x_{cp2} \) is the return spring stroke when the wave plate starts stroking, which is also when the load transfer starts, 2.941 mm.

\( K_{w2} \) is the wave spring rate, 5408 N/mm.

As shown above, the equations of clutch dynamic response are nonlinear due to the presence of two clutch springs (return spring and wave plate) with different spring rate, and the finite clutch piston stroke. In practice, the spring rate of wave plates is usually linear in the initial deflection, and nonlinear when the deflection exceeds a threshold. Here, we model the wave plate spring as a linear spring with higher stiffness. Care has to be taken when modeling clutch piston motions. In the first part of clutch stroke, the clutch return spring is the only spring involved. When the clutch stroke reaches \( x_{cp2} \), the clutch piston touches the wave plate, which is in parallel to the clutch return spring. The wave plate exerts force on the clutch packs, which transfers torque to the drivetrain.

For the 1-2 accumulator,

\[ B_{a2} \cdot x_{a2} + K_{a2} \cdot x_{a2} + F_{a2o} = (P_{c2} - P_{a2}) \cdot A_{a2} \tag{3.152} \]

where \( B_{a2} \) is the viscous damping coefficient for the 1-2 accumulator piston motion, \( 500 \, \frac{N}{m/s} \),

\( K_{a2} \) is the 1-2 accumulator spring stiffness, assumed to be constant, \( 7.4167 \times 10^3 \, \text{N/m} \),

\( A_{a2} \) is the 1-2 accumulator piston area, \( 1.945 \times 10^{-3} \, \text{m}^2 \), and
$P_{a2}$ is the 1-2 accumulator back pressure, assumed to be independently manipulated by a solenoid operated valve,

$F_{a2o}$ is the accumulator spring preload at installed height, 654 N,

$x_{a2}$ is the 1-2 accumulator spring deflection from its installed height,

$x_{a2\text{max}}$ is the maximum accumulator piston stroke, 30 mm.

The volumetric flow rate of fluid into the 2nd clutch cavity, $Q_{c2}$, and the volumetric flow rate into the 1-2 accumulator chamber, $Q_{a2}$, are,

$$Q_{a2} = A_{a2} \cdot x_{a2}$$

$$Q_{c2} = A_{c2} \cdot x_{c2}$$

The clutch cavity pressure $P_{c2}$ changes depending on the inflow into, and outflows from the clutch cavity, and the compressibility of the fluid.

$$\dot{P}_{c2} = \frac{\beta}{V_{\text{fin}}} (Q^{\text{in}}_{c2} - Q_{a2} - Q_{c2})$$

where

$\beta$ is the fluid bulk modulus, 70 MPa,

$V_{\text{fin}}$ is the volume of fluid at pressure $P_{c2}$, and is given by

$$V_{\text{fin}} = \int_{0}^{t} Q^{\text{in}}_{c2} \, dt$$

$Q^{\text{in}}_{c2}$ is the volumetric flow rate of fluid into the clutch-accumulator assembly, and is given by,

$$Q^{\text{in}}_{c2} = C_d \cdot A_{i2} \sqrt{\frac{2(P_i - P_{c2})}{\rho}}$$

$A_{i2}$ is the orifice area governing inflow into the 2nd clutch-accumulator assembly, $2.6273 \times 10^{-6}$ m$^2$. 

154
$P_t$ is the line pressure, 1000 kPa,
$\rho$ is the fluid density, 840 kg/m$^3$, and
$C_d$ is the orifice discharge coefficient, 0.61.

Limits on spring deflection are included in the simulation along with the appropriate logic. Similarly, the initial cavity volume is included in the simulation along with appropriate logic to represent the initial filling operation properly.

3.6 Drivetrain and longitudinal vehicle dynamics model

Vehicle dynamics is a subject in its own right and many texts are devoted to it (Wong, 1978). For the purpose of studying powertrain dynamics and control issues, we focus on the longitudinal dynamic behavior of a mid-size vehicle and couple this dynamic model with the powertrain model developed. Two types of vehicle and drivetrain models are used here. The first one is derived from Runde (1986), and is termed the gross vehicle dynamics model. The second one is derived from Cho and Hedrick (1989) and includes tire-road interaction effects.

3.6.1 Gross vehicle dynamics model

Figure 3.13 shows the schematic of a simple vehicle dynamics model considering the driveline to be a shaft with torsional spring constant $k$ connected to a lumped inertia. This is the gross vehicle dynamics model.

The vehicle dynamics model equations are

$$\dot{T}_s = k \cdot (\omega_s - \omega_v)$$  \hspace{1cm} (3.158)  
$$\dot{\omega}_v = \frac{1}{I_v} (T_i - T_L)$$  \hspace{1cm} (3.159)
where $I_v$ is the equivalent moment of inertia of a vehicle,

$$I_v = m \cdot r^2 \quad (3.160)$$

$m$ is vehicle mass, 1644 Kg,

$r$ is tire radius, 0.3214 m,

$T_L$ is the load torque due to wind and rolling resistance,

$$T_L = \left(158.2 + 4.479 \times 10^{-2} \cdot V_{kph}^2\right) \cdot r \quad (3.161)$$

$V_{kph}$ is the vehicle velocity in kilometers per hour,

$\omega_o$ is the output shaft speed of the transmission gear train,

$\omega_v$ is the wheel angular velocity,

$T_s$ is the output shaft torque of the transmission gear train, and

$k$ is the output shaft compliance, 7625 Nm/rad.

One simplification of the vehicle dynamic model is to assume a stiff output shaft, i.e., $K_s = \infty$. This simplification is useful for studying shift dynamics without considering shaft compliance. In this case, the dynamic equations become,

$$\dot{\omega}_v = \frac{T_s - T_L}{I_v} \quad (3.162)$$
3.6.2 Vehicle dynamics model with tire-road interaction

The longitudinal vehicle dynamics model including tire-road interactions is derived from Cho and Hedrick (1989). The final drive output shaft speed is the input to the differential and the axle shafts. Following the freebody diagrams in Figure 3.6.2, we get the model equations as follows. The output shaft dynamics are described by

\[ T_s = K_s \cdot (\omega_o - \omega_{wf}) \]  (3.163)

The driving front wheels are described by,

\[ I_{wf} \cdot \dot{\omega}_{wf} = T_s - h_f \cdot F_f - T_{rf} \]  (3.164)

The driven rear wheels are described by,

\[ I_{wr} \cdot \dot{\omega}_{wr} = h_r \cdot F_{tr} - T_{rr} \]  (3.165)

and the longitudinal dynamics of the vehicle body are described by,

\[ M \ddot{V} = F_f - F_{tr} - F_a - F_g \]  (3.166)

![Freebody diagrams for vehicle dynamics model](image)

Figure 3.14 Freebody diagrams for vehicle dynamics model (Cho, 1987)
where

\[ I_{wf} \] combined inertia of right and left front wheels, 2.8 Kgm²,

\[ I_{wr} \] combined inertia of right and left rear wheels, 2.8 Kgm²,

\[ h_f \] static ground-to-axle height of the front wheels, 0.31 m,

\[ h_r \] static ground-to-axle height of the rear wheels, 0.315 m,

\[ F_{tf} \] combined front tire force,

\[ F_{tr} \] combined rear tire force,

\[ T_{tf} \] combined front tire rolling resistance torque,

\[ T_{tr} \] combined rear tire rolling resistance torque (= \( T_{tf} \)),

\[ g \] acceleration due to gravity, 9.807 m/s²,

\[ F_a \] aerodynamic drag force

\[ F_a = C_a \cdot V^2 = 0.4298 \times V^2 \text{[N]} \] (3.168)

\[ C_a \] aerodynamic drag coefficient, 0.4298 N/(m/s)²,

\[ F_g \] road grade resistance.

\[ F_g = M \cdot g \cdot \sin \theta \] (3.169)

\[ \theta \] road grade, and

\[ M \] vehicle mass, 1644 Kg.

Tire forces \( F_{tf}, F_{tr} \) are usually given as linear functions of tire slip for low slip rates less than 15 ~ 20% (Wong, 1978). The tire force-slip relationship is usually determined empirically. In the linear range,

\[ F_{tf} = K_i \cdot i_f \] (3.170)

\[ F_{tr} = K_i \cdot i_r \] (3.171)

where \( i_f \) and \( i_r \) are the front and rear tire slip, \( K_i = 80000 \). There are many definitions of tire slip. Here, we will use the definitions given by Wong (1978). For a tire under driving torque, slip is defined as...
and for a tire under braking torque, slip (skid) is defined as

\[ i = \left(1 - \frac{r \cdot \omega}{V}\right) \times 100\% = \left(1 - \frac{r_e}{r}\right) \times 100\% \]  

(3.173)

where \( r \) is the tire free-rolling radius and \( r_e \) is the effective rolling radius of the tire, which is the ratio of the translatory speed of the tire center to the angular speed of the tire.

3.7 Model integration

The subsystem models are integrated to form a complete powertrain simulation – the Powertrain Dynamics Simulator shown in Figure 3.2. Each subsystem is represented by a Simulink block. Double clicking on each block yields the next layer of simulation details for that subsystem in a different Simulink window. The whole model thus has several layers, with different levels of detail shown in different layers. Such a configuration can give an overview of the model structure as well as design details inside each subsystem, which is very helpful for design engineers. The input and output variables for each subsystem are listed in Table 3.3.

The current powertrain simulator can simulate power-on upshifts from the first gear to the fourth gear. From the model equations of the transmission mechanical system, we can see that when the shift occurs, the energy flow path through the system is changed. This change of energy flow paths creates difficulties in developing simulation models. This is solved by creating a "pseudo" shifting status signal or "flag", which is used to monitor the shift process and generate a trigger to indicate to the simulation when to switch to a different subsystem. The "flag" signal is defined in Table 3.4.

Proper initialization of the simulation is important. Appropriate initial conditions are determined based on the steady operating conditions. All initial conditions are derived based on inverse models assuming the vehicle is moving at a constant speed. The procedure is programmed into an initialization file.
<table>
<thead>
<tr>
<th>Subsystems</th>
<th>Inputs</th>
<th>Outputs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Throttle Angle $\alpha$</td>
<td>Engine Speed $\omega_e$</td>
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<tr>
<td></td>
<td>Pump Torque $T_p$</td>
<td></td>
</tr>
<tr>
<td>Torque Converter</td>
<td>Engine Speed $\omega_e$</td>
<td>Pump Torque $T_p$</td>
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<tr>
<td></td>
<td>Turbine Speed $\omega_t$</td>
<td>Turbine Torque $T_t$</td>
</tr>
<tr>
<td>Transmission Mechanical System</td>
<td>Turbine Torque $T_t$</td>
<td>Gear Speeds</td>
</tr>
<tr>
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<td>Clutch/Band Torques $T_{cl} / T_{bj}$</td>
<td>($\omega_{si}, \omega_{sr}, \omega_{sr}, \omega_{cr}$) and</td>
</tr>
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<td></td>
<td>Output Shaft Torque $T_s$</td>
<td>Turbine Speed $\omega_t$</td>
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<td></td>
<td>Shifting Status – flag</td>
<td>Shifting Status – flag</td>
</tr>
<tr>
<td>Shift Hydraulic System</td>
<td>(Shifting Status – flag)</td>
<td>Clutch/Band Torque $T_{cl} / T_{bj}$</td>
</tr>
<tr>
<td></td>
<td>Gear Speeds ($\omega_{si}, \omega_{sr}, \omega_{sr}, \omega_{cr}$)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Turbine Speed $\omega_t$</td>
<td></td>
</tr>
<tr>
<td>Driveline Model</td>
<td>Stiff Output Shaft Case $K_s = \infty$</td>
<td>Wheel speed $\omega_w$</td>
</tr>
<tr>
<td></td>
<td>Output Shaft Torque $T_s$</td>
<td>And Vehicle Speed $V$</td>
</tr>
<tr>
<td>Compliant Output Shaft (Finite $K_s$)</td>
<td>Final Drive Output Speed $\omega_o$</td>
<td>Wheel speed $\omega_w$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>And Vehicle Speed $V$</td>
</tr>
<tr>
<td>Compliant Output Shaft with Tire-road Interactions</td>
<td>Final Drive Output Speed $\omega_o$</td>
<td>Output Shaft Torque $T_s$ ,</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Wheel Speeds $\omega_{nf}, \omega_{wr}$ ,</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Vehicle Speed $V$</td>
</tr>
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Table 3.3 Inputs and outputs of subsystems
<table>
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<th>flag</th>
<th>Shifting Status</th>
<th>Conditions</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>1st Gear</td>
<td>((T_{cl} &gt; RT_{sp1} &gt; 0) \cap (</td>
</tr>
<tr>
<td>1.2</td>
<td>Torque Phase of 1st - 2nd Gear Shift</td>
<td>((T_{cl} &gt; RT_{sp1} &gt; 0) \cap (</td>
</tr>
<tr>
<td>1.7</td>
<td>Inertia Phase of 1st - 2nd Gear Shift</td>
<td>((RT_{sp1} = 0) \cap (</td>
</tr>
<tr>
<td>2</td>
<td>2nd Gear</td>
<td>((RT_{sp1} = 0) \cap (</td>
</tr>
<tr>
<td>2.2</td>
<td>Torque Phase of 2nd - 3rd Gear Shift</td>
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<tr>
<td>2.7</td>
<td>Inertia Phase of 2nd - 3rd Gear Shift</td>
<td>((T_{C2} &gt; RT_{C3}) \cap (</td>
</tr>
<tr>
<td>3</td>
<td>3rd Gear</td>
<td>((T_{C2} &gt; RT_{C3}) \cap (</td>
</tr>
<tr>
<td>3.2</td>
<td>Torque Phase of 3rd - 4th Gear Shift</td>
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</tr>
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<td>Inertia Phase of 3rd - 4th Gear Shift</td>
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</tr>
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<td>4.0</td>
<td>4th Gear</td>
<td>((T_{C2} &gt; RT_{C3}) \cap (</td>
</tr>
</tbody>
</table>

Table 3.4 Shifting Status Signal
3.8 Simulation results for power-on upshift

Simulation results for power-on upshifts are presented in three parts. Section 3.8.1 presents the results for first-to-fourth gear power-on upshifts assuming the simplified clutch pressure profiles given in section 3.5.1. Section 3.8.2 presents the simulation results for first-to-second gear upshifts with the clutch-accumulator assembly model. Section 3.8.3 presents the simulation results for first-to-second gear upshifts with the compliant shaft model.

3.8.1 First-to-fourth gear power-on upshifts

The simulation results for power-on upshifts from the 1st gear to the 4th gear with the "stiff output shaft" model are shown in Figure 3.15 – Figure 3.22. The simulation assumes the following conditions:

(1) the vehicle starts moving at 8 kmph,
(2) the vehicle accelerates at wide open throttle,
(3) the 1st - 2nd gear upshift occurs when the vehicle speed reaches 24 kmph,
(4) the 2nd - 3rd gear upshift occurs when the vehicle speed reaches 40 kmph,
(5) the 3rd - 4th gear upshift occurs when the vehicle speed reaches 56 kmph, and
(6) the line pressure is 1000KPa, and the clutch pressure profiles follow those shown in section 3.5.1.

Other parameter values are given in Appendix A.

Figure 3.15 shows traces of the engine speed \( \omega_e \), the turbine speed \( \omega_t \), the input sun gear speed \( \omega_{si} \), the reaction ring gear speed \( \omega_r \), and the reaction sun gear speed \( \omega_{sr} \) during the gear shifts from the 1st gear to the 4th gear. The "flag" variable is also shown in the same figure indicating the shift status. In the first gear, \( \omega_i \) is equal to \( \omega_{si} \). During the 1st - 2nd gear upshift, the input sun gear assembly is decelerated and the reaction ring assembly is accelerated. In the 2nd gear, clutch \( C_2 \) is locked up and \( \omega_i \) is equal to \( \omega_r \). During the 2nd - 3rd gear upshift, the band \( B_{12} \) is released and the reaction sun gear
Figure 3.15 Turbine and subassembly speeds during 1st-4th gear upshifts
assembly is accelerated. During the 3\textsuperscript{rd} \sim 4\textsuperscript{th} gear upshift, the clutch C\textsubscript{4} is applied. Therefore, the input sun gear speed $\omega_{nu}$ becomes zero in the 4\textsuperscript{th} gear.

Figure 3.16 shows the transmission output shaft torque. The transient torque peak level is largest during the 1\textsuperscript{st} \sim 2\textsuperscript{nd} gear upshift and is smallest during the 3\textsuperscript{rd} \sim 4\textsuperscript{th} gear upshift. The transient magnitudes are functions of the assumed clutch pressure profiles. The clutch pressure profiles assumed in section 3.5.1 are rather rapid and give rise to shorter and more severe transients than would usually be the case. The effects of softening the pressure transients on the torque transients will be investigated in the following section. Figure 3.17 shows the torque converter turbine and pump torques $T_t$ and $T_p$. As shown in the figure, the torque converter operates in the torque amplification mode most of the time. Figure 3.18 shows the vehicle speed as it shifts from the 1\textsuperscript{st} to the 4\textsuperscript{th} gear.

Figure 3.19 shows the torque transfer during the 1\textsuperscript{st} \sim 2\textsuperscript{nd} gear upshift. As shown in the figure, the torque transfer occurs between the clutches C\textsubscript{1} and C\textsubscript{2}. During the torque phase, C\textsubscript{2} takes over load from C\textsubscript{1}. This is shown by the reaction torque traces $RT_{C2}$ and $RT_{sp}$. When $RT_{sp}$ goes to zero, the inertia phase starts. When the C\textsubscript{2} clutch torque capacity $T_{C2}$ is greater than the reaction torque $RT_{C2}$, C\textsubscript{2} is locked up and the transmission reaches the 2\textsuperscript{nd} gear.

Figure 3.20 shows the torque transfer during the 2\textsuperscript{nd} \sim 3\textsuperscript{rd} gear upshift. As shown in the figure, the torque is transferred from the band B\textsubscript{12} to the clutch C\textsubscript{3}. When the band reaction torque $RT_{B12}$ is equal to the torque capacity $T_{B12}$, the band starts slipping and the inertia phase starts. In this simulation, the band B\textsubscript{12} is depressurized shortly after the inertia phase starts. In practice, this relative timing of the actuation of the two friction elements is determined by calibration so that the shift quality is acceptable. The consequences of actuating the second element, i.e. B\textsubscript{12}, too soon or too late are illustrated by Figure 3.21.
Figure 3.21 (a) shows the normal case of the torque patterns during the 2\textsuperscript{nd} \sim 3\textsuperscript{rd} gear upshift. From time $t_1$ to $t_4$, there is an overlap of the band $B_{12}$ and the clutch $C_3$. Overlap is unavoidable, because "underlap" (Figure 3.21 (c)) is unacceptable. The ideal case would be (b), which shows the clutch $C_3$ just develops enough capacity when the band $B_{12}$ is fully depressurized. But in reality, this can not be the case. Therefore, overlap is unavoidable. Underlap shown in Figure 3.21 (c) indicates that the clutch $C_3$ has not developed enough capacity to take over the load from the band $B_{12}$ at time $t_1$, and the band $B_{12}$ is still locked up, which is not acceptable.

Figure 3.22 shows the torque transfer during the 3\textsuperscript{rd} \sim 4\textsuperscript{th} gear upshift. The load is transferred from the one-way clutch $C_3$ to clutch $C_4$, which is similar to that of the 1\textsuperscript{st} \sim 2\textsuperscript{nd} gear upshift. When the reaction torque $RT_{sp3}$ goes to zero, the inertia phase starts. When the $C_4$ torque capacity $T_{C4}$ is greater than the reaction torque $RT_{C4}$, clutch $C_4$ is locked up and the transmission is in the fourth gear.

![Figure 3.16 Transmission output shaft torque $T_s$](image)

Figure 3.16 Transmission output shaft torque $T_s$
Figure 3.17 Torque converter turbine and pump torques $T_t$ and $T_p$

Figure 3.18 Vehicle velocity $V$ (kmph)
Figure 3.19 Torque transfer during 1st ~ 2nd gear upshift

Figure 3.20 Torque transfer during 2nd ~ 3rd gear upshift
Figure 3.21 Torque patterns during 2\textsuperscript{nd} ~ 3\textsuperscript{rd} gear upshift
3.8.2 First-to-second gear upshift including hydraulic system simulation

The simulation results in the previous section are based on the simplified hydraulic pressure profiles given in section 3.5.1. In this section, the simulation results for the first-to-second gear upshift including a detailed model of the clutch-accumulator assembly dynamic behavior are presented.

The accumulator backpressure $P_{a2}$ is assumed to be an input. Two levels of $P_{a2}$ are examined. In addition, two types of clutch plate friction characteristics given in equation (3.174) below are used in the simulation to evaluate the effect of friction on shift quality.
\[
\mu(\Delta \omega_{C2}) = \begin{cases} 
0.08 - 0.02e^{-0.01|\Delta \omega_{C2}|} & \text{TypeA} \\
0.0631 + 0.0504e^{-0.033|\Delta \omega_{C2}|} & \text{TypeB}
\end{cases}
\]  

(3.174)

where \( \mu(\Delta \omega_{C2}) \) is the coefficient of friction. The \( \mu \) versus slip speed curves for the TypeA and TypeB friction characteristics are shown in Figure 3.23.

![Friction characteristics](image)

Figure 3.23 TypeA and TypeB friction characteristics

Simulation results for different levels of the accumulator back pressure \( P_a \) together, with TypeA friction characteristics, are shown in Figure 3.24 – Figure 3.29. Figure 3.24 – Figure 3.26 correspond to a pressure level of 100 kPa, whereas Figure 3.27 – Figure 3.29 correspond to 180 kPa. Shaft output torques are shown in Figure 3.25 and Figure 3.28. The relevant speeds are shown in Figure 3.24 and Figure 3.27. A flag indicating the shift status of the transmission is also noted on the figures, the four levels
indicating, starting from the lowest level, 1\textsuperscript{st} gear, 1-2 torque phase, 1-2 inertia phase, and 2\textsuperscript{nd} gear respectively. As expected, the shaft torque drops during the torque phase as a result of the drop in the gear ratio, and rises subsequently during the inertia phase.

The clutch pressure transients in Figure 3.26 and Figure 3.29 show the sharp initial rise in pressure that results from the spring preload. The manner of the subsequent rise in clutch pressure is governed by the effective spring stiffness in the system and details of clutch construction. The effective clutch spring is modeled as having a higher stiffness for deflections above a specified threshold, and smaller stiffness below this threshold. Details are given in section 3.5.2. The lower initial spring stiffness results in the slower rise of clutch pressure in Figure 3.26 and Figure 3.29, followed by the faster rise of clutch pressures corresponding to the higher stiffness. The piston moves a finite distance prior to engaging the clutch pack and initiating a load transfer. At high enough clutch pressures, the accumulator spring preload is overcome and the accumulator starts stroking, resulting again in a slower clutch pressure rise. When the accumulator stops stroking, the clutch pressure rises sharply to the line pressure level, in this case, 1000 kPa. As noted by the transmission shift status flag in the figures, the inertia phase is completed during the accumulator stroke. The completion of the shift during the accumulator stroke is an essential feature of a shift with acceptable quality. It should also be noted that the shift completion times noted here are more realistic than those corresponding to the assumed clutch pressure profiles in section 3.5.1.

Figure 3.24 and Figure 3.27 show engine and transmission speeds before, during, and after the shift. Before the shift, clutch C\textsubscript{1} is locked and the turbine and the input sun gear move at the same speed. During the inertia phase of the shift, the turbine is decelerated and the reaction ring gear assembly is accelerated until the two speeds become equal. After the shift, the turbine and the reaction ring gear assembly move together at the same speed.

Comparing the shift transients for different levels of the accumulator back pressure $P_{a2}$, we note that increasing the accumulator back pressure lowers the inertia
phase duration from about 0.69 second to 0.31 second, by elevating the clutch pressure $P_{c2}$ and the clutch torque $T_{c2}$. The shaft torque $T_s$ is also higher during the inertia phase if the accumulator back pressure is higher.

Change in the clutch plate friction characteristics changes the shift transients significantly, as expected. The Type B friction characteristics described in equation (3.174) results in the transients noted in Figure 3.30—Figure 3.32, other conditions in the simulation being the same as those for Figure 3.24—Figure 3.26. The coefficient of friction increases as the slip speed decreases in this case, and the resulting higher coefficient of friction results in a shorter inertia phase of about 0.30 second. It also results in a higher shaft torque prior to lockup and a sharper drop in torque at lockup.

![Figure 3.24 Engine, turbine, input sun gear and reaction ring gear speeds with Type A Friction, $P_{a2} = 100kPa$](image-url)
Figure 3.25 Shaft torque with Type A Friction and $P_{a2} = 100kPa$

Figure 3.26 Clutch pressure with Type A Friction and $P_{a2} = 100kPa$
Figure 3.27 Engine, turbine, input sun gear and reaction ring gear speeds with Type A friction characteristics and $P_{a2} = 180kPa$

Figure 3.28 Output shaft torque with Type A friction characteristics and $P_{a2} = 180kPa$
Figure 3.29 Clutch pressure with Type A friction characteristics and \( P_{a2} = 180kPa \)

Figure 3.30 Engine, turbine, input sungear and reaction ring gear speeds with TypeB Friction and \( P_{a2} = 100kPa \)
Figure 3.31 Shaft torque with TypeB friction characteristics and $P_{a2} = 100kPa$

Figure 3.32 Clutch pressure with TypeB friction and $P_{a2} = 100kPa$
Figure 3.33—Figure 3.35 show simulation results corresponding to different values of the fluid bulk modulus. The fluid bulk modulus changes with the amount of air entrained. Since the clutch empties and fills up between gear shifts, it is likely that some air may be trapped in the clutch cavity. The amount of trapped air would depend on the design of the clutch, especially as it relates to the manner of fluid filling of the clutch and the way the air is forced out of the clutch cavity. The simulation assumes TypeA friction characteristics for the clutch plate, and an accumulator backpressure of 100 kPa. The bulk modulus is reduced by 50% and 80% of the nominal value of 70 MPa respectively (The effective bulk modulus calculated from experiments in Chapter 5 is between 41 and 102.6 MPa). The output shaft torque does not show much change, in Figure 3.35. The clutch pressure response shows slightly different response in the final stages of the pressure rise, in Figure 3.33. During the earlier stages of pressure rise, the clutch pressure is almost the same under the three conditions. This is due to the fact that the hydraulic spring is much stiffer than the mechanical spring, even for the lower bulk modulus values considered. During the clutch and accumulator stroking periods, therefore, the mechanical spring compliance will dominate. Only after the accumulator reaches the end of its stroke is the compliance determined primarily by the hydraulic compliance. At the lower bulk modulus values, both the torque phase and the inertia phase are slightly longer than at the higher bulk modulus values, as shown in Figure 3.34.
Figure 3.33 Clutch pressure traces at different bulk modulus values

Figure 3.34 Shift duration traces at different bulk modulus values
Automatic transmission fluids operate over wide temperature ranges, such as -20 to 150 °C. When fluid temperature changes, the fluid viscosity changes and changes the damping between the clutch piston and cylinder. The fluid density also changes. In addition, the friction characteristics also change with temperature (Griffen, 1994). To study this effect, the following fluid property information (Kemp and Linden, 1990; Griffen, 1994) is programmed into the model. Among these properties, the dependence of clutch plate friction coefficient on temperature given in equation (3.180) is a modified version of that shown in Griffen (1994) to adapt to the particular clutch design in this chapter. The other properties are from (Kemp and Linden, 1990).

Fluid density at temperature T (°C):

\[ \rho_{ar, T} = (-6.1 \times 10^{-4} \times Temp - C + 0.8784) \times 1000 \text{ Kg/m}^3 \]  

(3.175)

Viscosity at temperature T (°C):
\[
\log_{10}\left(\frac{\gamma_{\text{ATF}} \times 10^6}{\rho}\right) = 10^{-3.146 \times \log_{10}(273.15 + \text{Temp}_{\text{C}}) + 8.044} - 0.7 \quad (3.176)
\]

Nominal temperature:

\[\text{Temp}_{\text{C norm}} = 63^\circ \text{C} \quad (3.177)\]

Clutch and accumulator viscous damping coefficient at temperature \(T\) (°C):

\[
B_{c2, \text{at时报Temp}_{\text{C}}} = B_{c2} \times \frac{\gamma_{\text{ATF, at时报Temp}_{\text{C}}}}{\gamma_{\text{ATF, at时报Temp}_{\text{C norm}}}} \quad (3.178)
\]

\[
B_{a2, \text{at时报Temp}_{\text{C}}} = B_{a2} \times \frac{\gamma_{\text{ATF, at时报Temp}_{\text{C}}}}{\gamma_{\text{ATF, at时报Temp}_{\text{C norm}}}} \quad (3.179)
\]

Clutch plate friction coefficient at temperature \(T\) (°C):

\[
\mu_{\text{at时报Temp}_{\text{C}}} = (-0.0001 \times \text{(Temp}_{\text{C}} \times 1.8 + 32) + 0.089) - 0.02e^{-0.01|\Delta\alpha|} \quad (3.180)
\]

The simulation results corresponding to a fluid temperatures of 63°C (the condition for all previous simulation results) and a fluid temperature of 150°C are compared in Figure 3.36 – Figure 3.39. Also shown in the figures are the simulation results corresponding to the nominal case, i.e. \(\text{Pa2}\) at 100 kPa and temperature at 63°C.

As shown in Figure 3.36, the accumulator-stroking phase is shorter at the higher temperature with \(\text{Pa2}\) at 200 kPa. The main difference is the clutch plate friction coefficient changes with temperature. As shown in Figure 3.39, the friction coefficient at the higher temperature is lower than that at the lower temperature. Figure 3.37 shows that the inertia phase is much longer at the higher temperature. Figure 3.38 compares the shaft torque traces corresponding to the two temperatures. The shaft torque peak at the higher temperature is due to the fact that clutch lockup occurs after the accumulator stroke ends, which is not acceptable. This comparison indicates that the change of friction coefficient characteristics with temperature is critical to shift quality.
Figure 3.36 Clutch pressure traces under different temperatures

Figure 3.37 Shift duration under different temperatures
Figure 3.38 Shaft torque under different temperatures

Figure 3.39 Clutch plate friction coefficient under different temperatures
3.8.3 First-to-second gear power-on upshifts with compliant shaft model

The simulation results presented in the previous sections assume a stiff shaft model. In this section, the simulation results corresponding to a compliant shaft drivetrain model will be presented. The simulation assumes the same conditions as those in section 3.8.1

Figure 3.40 - Figure 3.41 show the simulation results with compliant shaft model without tire-road interaction. Figure 3.40 shows the engine, turbine, input sun gear and reaction ring gear speeds, as well as vehicle speed during a power-on 1-2 upshift. The turbine speed trace is oscillatory due to the output shaft compliance effect. Figure 3.41 shows the output shaft torque, pump and turbine torques, and the load transfer between C2 clutch and C1 clutch. The oscillation of the output shaft torque is caused by the shaft compliance effect.

Figure 3.42 and Figure 3.43 show simulation results for a compliant shaft model with tire-road interaction. Figure 3.42 shows the engine, turbine, input sungear and reaction ring gear speeds, as well as tire slip. Figure 3.43 shows the output shaft torque, pump and turbine torques, and the clutch reaction torques for clutches C1 and C2. The oscillations in these two sets of simulations are the result of shaft compliance.

Figure 3.44 and Figure 3.45 show the comparison of the simulation results corresponding to the stiff shaft (SS) case, the compliant shaft case without tire-road interaction model (CSNT) and the compliant shaft case with tire-road interaction model (CST). Figure 3.44 shows the comparison of the output shaft torques and turbine speeds. When including shaft compliance in the simulation, the responses after lockup are oscillatory compared to the stiff shaft case, which is expected. Comparing the traces for the compliant shaft model with and without the tire-road interaction effect, the output shaft dynamic response without tire-road interaction has higher oscillation, which is expected. This is because tire-road interaction adds damping to the system. The response corresponding to the compliant shaft with tire-road interaction settles faster. Figure 3.45 compares the C2 and C1 reaction torques during the 1-2 upshift. The load is transferred
from the $C_1$ clutch to $C_2$ clutch during the torque phase. The torque spike in the $RT_{C2}$ traces are due to lockup of clutch $C_2$.

Figure 3.40 Simulation results for compliant shaft model (without tire-road interaction): speeds during 1-2 power-on upshift
Figure 3.41 Simulation results for compliant shaft model (without tire-road interaction): torques during 1-2 power-on upshift
Figure 3.42 Simulation results for compliant shaft model with tire-road interaction: speeds and tire slip during 1-2 power-on upshift
Figure 3.43 Simulation results for compliant shaft model with tire-road interaction: torques during 1-2 power-on upshift
Figure 3.44 Comparison of output shaft torque and turbine speed traces during 1-2 power-on upshift: stiff shaft (SS), compliant shaft without tire-road interaction (CSNT), and compliant shaft with tire-road interaction (CST)
3.9 Conclusion

In this chapter, a representative powertrain is described in detail, along with the equations constituting its dynamic model. A simulator is developed, using the Matlab/Simulink software package, to study powertrain dynamics with more emphasis on the automatic transmissions. Transmission response predicted by this dynamic model for power-on upshifts from the 1st to 4th gear is examined, and sample results from the simulations are presented here. Furthermore, a detailed hydraulic network model for the
1st to 2nd gear power-on upshift is derived and implemented. With this model, the effect of changes in fluid property on transmission performance is studied. The effect of changes in drivetrain and vehicle longitudinal dynamic behavior on transmission response is also evaluated. The simulator can be useful in gaining physical insight into how design features affect the dynamic response of the transmission, and how they may be modified appropriately to improve transmission performance during a shift. The powertrain model presented here provides a basis for the controller design to be presented in Chapter 4. It also provides a framework to study the neutral-idle control problem, which will be presented in Chapter 6.
CHAPTER 4

CONTROL STRATEGIES FOR SHIFT CONTROL IN AN AUTOMATIC TRANSMISSION

4.1 Introduction

Electronic control is being extended to an increasing number of automotive power train functions in order to improve their performance while retaining flexibility of control. Such control, if it is based on quantitative models of the systems and components being controlled, is clearly capable of better performance. It is no surprise to find, therefore, that the development of quantitative models of engine components and subsystems, and of control algorithms based on these models, has been the subject of much automotive research (Cook and Powell, 1988; Hrovat and Powers, 1988; Powell, 1993; Grizzle et al., 1991). By contrast, the development of quantitative models of automatic transmission response, and of model-based algorithms for the control of transmission functions, has lagged considerably. Integrated control of automotive power trains involves coordinated control of engine and transmission functions, and in turn requires improved understanding and control of both of these power train subsystems if its potential is to be fully realized.

Electronic control of transmission functions has often relied on open loop control. In this case, controller actions are arrived at by a lengthy calibration procedure, the calibration combining implicit modeling of transmission response with controller adjustments. There are, however, a few reported cases of transmission control which have
involved explicit modeling of transmission response, and of controller adjustment based on these explicit models. Hrovat and Powers (1988) have described closed loop control of speed ratio during a shift in a discrete ratio automatic transmission. They refer to some aspects of the transmission dynamic model, and to their derivation of a linearized version of this model for purposes of controller design, but do not give any details. Taniguchi and Ando (1991) have described the closed loop control of turbine speed during the inertia phase, for the automatic transmission in the Lexus LS400. The transmission model is described in some detail, and the controller gains are apparently arrived at by trial and error based on a computer simulation of transmission response, and experimental evaluation. Hebbale and Kao (1995) have described control of clutch-to-clutch shifts. They refer to the nonlinear nature of automotive power train dynamics, and avoid determination of linearized models for controller design. Instead, gradients of error variables with respect to the manipulated variables are determined on-line and used for controller adaptation. Osawa et al. (1995) have described procedures for determining linearized dynamic models, and their use in the design of a slip control system for the torque converter clutch in the Toyota A541E automatic transmission. The variation of the linearized dynamic models under different operating conditions is used to define a model error, which then forms the basis for a robust controller design procedure.

The focus of this chapter is on the development of transmission models suitable for devising improved control during a shift, as well as the development of control strategies for torque phase and inertia phase control. Closed loop control is capable of achieving inertia phase shift characteristics which are relatively invariant with variations in operating conditions such as engine power levels, or in transmission parameters such as transmission fluid viscosity. The use of integrated open loop control of the engine and transmission for improving torque phase response is investigated as well.

The organization of this chapter is as follows. The transmission response predicted by the dynamic model given in Chapter 3, for power-on up shifts from the 1st to 2nd gear under different operating conditions, is examined. Linearized models relating the controlled variable for inertia phase control, in this case turbine speed, to the manipulated variable, in this case accumulator back pressure, are determined numerically by
perturbation techniques applied to the nonlinear dynamic model. Insights into the linearized model dependence on design features of the transmission, as well as operating conditions, are offered. A closed loop controller design for the inertia phase of the 1st to 2nd gear shift is carried out and evaluated on the simulation. Finally, an integrated open loop control strategy for the torque phase of the 1st to 2nd gear shift is described and evaluated on the transmission.

4.2 Determination of linearized models for the inertia phase of the 1-2 upshift

The powertrain model presented in Chapter 3 is used here to develop the 1-2 power-on shift controller. Recall that the engine model has three states. The powertrain model we use here for controller design uses a one-state engine model, by assuming that external inputs to the engine and the engine indicated torque \( T_i \) are constant during the shift. Engine inertial and frictional effects are retained in the model. The three-element torque converter model used here is derived from Kotwicki (1982). Thus, fluid inertial effects are neglected. The inputs to the transmission mechanical system are turbine torque \( T_t \) and output shaft torque \( T_o \). The outputs are turbine speed \( \omega_t \) and output shaft speed \( \omega_o \). The transient behavior of clutch hydraulic pressures during shifts is a function of the design and operation of the shift hydraulic system.

Consistency of inertia phase transients can be achieved by closed loop control of the turbine speed, by manipulating the accumulator back pressure based on turbine speed feedback so that the turbine speed follows a specified trajectory accurately (Taniguchi and Ando, 1991). The specification of a desired turbine speed trajectory is equivalent to specifying a rate of change of turbine speed. This is consistent with the Hayabuchi et al. (1996) observation about the range of rates of turbine speed change that would be considered pleasing to the passenger. A dynamic model of turbine speed response to accumulator back-pressure manipulation would help in formal controller design. There are a number of phenomena which result in such a model being nonlinear for the powertrain of interest here and, in practice, for most powertrains.

Some of the nonlinearities are significant during the shift process. The effective spring stiffness of the clutch-accumulator assembly varies during the shift depending, for
example, on whether the accumulator spring is stroking or not, and the deflection level of the clutch release spring. The flow-pressure drop relation for the orifice governing fluid flow into the clutch is also a nonlinear one. The clutch plate friction coefficient is a nonlinear function of slip speed and, furthermore, the clutch torque $T_{c2}$ is a nonlinear function of the clutch pack force $F_{c2}$ and the friction coefficient as indicated in Chapter 3. The torque converter relationships between pump and turbine speeds and torques are also nonlinear.

Other nonlinear effects are much less significant. Vehicle wind resistance is a nonlinear function of vehicle speed, but may be easily modeled as a constant during a shift as vehicle speed does not change much during a shift. Simplifications in the engine model already noted have eliminated engine-related nonlinearities from consideration.

One approach to controller design for such a system is to use the nonlinear model explicitly, as done by Hrovat and Colvin (1996) for the simpler case of a torque converter bypass clutch slip control. The inverse of the nominal nonlinear actuator model is used in this case to determine the feedforward component of the control action. The approach here, however, is to determine linearized models under a variety of operating conditions, and to use these models subsequently as the basis for controller design. The procedures used for such linearization, and the results from the linearization are considered.

The SIMULINK software package provides procedural support for the numerical determination of linearized state space models for a system from its nonlinear simulation diagram. The user specifies the operating point about which the small-signal linearization is to be performed, as well as the perturbation levels in the state variables and inputs. The routine *linmod* performs the linearization and returns the parameters of a linear model. Features of the linear models thus obtained are examined here in terms of the frequency response $\omega_{s}/P_{a2}$, and the poles and zeros of the corresponding transfer function.

The simulation model is seventh order. The seven states are: $P_{c2}$, $\omega_{t}$, $\omega_{e}$, $x_{c2}$, $x_{a2}$, $V_{v}$ and $V_{fa}$, where

- $P_{c2}$ is the second clutch chamber pressure, Pa
- $\omega_{t}$ is the torque converter turbine speed, rad/sec
$\omega_e$ is the engine speed, rad/sec

$x_{c2}$ is the second clutch piston displacement, m

$x_{a2}$ is the second accumulator piston displacement, m

$V_v$ is the vehicle speed, m/s

$V_{in}$ is the fluid entering the clutch-accumulator chamber, m$^3$

The simplified powertrain model is simulated and the values of the state variables and inputs corresponding to the operating conditions of interest are determined prior to the application of the linearization procedure.

Sample results from the linearization are given here. Figure 4.1 compares the frequency responses of the linearized system for three operating conditions. The first, labeled '20%', corresponds to an instant of time 20% through the inertia phase for the case where the accumulator back pressure $P_{a2}$ is 100 kPa. The second curve on the figure corresponds to an instant of time halfway into the inertia phase for the same shift. The third curve on the figure corresponds to an instant of time 90% into the inertia phase for the same shift. Figure 4.2 compares the frequency responses of the linearized system for the '50%' case from Figure 4.1, and two other operating conditions, 50% into the inertia phase, for shifts with accumulator back pressures $P_{a2}$ of 180 kPa and 20 kPa respectively. The latter shift, with $P_{a2}$ of 180 kPa, is the same as the one in Figures 3.27 – 3.29.

The results shown in Figure 4.1 and Figure 4.2 indicate that the effective frequency response relating the turbine speed $\omega_t$ to the accumulator back pressure $P_{a2}$ varies little in amplitude ratio and phase shift during the inertia phase of a single shift, for the conditions tested. The variation in amplitude ratio is within about 2 decibels and that in phase shift within about 10°. The small variation in linearized transfer characteristics is a result of the design of the shift hydraulic system which includes an accumulator for introducing sizable compliance in the system. Furthermore, the linearization could be performed only over the portion of the inertia phase when the accumulator is stroking. Thus, the nonlinear response of the shift hydraulic system prior to the accumulator stroking is immaterial as far as the linearization is concerned.
Figure 4.1 Frequency response $\omega_1 / P_{a2}$ - $P_{a2} = 100$ kPa, 20%, 50% and 90% into inertia phase

Figure 4.2 Frequency response $\omega_1 / P_{a2}$ - $P_{a2} = 20$ kPa, 100kPa and 180 kPa (50% into inertia phase)
It is expected that, in shift hydraulic systems designed for active manipulation of the clutch hydraulic pressure by solenoid control, the variation in the linearized frequency response over the duration of the inertia phase would be greater. The extent of such variation would be indicated quantitatively by the procedure used here, and can form a rational basis for controller design to ensure effective closed loop control of the inertia phase. Thus, trial and error controller design procedure can be avoided.

The results of the linearization can also be viewed alternatively by considering how the poles and zeros of the linear transfer functions vary with operating conditions within a shift. When combined with model analysis, which reveals how different states participate in the different modes of the linearized system response, we can gain physical insight into how design features of the transmission affect the variation of its dynamic response.

Table 4.1 and Table 4.2 list the eigenvalues of the linearized models corresponding to the conditions shown in Figure 4.1 – Figure 4.2. Using modal analysis and examining the extent of participation of the different state variables in each mode as indicated by the corresponding eigenvector components, we conclude that eigenvalue E5 is dominated by the state $P_{C2}$, E2 is dominated by the state $V_{fin}$, and E1 is dominated by the state $V_{r}$. Eigenvalues E3 and E4 are dominated jointly by states $\omega_{e}$ and $\omega_{r}$. However, the eigenvalues E6 and E7 are not dominated by one or two states. All states contribute significantly to these two modes.

Modal analysis indicates also that four states are crucial to the dynamic behavior of the inertia phase over the frequency range of interest. Given the finite duration of the inertia phase, it is reasonable to estimate the lower bound on the frequency range of interest as 1 rad/sec. Furthermore, given the nature of the shift transients in Figures 3.24 – 3.32 and the high frequency attenuation noted in Figure 4.1 and Figure 4.2, the upper bound on the frequency range of interest may be placed at 100 rad/sec. Within this frequency range, the significant eigenvalues are E3, E4, E5 and E7. The states which contribute most significantly to the corresponding modes are $\omega_{e}$, $\omega_{r}$, $x_{c2}$ and $x_{a2}$. Thus, design changes in the associated inertias and spring stiffnesses would affect the linearized frequency response significantly.
<table>
<thead>
<tr>
<th>Friction Characteristics</th>
<th>Type A</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_{a2} ) = 100 kPa</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Linearization point (% into inertia phase)</th>
<th>20%</th>
<th>50%</th>
<th>90%</th>
</tr>
</thead>
<tbody>
<tr>
<td>E1</td>
<td>-0.0041</td>
<td>-0.0043</td>
<td>-0.0044</td>
</tr>
<tr>
<td>E2</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>E3</td>
<td>-43.3716</td>
<td>-37.4136</td>
<td>-31.9167</td>
</tr>
<tr>
<td>E4</td>
<td>-2.1609</td>
<td>-2.1753</td>
<td>-2.2706</td>
</tr>
<tr>
<td>E5</td>
<td>-6.86e4</td>
<td>-5.706e4</td>
<td>-4.697e4</td>
</tr>
<tr>
<td>E6</td>
<td>-407.814</td>
<td>-402.86</td>
<td>-397.42</td>
</tr>
<tr>
<td>E7</td>
<td>-0.6041</td>
<td>-0.5488</td>
<td>-0.5007</td>
</tr>
</tbody>
</table>

Table 4.1 Eigenvalues of linearized models for different instants in the inertia phase (Figure 4.1)

<table>
<thead>
<tr>
<th>Linearization point</th>
<th>Linearized at 50% into inertia phase</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction Type</td>
<td>Type A</td>
</tr>
<tr>
<td>( P_{a2} ) (kPa)</td>
<td>20</td>
</tr>
<tr>
<td>E1</td>
<td>-0.0043</td>
</tr>
<tr>
<td>E2</td>
<td>0</td>
</tr>
<tr>
<td>E3</td>
<td>-44.328</td>
</tr>
<tr>
<td>E4</td>
<td>-2.1189</td>
</tr>
<tr>
<td>E5</td>
<td>-4.86e4</td>
</tr>
<tr>
<td>E6</td>
<td>-397.92</td>
</tr>
<tr>
<td>E7</td>
<td>-0.4914</td>
</tr>
</tbody>
</table>

Table 4.2 Eigenvalues of linearized models for different \( P_{a2} \) values (50\% into inertia phase, Figure 4.2)

198
The results in Table 4.1 and Table 4.2 also indicate that the eigenvalues change by small amounts, which is in agreement with the small change in the frequency response characteristics shown in Figure 4.1 and Figure 4.2. This would suggest that adjustment of the open loop control algorithm for accumulator backpressure adjustment, by transmission calibration, should be straightforward. Alternatively, closed loop control of the turbine speed by manipulation of the accumulator backpressure should be achievable by a controller with fixed gains.

Figure 3.30 — Figure 3.32 have already indicated that change in the clutch plate friction characteristics changes the shift transients significantly. The effect of this change on the frequency response \( \omega_r/P_{a2} \) is displayed in Figure 4.3, where the ‘90%’ case of Figure 4.1 is compared with the ‘90%’ case when the Type B friction characteristic is assumed for the clutch plate. The accumulator backpressure \( P_{a2} \) and the instant in the inertia phase where the linearization is performed are the same for the two cases in Figure 4.3. The change in the frequency response occurs mainly as a change in gain of about 2 decibels or 25%. The eigenvalues of the linearized models for different friction characteristics are listed in Table 4.3.

It should be noted that the change in frequency response noted in Figure 4.3 does not apply to a single shift and therefore need not be accommodated by a shift controller with fixed parameters. The result simply points to the need for tuning the shift controllers to the specific dynamic response of the transmissions they control.

4.3 Controller design for the inertia phase of the 1-2 power-on upshift

4.3.1 Plant model during the inertia phase of the 1-2 power-on upshift

The linear model obtained is examined here in terms of the frequency response \( \omega_r/P_{a2} \). The frequency response of the obtained 7th order linear model is given in Figure 4.4. The operating conditions are chosen at a time instant halfway through the inertia phase for the case where the accumulator backpressure \( P_{a2} \) is 100 kPa. The poles and zeros of the linearized model are given in Table 4.4. Note that in Figure 4.4, the phase plot of the frequency response starts from \(-180^\circ\). This is due to the sign of steady-state
gain $K_p$, which is negative. Physically, this negative sign indicates that when the accumulator backpressure $P_{a2}$ (system input) increases, the turbine speed $\omega_4$ (system output) decreases.

Table 4.4 shows that poles $P1$ and $P2$ cancel zeros $Z3$ and $Z4$. Further examination of the pole/zero locations indicates that pole $P5$ is out of the frequency range of interest ($1 - 100$ rad/sec), thus reducing the system model to 4$^{th}$ order. The frequency response of the 4$^{th}$ order model in the frequency range of interest is shown in Figure 4.5, and compares well with the full order model in the frequency range chosen. Therefore, the following shift controller design is based on the 4$^{th}$ order model.

Figure 4.3 Frequency response $\omega_4/P_{a2} - P_{a2} = 100$ kPa, 90% into inertia phase, Type A and B friction characteristics
<table>
<thead>
<tr>
<th>Linearization point</th>
<th>Linearized at 90% into inertia phase</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction Type</td>
<td>Type A</td>
</tr>
<tr>
<td>P_{a2} (kPa)</td>
<td>100</td>
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<tr>
<td>E1</td>
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<tr>
<td>E2</td>
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</tr>
<tr>
<td>E3</td>
<td>-31.9167</td>
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<tr>
<td>E4</td>
<td>-2.2706</td>
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<td>E5</td>
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<td>-0.5007</td>
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<tr>
<td>E8</td>
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<td>E9</td>
<td>0</td>
</tr>
<tr>
<td>E10</td>
<td>0</td>
</tr>
<tr>
<td>E11</td>
<td>0</td>
</tr>
<tr>
<td>E12</td>
<td>0</td>
</tr>
<tr>
<td>E13</td>
<td>0</td>
</tr>
<tr>
<td>E14</td>
<td>0</td>
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<td>E15</td>
<td>0</td>
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<td>E16</td>
<td>0</td>
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<td>E17</td>
<td>0</td>
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<tr>
<td>E18</td>
<td>0</td>
</tr>
<tr>
<td>E19</td>
<td>0</td>
</tr>
<tr>
<td>E20</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 4.3 Eigenvalues of linearized models for different friction characteristics

(Figure 4.3)

Figure 4.4 Frequency response of the 7th order model
Table 4.4 Poles and zeros of the linearized 7th order system model

<table>
<thead>
<tr>
<th>Poles: P_i (rad/sec)</th>
<th>Zeros: Z_i (rad/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1 -0.0043</td>
<td>Z1 -12.7768</td>
</tr>
<tr>
<td>P2 0</td>
<td>Z2 -0.0405</td>
</tr>
<tr>
<td>P3 -37.4136</td>
<td>Z3 0.0000</td>
</tr>
<tr>
<td>P4 -2.1753</td>
<td>Z4 -0.0043</td>
</tr>
<tr>
<td>P5 -5.7060×10^4</td>
<td></td>
</tr>
<tr>
<td>P6 -402.8598</td>
<td></td>
</tr>
<tr>
<td>P7 -0.5488</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4.5 Frequency response of the 4th order model
4.3.2 Solenoid model

The accumulator backpressure $P_{a2}$ is the input to the plant. Electronic control of hydraulic pressure requires the use of electro-hydraulic actuators. A highly simplified 2nd order linear solenoid dynamic model, given in equation (4.1), is employed to manipulate the accumulator backpressure.

$$G_s(s) = \frac{P_{a2}}{V_{s2}} = \frac{K_{s2}}{s^2 + \frac{2\zeta_{s2}\omega_{s2}}{\omega_{s2}} + 1}$$

where

- $V_{s2}$ is the input voltage to the solenoid
- $K_{s2}$ is the solenoid gain, 68950 Pa/volt
- $\omega_{s2}$ is the solenoid natural frequency, 10 Hz
- $\zeta_{s2}$ is the solenoid damping ratio, 0.707.

The frequency response of the 4th order system and the solenoid is shown in Figure 4.6. Comparing Figure 4.6 with Figure 4.5, it is seen that the chosen solenoid dynamics contribute significantly to system dynamic behavior in the frequency range of interest. A detailed experimental and theoretical study of a solenoid controlled pressure system is presented in Chapter 5, and results in similar findings.

4.3.3 1-2 upshift controller design

The objective of the 1-2 upshift controller design studied in this work is to control the accumulator back pressure $P_{a2}$ in order to let the turbine speed $\omega_t$ closely track a specified trajectory during the inertia phase. The control system is represented by the block diagram shown in Figure 4.7, where $G_c$ represents the controller, $G_s$ represents the solenoid dynamics and $G_p$ represents the plant dynamics.
Figure 4.6 Frequency response of the 4th order model and solenoid

Figure 4.7 Control system block diagram

Previous studies (Narita, 1991) show that the inertia phase duration is closely related to shift quality. Hayabuchi et al. (1996) also mention that the passenger perception of a shift is good when the rate of turbine speed change is between 2600 ~ 204
4000 rpm/sec as shown in Figure 6.5. Therefore, a feedback control system can be designed to achieve a desired duration of the inertia phase. The optimum value of this duration is generally obtained by calibration, that is, repeated tests on the transmission. In this study, it is assumed that the optimum inertia phase trajectory of the turbine speed has been determined by such calibration tests, and that the desired trajectory during the 1-2 upshift is in the form of a ramp signal. By varying the slope of this ramp signal, the duration of the inertia phase is varied. The waveform of the shaft torque is certainly of interest in defining the shift quality. However, it is not usually available for feedback.

Controller design is performed in the frequency domain. The controller has the form shown in equation (4.2), the free integrator in the controller being needed for tracking a ramp type trajectory. The lag compensator is used to increase the type number of the system.

\[
G_c(s) = K_c \cdot \alpha \cdot \frac{T_1 + 1}{s \cdot (\alpha T_1 + 1)} \quad (0 < \alpha < 1)
\]  \hspace{1cm} (4.2)

The controller has poles at 0 and -1/\(\alpha T\) and a zero at -1/T. The controller parameters are chosen by trial and error as follows:

\[
K_c = 2.4, \alpha = 0.1, T = 0.5
\]  \hspace{1cm} (4.3)

The frequency response of the controller is shown in Figure 4.8. It is clear that the controller would reduce the magnitude crossover frequency and the phase margin of the compensated system. The frequency response of the 4th order model with the solenoid and the designed controller is given in Figure 4.9. The gain and phase margins for the compensated system are approximately 5db and 30° respectively. The magnitude crossover frequency is approximately 23 rad/sec.
Figure 4.8 Frequency response of the controller

Figure 4.9 Frequency response of the compensated system
4.3.4 Simulation results

Shift acceptability is a function of many aspects. One is shift smoothness, which is determined by the smoothness of the output shaft torque or the jerk of the vehicle (the derivative of the angular speed) during the shift. The other is clutch life. A longer shift duration means more clutch wear, i.e. shorter clutch life, while a shorter shift duration generally involves larger torque changes during the shift, which may be unpleasant to the passengers in the vehicle. Good shift quality involves a balance between these two aspects. In this study, the emphasis is on model based controller design that assumes that the optimum shift duration has already been given. Therefore, turbine speed trajectories with different slopes are used as different instances of desired trajectories. Steeper trajectories represent shorter shift duration, while shallower trajectories represent longer shift duration. In practice, the optimum slope of the desired turbine speed trajectory would be determined by calibration.

The controller performance is verified on the nonlinear simulation of the powertrain. The shift controller is turned on when the inertia phase starts, and is turned off when the slip speed of the oncoming clutch is lower than a specified threshold. The accumulator backpressure $P_{a2}$ is kept constant after the closed loop control is terminated. Thus, the second clutch lockup occurs under open loop control. The results shown in this section demonstrate the performance and robustness of the designed controller for a variety of desired trajectories. The issue of robustness arises because the controller design is based on a linearized model of the powertrain response for a specific condition, whereas the evaluation is performed under a variety of operating conditions. Obviously, controller robustness is of great significance for practical application.

Figure 4.10 - Figure 4.15 show the simulation results for desired turbine speed trajectories with different slopes. The full nonlinear simulation is being used here for controller evaluation, not the one using a simplified engine model which was used for obtaining the linearized models. Figure 4.10 shows a steady state tracking error of 2 rad/sec when the slope is $-100$ rad/sec$^2$, the duration of the resulting inertia phase being about 0.85 seconds. Figure 4.11 shows the 2$^{nd}$ clutch pressure trace $P_{c2}$ and the accumulator
backpressure $P_{a2}$. Figure 4.12 shows the output shaft torque $T_i$, and indicates a peak value at clutch lockup of about 685 Nm.

Figure 4.13 – Figure 4.14 show the results for a steeper turbine speed trajectory with a slope of $-200 \text{ rad/sec}^2$. The steady state tracking error is about 10 rad/sec, and the inertia phase lasts about 0.5 second. The output shaft torque, in this case, is approximately 740 Nm, higher than that in Figure 4.12. In the figures, the “flag” trace indicates the phases of the 1-2 upshift, the four levels, starting from the lowest, corresponding to “first gear”, “torque phase”, “inertia phase”, and “second gear”.

Figure 4.10 Simulation results for $-100 \text{ rad/sec}^2$ slope of $\omega_t$: turbine speed
Figure 4.11 Simulation results for $-100 \text{ rad/sec}^2$ slope of $\alpha$: clutch pressure and the accumulator back pressure.

Figure 4.12 Simulation results for $-100 \text{ rad/sec}^2$ slope of $\alpha$: shaft torque.

209
Figure 4.13 Simulation results for $-200 \text{ rad/sec}^2$ slope of $\omega_t$: turbine speed

Figure 4.14 Simulation results for $-200 \text{ rad/sec}^2$ slope of $\omega_t$: clutch pressure and the accumulator back pressure

210
In conventional transmissions, the accumulator plays an important role by softening the clutch pressure transients and hence the shift transients at lockup. However, closed loop control reduces the dependence on the accumulator’s mechanical design for controlling transients, as the control algorithm influences the transients also. The effects of changing the accumulator’s mechanical parameters such as spring preload and accumulator area are examined here. It is expected that the responsiveness of the system would improve if the accumulator spring preload $F_{ca0}$ is reduced to a quarter of the original value, and the accumulator piston area $A_{a2}$ is reduced to half of the original value.

Figure 4.16 shows the comparison of the frequency response plots for the system with the original and modified accumulator parameters obtained at the instant of time half way into the inertia phase. As shown in the figure, the magnitude plot with the modified accumulator parameters has lower magnitude than that corresponding to the original accumulator parameters. In addition, one pole of the system transfer function is moved closer to the origin more nearly approximating the addition of an integrator to the system frequency response at the operating point. Figure 4.17 shows the comparison of the
compensated system Bode plots with the original and modified accumulator parameters, and the same controller given in equations (4.2) and (4.3). The compensated system with the modified accumulator parameters has a gain margin of 12dB, and a phase margin of 92 degrees, both of which are greater than for the system with the original accumulator parameters. The modified system has better tracking performance than the original system, which is clearly shown in the following simulation results.

Figure 4.18 and Figure 4.19 show the simulation results for clutch pressure and output shaft torque with the modified accumulator, assuming a constant accumulator back pressure at 100 kPa. In this configuration, the torque phase ends during the accumulator stroke. The inertia phase occurs totally in the accumulator stroke, which is different from the original design. This is acceptable for the modified design, since the slope of the clutch pressure trace in the accumulator stroke is steeper, which makes the total duration of the torque phase still acceptable. It would not be desirable if the torque phase duration during the accumulator stroke was much longer. The duration of the inertia phase depends on the level of the torque difference $T_{c2} - T_r$ causing the deceleration of the turbine inertia, and the speed difference $\omega_r - \omega_r^*$ to be made up. If the clutch pressure is too low when the torque phase ends, there is not enough clutch torque capacity to decelerate the turbine speed, which will lengthen the duration of the inertia phase and cause more heat dissipation in the clutch $C_2$, and that is not acceptable.

Figure 4.20 to Figure 4.25 show results using the same controller, but with the modified accumulator. With the original system design, the controller tracks a desired turbine speed trajectory with a slope of $-200$ rad/sec$^2$ with about 10 rad/sec steady state tracking error (Figure 4.13). The same controller with the modified accumulator tracks the same trajectory with a reduced steady state tracking error of less than 1 rad/sec. The duration of the inertia phase is about 0.4 second. However, the torque phase is lengthened to about 0.36 second. The shaft torque has a peak of about 760 Nm at clutch lockup. When tracking a desired turbine speed trajectory with a slope of $-300$ rad/sec$^2$, the steady state tracking error is about 10 rad/sec, and the duration of the inertia phase reduces to 0.33 sec. The peak output shaft torque at lockup is over 800 Nm.
Figure 4.16 Comparison of the frequency response of the system with original and modified accumulator parameters (50% into the inertia phase)

Figure 4.17 Comparison of the frequency response of the compensated systems with original and modified accumulator parameters
Figure 4.18 Pressure versus time with modified accumulator, $P_{a2} = 100kPa$

Figure 4.19 Shaft torque versus time with modified accumulator, $P_{a2} = 100kPa$
Figure 4.20 Simulation results for $-200 \text{ rad/sec}^2$ slope $-\frac{1}{4} F_{a2o}, \frac{1}{2} A_{a2}$: turbine speed

Figure 4.21 Simulation results for $-200 \text{ rad/sec}^2$ slope $-\frac{1}{4} F_{a2o}, \frac{1}{2} A_{a2}$: clutch pressure and the accumulator backpressure
Figure 4.22 Simulation results for – 200 rad/sec² slope – $\frac{1}{4} F_{a2o}, \frac{1}{2} A_{a2}$: shaft torque

Figure 4.23 Simulation results for – 300 rad/sec² slope – $\frac{1}{4} F_{a2o}, \frac{1}{2} A_{a2}$: turbine speed
Figure 4.24 Simulation results for $-300 \, \text{rad/sec}^2$ slope $-\frac{1}{4} F_{a2o} \frac{1}{2} A_{d2}$: clutch pressure and accumulator backpressure

Figure 4.25 Simulation results for $-300 \, \text{rad/sec}^2$ slope $-\frac{1}{4} F_{a2o} \frac{1}{2} A_{d2}$: shaft torque
Comparing the results shown in Figure 4.10 to Figure 4.25, it can be seen that the system is more responsive with the modified accumulator parameters, and can track a steep turbine speed trajectory with less error. It can also be seen that, as the turbine speed slope increases, the output shaft torque has a higher peak. This confirms the findings reported by Narita (1991).

4.4 Open loop control strategy for the 1-2 upshift

Shift control is not restricted to inertia phase control. Torque phase control is also important if the output shaft torque variation during this phase is to be limited. In the torque phase, there is a large shaft torque drop due to the gear ratio change, which is often considered unavoidable. In fact, there is little published literature on the control of the torque phase of a conventional powertrain. The only exception is Brown and Hrovat (1988). In this section, an open loop torque phase control strategy is proposed.

Recall the model equations for the 1-2 upshift derived in Chapter 3 for the powertrain of interest:

**Engine crankshaft dynamics:**

\[ I_e \cdot \dot{\omega}_e = T_i(P_m, AFR, SA, \omega_e) - T_f(\omega_e) - T_a - T_p(\omega_e, \omega_t) \]  
(4.4)

**Torque converter:**

\[ T_p = \left( \omega_e \cdot K^{-1} \left( \frac{\omega_t}{\omega_e} \right) \right)^2 \]  
(4.5)

\[ T_i = r_t \left( \frac{\omega_t}{\omega_e} \right) T_p \]  
(4.6)

In the first gear:

\[ \left[ (I_t + I_{sl}) + I_{Rt}d_{11}^2 + I_{Ct}d_{21}^2 \right] \dot{\omega}_t = T_i - d_{21}R_D \cdot T_s \]  
(4.7)

In the torque phase of 1-2 upshift:

\[ \left[ (I_t + I_{sl}) + I_{Rt}d_{11}^2 + I_{Ct}d_{21}^2 \right] \dot{\omega}_t = T_i + (d_{11} - 1)T_{c2} - d_{21}R_D \cdot T_s \]  
(4.8)

\[ RT_{spl} = T_i - T_{c2} - I_t \cdot \dot{\omega}_t \]  
(4.9)
In the inertia phase of 1-2 upshift:

\[
I_t \cdot \dot{\omega}_t = T_t - T_{c2} \tag{4.10}
\]

\[
(I_{Rr} + I_{si} \cdot e_{11}^2 + I_{cr} \cdot e_{21}^2) \omega_{kr} = T_{c2} - e_{21} \cdot R_D \cdot T_s \tag{4.11}
\]

\[
RT_{pl} = 0 \tag{4.12}
\]

\[
RT_{c2} = T_{c2} \tag{4.13}
\]

In the 2\textsuperscript{nd} gear:

\[
(I_t + I_r + I_{si} \cdot e_{11}^2 + I_{cr} \cdot e_{21}^2) \dot{\omega}_t = T_t - e_{21} \cdot R_D \cdot T_s \tag{4.14}
\]

\[
RT_{c2} = T_t - I_t \cdot \omega_t \tag{4.15}
\]

From the above equations, the output shaft torque during the torque phase and the inertia phase can be calculated as follows.

Denote:

\[
I_1 = I_{si} + I_{Rr} \cdot d_{11}^2 + I_{cr} \cdot d_{21}^2 \tag{4.16}
\]

\[
I_2 = I_{Rr} + I_{si} \cdot e_{11}^2 + I_{cr} \cdot e_{21}^2 \tag{4.17}
\]

Then,

\[
T_{s\_torque} = \frac{1}{d_{21}R_D} \left[ T_t + (d_{11} - 1)T_{c2} - (I_t + I_1) \cdot \dot{\omega}_t \right] = 8.2958 \left[ T_t - 0.4633T_{c2} - (I_t + I_1) \cdot \dot{\omega}_t \right] \tag{4.18}
\]

\[
T_{s\_inertia} = \frac{1}{e_{21} R_D} \left[ T_{c2} - I_2 \cdot \omega_{kr} \right] = 4.4523 \left[ T_{c2} - I_2 \cdot \omega_{kr} \right] \tag{4.19}
\]

Equation (4.18) shows that the output shaft torque during the torque phase is composed of three parts: the turbine torque, the 2\textsuperscript{nd} clutch torque, and the turbine inertia torque.

In the torque phase, the change in turbine speed is small (proportional to the vehicle speed change). In addition, if the engine torque is not controlled, the change in
turbine torque will be small. Therefore, an increase in $T_{c2}$ will cause a sharp output shaft torque drop directly related to the C2 clutch torque increase. This reason for the drop in the shaft torque during the torque phase was noted first by Winchell (1954), and is unavoidable if only transmission functions are controlled. In order to reduce or eliminate the torque drop during the torque phase, engine operation, more specifically, engine torque, has to be controlled in conjunction with transmission control.

Figure 4.26 shows a simple representation of shaft torque variation during an upshift. It is desired to have a smooth transition of output shaft torque during a gear shift, as shown by the dashed line. To overcome the torque drop in the torque phase, the turbine torque $T_t$ has to be increased appropriately in the torque phase to compensate for the increase in $T_{c2}$. This increase can be achieved by increasing engine torque.

Engine control inputs are spark advance, throttle angle, and fuel command. Consider first the use of spark advance to increase engine torque. The limitation in this approach is the limited range of torque adjustment available using spark advance adjustment. This is demonstrated by the results in Figure 4.27 and Figure 4.28.

Figure 4.26 Output shaft torque during an upshift
Figure 4.27 shows the output shaft torque at 40 degree throttle opening, 12 degree spark advance and 12 → 30 degree spark control during the torque phase. With the 12 → 30-degree spark (maximum) advance, the shaft torque drop is reduced by 62 Nm as compared to the 12-degree spark advance case. Figure 4.28 shows the results for the 90-degree throttle opening. The torque drop is reduced by 104 Nm when the spark is advanced to maximum during the torque phase. The benefit of spark advance control in the torque phase is obvious. However, there is a limit to how much torque reduction can be achieved by this method. It depends on the engine operating conditions. If the engine is running at maximum spark advance, then there is no room for improvement. The simulation results obtained are for the stiff output shaft case with Type A clutch friction characteristics.

Figure 4.27 Comparison of output shaft torque during the torque phase for 40 degree throttle opening, 12 degree spark and 12 → 30 degree spark
Figure 4.28 Comparison of output shaft torque during the torque phase for 90 degree throttle opening, 12 degree spark and 12 → 30 degree spark

With an ETC (Electronic Throttle Control) equipped vehicle, the throttle can be controlled without driver intervention. In this case, the spark control and throttle control can be combined to increase engine torque and, consequently, turbine torque in the torque phase. However, there are several challenges. The torque phase is very short, typically less than 500 ms. The turbine torque increase has to occur in this time frame. If it occurs too late, then the output shaft torque level during the inertia phase is increased. Therefore, the timing of the turbine torque increase is critical. This requires integrated engine and transmission control.

To demonstrate this concept, a first order ETC model is assumed, the time constant of which is 100 ms. The approach is to ramp up the throttle to wide open when the system detects the starting point of the torque phase. Figure 4.29 shows the simulation results for the output shaft torque under three conditions. The first condition corresponds to a throttle input of 40 degrees, and base spark control of 12 degrees. The second
condition involves ramping up the throttle to wide open during the torque phase and keeping the spark advance at 12 degrees. The third condition involves combining throttle control and spark control together during the torque phase. As shown in the figure, with combined throttle and spark control during the torque phase, the torque drop is almost eliminated. However, the torque peak at lock up is higher under the third condition, calling for appropriate modification of engine operation during the inertia phase.

These simulation results demonstrate how the torque drop can be reduced during the torque phase by using both engine and transmission control. However, it is preferable for the desired throttle opening to be computed from an inverse static model and a desired output shaft torque trajectory rather than the throttle opening being fixed without an explicit link to the expected output shaft torque value. Therefore, a static model-based open-loop control strategy is considered as follows.

![Figure 4.29 Effect of combined throttle and spark control](image-url)
Recall that the desired output shaft torque during the torque phase and the inertia phase is given in equations (4.18) and (4.19). In the torque phase, the objective is to finish the load transfer as soon as possible. Therefore, we choose the 2\textsuperscript{nd} clutch torque $T_{C2}$ as input, and consider the static relationship only. Then,

$$T_{r\_desired} = T_{s\_desired} \cdot d_{21} \cdot R_D - (d_{11} - 1)T_{C2}$$

(4.20)

In the inertia phase, the output shaft torque depends on $T_{C2}$. Therefore,

$$T_{C2\_desired} = T_{s\_desired} \cdot e_{21} \cdot R_D$$

(4.21)

The desired output shaft trajectory is given by:

$$T_{s\_desired} = T_{s0} + K_{Tr} \cdot (t - t_0)$$

(4.22)

where $T_{s0}$ is the output shaft torque value at the beginning of the torque phase, $K_{Tr}$ is the slope of the output shaft trajectory, and $t_0$ is the time instant when the torque phase starts.

From equation (4.20), we can calculate the desired throttle opening required to achieve the desired output shaft trajectory as follows:

$$T_{p\_desired} = \left[ \frac{\omega_{e\_desired}}{K \left( \frac{\omega_{t0}}{\omega_{e\_desired}} \right)} \right]^2$$

(4.23)

$$T_{r\_desired} = r_T \left( \frac{\omega_{t0}}{\omega_{e\_desired}} \right) \cdot T_{p\_desired}$$

(4.24)

From equations (4.23) and (4.24), we can solve for $T_{p\_desired}$ and $\omega_{e\_desired}$ given the turbine speed at the beginning of the torque phase $\omega_{t0}$ and the desired turbine torque trajectory $T_{r\_desired}$.

$$T_{i\_desired}(P_m, AFR, SA, \omega_{e\_desired}) = T_f(\omega_{e\_desired}) + T_{p\_desired}$$

(4.25)
\[
\eta_{vol} \left( \omega_{e, \text{desired}}, P_{m, \text{desired}} \right) \cdot V_e \cdot \omega_{e, \text{desired}} \cdot p_m = \text{MAX} \cdot (1 - \cos(1.14459 \alpha_{\text{desired}} - 1.06)) \cdot \left( 1 - e^{\frac{\mu}{\mu_m - 1}} \right)
\]

From equations (4.25) and (4.26), we can solve for the desired throttle opening \( \alpha_{\text{desired}} \).

In the inertia phase, the desired 2\(^{nd}\) clutch pressure can be calculated as follows:

\[
P_{c2, \text{desired}} = \frac{T_{c2, \text{desired}}}{\mu(\omega_t - \omega_{Rr}) \cdot k_{c2}}
\]

where

\[
k_{c2} = A_{c2} \cdot R_{c2}
\]

The engine control variables are returned to the original values before the torque phase control starts, and are not controlled during the inertia phase.

Figure 4.30 and Figure 4.31 show the simulation results with the model-based open-loop control strategy. Figure 4.30 shows the comparison of two open-loop control strategies. The curve labeled “WOT torque phase control & inertia phase control” corresponds to wide-open throttle control in the torque phase, and pressure control in the inertia phase. The curve labeled “Model-based torque and inertia phase control” corresponds to the static-inverse throttle control in the torque phase, and clutch pressure control in the inertia phase. As shown in the figure, the output shaft torque drop in the torque phase is about 40 Nm, and the output shaft torque does not overshoot as much as in Figure 4.29. The static-inverse throttle control gives about the same performance as compared to the wide open throttle (WOT) throttle control. Obviously, the static-inverse throttle control is preferable. In the inertia phase, the output shaft torque tracks the desired trajectory perfectly. The torque overshoot in Figure 4.30 is eliminated.

Figure 4.31 shows the throttle angle response and the 2\(^{nd}\) clutch pressure trajectories. The pressure trajectories are similar to those shown in Chapter 3, i.e. there is a sharp rise of pressure in the torque phase, followed by a shallower pressure trace during the inertia phase. These results show that if the clutch pressure is not actively controlled, relatively good shift performance in the inertia phase is achievable by selecting the clutch.
design parameters to achieve pressure trajectories like that shown in Figure 4.31. However, due to manufacturing variations and operating condition changes, there is no guarantee of satisfactory shifts with such an approach. Therefore, in order to achieve repeatable and satisfactory shift performance under wide operating conditions, closed-loop control of powertrain functions such as ETC and clutch pressure is necessary.

The method demonstrated here is open loop control based on the physical system model presented in Chapter 3. Therefore, it is highly dependent on details of system behavior and would require calibration in practice. But it does provide insight into system behavior and the requirements of calibration. More sophisticated control strategies may be possible if the powertrain system has appropriate hardware, such as ETC, and allows for direct control of clutch pressure. We note that any practical control strategy would also be highly dependent on system configuration. If the output shaft torque were the control target, control implementation in practice would include an open loop component, because installing a torque sensor in the driveline in production vehicles is not economically feasible with current technology. Thus, the most feasible approach to deal with the torque phase control for current generation systems involves some open loop control, such as that demonstrated in this section.

4.5 Control issues in shift control

Shift control issues we have investigated in this chapter are open loop output shaft torque control to reduce the shaft torque drop in the torque phase and shaft torque overshoot in the inertia phase, and closed loop inertia phase control to let the turbine speed track a specified trajectory, for a one-way clutch to clutch gear shift. Other shift control applications include pressure manipulation, clutch-to-clutch shift control and shift schedule control. The approach employed in the inertia phase control here can be extended easily to clutch-to-clutch and band-to-clutch shifts. The shaft torque control approach needs to be adapted in conjunction with clutch pressure control, if applied to clutch-to-clutch and band-to-clutch shifts. Incorporating engine control functions into clutch-to-clutch and band-to-clutch shifts should be able to achieve better control performance due to the additional degree of control freedom.
Figure 4.30 Open loop control of torque and inertia phase shaft torque

Figure 4.31 Comparison of throttle angle response and 2\textsuperscript{nd} clutch pressure using two open loop control schemes
In summary, the objective of shift control is to achieve a smooth shift and short shift duration in order to reduce clutch wear. Regardless of the specific control problem, some generic features will always be encountered in transmission shift control. The first feature is that the control objective is normally indirectly related to the controlled variable. For example, the controlled variable for the 1-2 upshift inertia phase problem studied above is the turbine speed trajectory, however the control objective is a pleasing shift. However, the common feature in shift control problems is that the manipulated inputs are engine inputs such as throttle angle, spark advance or transmission inputs, such as hydraulic pressure or solenoid input voltage. This feature calls for powertrain system modeling to relate the control objectives to the controlled variables. Such system models are usually highly nonlinear, which is the second feature of transmission control problems. Model analysis and simplification techniques are necessary to identify dominant dynamics in the specific control problem for the purpose of controller design. One example of such analysis and simplification is given in sections 4.2 and 4.3.

The third feature of transmission control problems is the manipulation of frictional forces. Gear shifting in a stepped automatic transmission is done by engaging and disengaging clutches, and transferring load from one set of friction elements to another. How well the shift is performed depends strongly on the manner of manipulation of frictional forces. As noted earlier, the friction torque for a plate or band clutch is a function of the fluid pressure, the clutch/band geometry, and the friction coefficient of the contacting surfaces. One feasible control input is clutch pressure, which can be indirectly controlled by the manipulation of the accumulator back pressure, such as the case investigated in this chapter, or directly controlled by a solenoid valve, which will be explored in detail in Chapter 5. As discussed in this chapter, using the accumulator back pressure to control clutch pressure makes the system less responsive, and is not desirable for operations calling for fast and responsive clutch pressure control. In the case of using solenoid to control clutch pressure to be presented in Chapter 5, the dynamics of the solenoid become crucial for accurate manipulation of clutch pressure. Moreover, the friction characteristics of the contacting surfaces are also critical to the performance of clutches. The nonlinearity in the $\mu - \omega_{tip}$ curve has a significant effect on clutch
performance near lockup and at low slip speeds, as demonstrated in Chapter 3 and this
chapter. Furthermore, the friction characteristics may change with temperature and wear.
This calls for robustness of shift controllers in practice.

The fourth feature of transmission control problems is that the stiffness in the
system is subject to large variations. For example, the clutch spring stiffness varies with
clutch piston motion and deflection of wave springs, and furthermore, is different from
that of the accumulator spring. In addition, when the springs reach their limits, the
effective stiffness changes from that of the springs to that of the fluid in the system. The
fluid properties, such as bulk modulus and viscosity etc., also vary over a wide range. As
a result, the dynamics of the plant vary significantly.

The fifth feature is that shift control involves change in the power flow paths
within the transmission, which causes system structure changes, in addition to changes in
the parameter values. This can be seen, for example, from the shift dynamic equations
derived in Chapter 3. During the 1st to 2nd gear shift, the input to the transmission changes
from the input sun gear assembly to the reaction ring gear assembly. Therefore, the
dependent and independent variables of the model change when the transmission shifts
from the 1st to the 2nd gear. This calls for smooth transition of control algorithms from
torque phase control to inertia phase control, and good coordination with other vehicle
control functions.

The last noteworthy feature is that transmission control objectives are sometimes
best stated in terms of variables not available for measurement. For example, shift feel is
best stated in terms of vehicle acceleration and jerk or transmission output shaft torque.
Vehicle acceleration may not be measurable economically, and jerk and shaft torque are
not easy to measure. In such cases, control action has to rely on other information
indirectly related to shift feel, which in turn requires much experimental work to establish
the relationship between the desired design objectives and measured response.
4.6 Conclusion

In this chapter, controller design for a 1-2 power-on upshift, based on a powertrain model, is presented and evaluated.

The advantage of model-based controller design is demonstrated in this work. A model-based approach identifies the dominant factors in the transmission shift response and leads to a straightforward controller design. In this case, a fixed gain controller works fine during the inertia phase and the controller is quite robust. It is also demonstrated that improved dynamic models of transmission behavior can lead to improved control of shifts and, furthermore, will lead to better understanding of how transmission dynamic behavior depends on its mechanical design. Furthermore, dynamic models of transmission behavior provide guidance to open loop control design, which will aid the calibration process.

Shifts involving clutch-to-clutch load transfer or rapid pressurizing of the input clutch as in neutral-idle transitions require higher responsiveness of the system. In these applications, direct manipulation of the clutch pressure is desired in order to achieve a fast and well-behaved response. Indirect control of clutch pressure by manipulation of the accumulator backpressure, as done here, is generally unacceptable in such cases.

Simultaneous and independent control of the output shaft torque during both the torque and inertia phases, and of the turbine speed during the inertia phase, requires coordination of engine and transmission control functions to achieve satisfactory control. Model based control for such cases is also a challenging controller design problem. Preliminary results indicating the nature of the benefits achievable using such an approach, are also given.
CHAPTER 5

CLUTCH PRESSURE CONTROL SYSTEM

5.1 Introduction

For neutral-idle control, electronic throttle control (ETC) and forward clutch application have to be closely coordinated to achieve acceptable shift quality. The clutch pressure control system plays an important role in this coordination. This chapter focuses on the study of a pressure control system for application in neutral-idle control. The objectives here are to identify the critical dynamics in the clutch pressure control system through analytical modeling and experimental validation, and to design satisfactory closed-loop control. The closed loop clutch pressure control system developed here is then employed in neutral-idle control, which is presented in Chapter 6.

A neutral-idle pressure control test system is set up in the Fluid Power Laboratory of the Department of Mechanical Engineering at The Ohio State University, for the purpose of experimentation. A forward clutch assembly similar to that used in production is fully instrumented, and the clutch pressure controlled by a two-stage valve. An analytical model for clutch pressure dynamic response is developed and validated by experimental data. Based on the analytical model, a closed loop clutch pressure controller is designed and implemented in the test setup. In the following sections, the pressure control system configuration, model development and validation, model simplification, controller design and implementation, and robustness analysis are presented. Furthermore, the relationship of the pressure control system studied here to pressure control in other hydraulic control applications is also explored.
5.2 Test system configuration

The neutral-idle pressure control test system consists of a hydraulic power unit, a two-stage pressure control valve, an instrumented forward clutch assembly, pressure sensor signal conditioning electronics, valve drive electronics, and a dSPACE Model DS1102 data acquisition system. General signal analysis instrumentation such as a signal generator and an oscilloscope is also used as needed. A photo of the test setup is shown in Figure 5.1.

Figure 5.2 shows a schematic of the test setup. The "Hydraulic Power Unit" supplies pressure to the forward "Clutch Assembly". The supply pressure is set by a bypass relief valve. The "Manual Control Box" controls a force motor, which is the first stage of the "Two-Stage Pressure Control Valve" and controls the pilot pressure. There
are five pressure sensors in the test setup. They measure hydraulic line pressure, pilot pressure, back chamber pressure in the spool valve which controls flow to the forward clutch, feeding chamber pressure in the valve manifold, inner clutch chamber pressure, and outer clutch chamber pressure. As will be described subsequently, the forward clutch has two chambers connected by an orifice. The sensor signals are processed by the "Signal Conditioning" box, and sent to a PC (Gateway 2000, P5-90, 90MHz) through the "dSPACE I/O" box. The computer sends the valve control signal to the manual control box through the dSPACE interface.

Figure 5.2 Schematic of test setup
5.3 Neutral-idle valve description and dynamic model

There are two stages of pressure control in the neutral-idle pressure control valve. The first stage is the pilot stage, with the pilot pressure being controlled by a solenoid or linear force motor. The second stage is the power stage, where the pilot pressure controls the motion of a spool valve and the resulting flow from the main hydraulic line to the clutch cavity and hence the clutch pressure. Figure 5.3 shows the physical schematic of the power stage of the pressure control valve. The key elements are labeled in the figure. The feedback orifice between the feeding chamber and the back chamber introduces pressure feedback through the force balance on the spool.

Figure 5.4 shows a hydraulic schematic of the system model. The dotted block represents the hydraulic power unit with motor, pump and relief valve. A pressure gage is mounted on the relief valve. Line pressure is set by mechanically loading the spring inside the relief valve. The linear force motor controls the pilot pressure \( P_p \), which acts on the pilot end of the valve spool. The force balance on the spool determines the

![Figure 5.3 Physical schematic of the power-stage of the pressure control valve](image)

234
valve spool position. $x_{sp}$ is the valve spool displacement from the extreme left end.

When $x_{sp}$ is greater than 4.0 mm, the pressure port in the spool valve opens with its area $a_s$ being a function of the valve spool position, and the supply fluid enters the feeding chamber with the chamber pressure being $P_f$. The orifices restricting fluid flow are labeled by their corresponding areas, and normal flow directions are defined in Figure 5.4. $a_{lv}$ is the cumulative flow area associated with leakage paths between the feeding chamber and the exhaust and the valve exhaust opening area labeled as port No. 3 and 4 in Figure 5.3, which is a function of the spool position. The orifice $a_f$ is the feedback orifice area connecting the valve back chamber with the feeding chamber. In steady state, the back pressure $P_b$ is equal to the feeding chamber pressure $P_f$. Some of the fluid in the feeding chamber will also leak through the valve mounting surface with an equivalent orifice area of $a_{lim}$, which is between port No. 8 and the forward clutch support (not shown in the figure) shown in Figure 5.3. Fluid from the feeding chamber also flows through an orifice with flow area $a_t$, and enters the inner clutch chamber, which is at pressure $P_{ci}$. Another leakage path is the shaft end leakage path characterized by leakage area $a_{lsr}$. $P_x$ is the pressure at an appropriate intermediate point between the feeding chamber and the inner clutch chamber. $a_{ci}$ is the area of the supply orifice to the inner clutch chamber. There is a restriction orifice, with area $a_{co}$, between the inner and outer clutch chambers. The outer clutch chamber pressure is $P_{co}$. The “cloud” shaped bubble represents the air trapped inside the outer clutch chamber.

The amount of air trapped inside the outer clutch chamber depends on the location of a check valve which allows fluid in the outer clutch chamber to drain when it is depressurized. When the outer clutch chamber is filling, as is the case during the neutral-idle shift, the check valve closes off the drain port. Some air is likely to be trapped in the outer clutch chamber, the amount of trapped air depending on the location of the check valve relative to the orifice connecting the clutch chambers. In the specific forward clutch assembly under consideration, the check valve assembly is part of the clutch housing, and
the clutch housing can rotate during some of the gearshifts. Thus, the amount of the
trapped air is likely to vary between different instances of the neutral-idle shift.

To summarize, the symbols used in Figure 5.4 are defined as follows:

- $P_l$: main hydraulic line pressure
- $P_f$: feeding chamber pressure
- $P_b$: spool valve back chamber pressure
- $P_p$: pilot pressure
- $P_{ci}$: clutch inner chamber pressure
- $P_{co}$: clutch outer chamber pressure
- $x_{esp}$: valve spool displacement from initial (preloaded) position
- $Q_s$: volumetric flow rate through orifice area $a_s$, see Figure 5.4
- $Q_1$: volumetric flow rate out of feeding chamber, through orifice area $a_1$
- $Q_2$: volumetric flow rate from valve back chamber to feeding chamber
- $Q_{ci}$: volumetric flow rate to clutch inner chamber
- $Q_{co}$: volumetric flow rate to clutch outer chamber
- $Q_{lkv}$: volumetric flow rate through flow area $a_{lkv}$
- $Q_{lkve}$: leakage volumetric flow rate at valve mount
- $Q_{lkse}$: leakage volumetric flow rate at shaft end

The input to the system is the solenoid input voltage. The pilot-stage dynamic
model is discussed in section 5.6. In this section, the focus is on the power-stage of the
neutral-idle pressure control valve. The input to the power stage is therefore the pilot
pressure $P_p$. Below is the summary of the model equations.

**Force balance on the spool:**

$$m_{esp} \cdot \ddot{x}_{esp} + B_{esp} \cdot \dot{x}_{esp} + K_e \cdot x_{esp} = P_p \cdot a_{sp1} - P_b \cdot a_{sp2} - F_{ko}$$  \(5.1\)
Figure 5.4 Schematic of hydraulic system

where $m_{vsp}$ is the spool mass

$B_{vsp}$ is the friction coefficient between the valve spool and sleeve and is calculated as

$$B_{vsp} = \frac{\mu \cdot A}{\Delta r} \quad (5.2)$$

$\mu$ is the fluid viscosity

$A$ is the contact area between the spool and valve sleeve

$\Delta r$ is the radial clearance between the spool and valve sleeve
\( K_e \) is the effective spring constant including the effect of flow forces (Merritt, 1967). The factor 0.8 is used to match experimental data.

\[
K_e = K_s + 0.43\pi \cdot d_{sp2} \cdot (P_i - P_f) \cdot 0.8
\]

(5.3)

\( K_s \) is the mechanical spring constant of the preload spring

\( d_{sp2} \) is the spool diameter at the spring end

\( d_{sp1} \) is the spool diameter at the pilot end

\( a_{sp1} \) is the spool cross sectional area at the pilot end, i.e.

\[
a_{sp1} = \pi \left( \frac{d_{sp1}}{2} \right)^2
\]

(5.4)

\( a_{sp2} \) is the secondary spool cross sectional area, i.e.

\[
a_{sp2} = \pi \left( \frac{d_{sp2}}{2} \right)^2
\]

(5.5)

\( F_{k0} \) is the spring preload.

The differential force on the right side of the spool at area \( a_{sp1} \) and the left face of the spool at area \( a_{sp2} \) is not included in equation (5.1). This is because the exhaust port No. 3 is always open as shown in Figure 5.3.

Flow rate through spool valve opening:

\[
Q_s = C_d \cdot a_s(x_v) \cdot \sqrt{\frac{2(P_i - P_f)}{\rho}}
\]

(5.6)

\[
a_s(x_v) = (x_{vp} - 4.0e^{-3})\pi \cdot d_{sp2} = x_v \cdot \pi \cdot d_{sp2}
\]

(5.7)

The 4mm offset in equation (5.7) is the required valve spool displacement from the initial valve spool position before the supply port opens. It is also the length of the valve land between port 5 and 6 shown in Figure 5.3.

Feeding chamber continuity equation:

\[
(Q_s + Q_2 - Q_{lv} - Q_{lim}) - Q_1 = \frac{V_f}{\beta} \frac{dP_f}{dt}
\]

(5.8)
where $\beta$ is fluid bulk modulus

$V_f$ is the feeding chamber volume

We assume that the fluid compliance effect is negligible for the small volume $V_f$, that is, $\frac{V_f}{\beta} = 0$. Equation (5.8) is therefore, simplified to

$$Q_1 = Q_s + Q_2 - Q_{dv} - Q_{lim}$$

(5.9)

The assumption to neglect the capacitance effect is based on inspection of the valve mechanical drawings, and observation of the high leakages from the feeding chamber to the exhaust ports and the mounting surface.

Spool back chamber continuity equation:

$$-Q_2 = \frac{dV_b}{dt} + \frac{V_b}{\beta} \frac{dP_b}{dt}$$

(5.10)

$$-Q_2 = C_d \cdot a_f \cdot \sqrt{\frac{2}{\rho} \left[ \frac{P_f - P_b}{P_f - P_b} \right] \cdot \text{sgn}(P_f - P_b)}$$

(5.11)

where

$\rho$ is the fluid density

$C_d$ is the orifice flow discharge coefficient, assumed to be 0.61

$a_f$ is area of the feedback orifice between the feeding chamber and the spool valve back chamber

Assuming $\frac{V_b}{\beta} = 0$, that is, the compressibility effect of the fluid in the back chamber can be ignored because of its small volume, equation (5.10) can be simplified to

$$Q_2 = -\frac{dV_b}{dt} = a_{sp2} \cdot x_v$$

(5.12)

The assumption of negligible capacitance is based on a mechanical drawing of the valve. Compared to the volume of the clutch chamber, the neglected back chamber volume is much smaller. At the initial positions shown in Figure 5.3, the back chamber volume is approximately $7 \times 10^{-7} \text{ m}^3$, whereas the outer clutch chamber volume is approximately $5.1 \times 10^{-3} \text{ m}^3$. 

239
Clutch inner and outer chambers:

In the neutral-idle application, the forward clutch is always filled and the clutch piston is nearly fully stroked, in order to allow for a rapid buildup of clutch pressure and transition to drive (Hayabuchi et al., 1996). Therefore, the clutch filling period can be ignored in the model. This situation is specific to the neutral-idle shift because of the need to keep the shift time short. In other instances of clutch-to-clutch shifts, clutch filling needs to be explicitly accommodated as part of the clutch control strategy. Furthermore, in the specific forward clutch being considered, there are two clutch chambers. The hydraulic fluid enters the inner chamber first through an orifice \( a_{ci} \), and then the outer chamber through a smaller control orifice \( a_{co} \). As has already been mentioned, the air trapped in the clutch outer chamber, and hence the effective fluid compressibility in the clutch outer chamber, is likely to vary between one neutral-idle transition and another.

\[
Q_{ci} = C_d \cdot a_{ci} \cdot \sqrt{\frac{2\left(P_x - P_{ci}\right)}{\rho}} \cdot \text{sgn}(P_x - P_{ci}) \quad (5.13)
\]

\[
Q_{co} = C_d \cdot a_{co} \cdot \sqrt{\frac{2\left(P_{ci} - P_{co}\right)}{\rho}} \cdot \text{sgn}(P_{ci} - P_{co}) \quad (5.14)
\]

\[
Q_{ci} - Q_{co} = \frac{V_{ci}}{\beta} \frac{dP_{ci}}{dt} \quad (5.15)
\]

\[
Q_{co} = \frac{V_{co}}{\beta_e} \frac{dP_{co}}{dt} = C_{co} \frac{dP_{co}}{dt} \quad (5.16)
\]

In practice, as compared to \( a_{co} \), the orifice \( a_{ci} \) governing flow to the clutch inner chamber is much larger, i.e. the pressure drop is small, a typical number being 2 psi for this application. This results from the fact that the orifice restriction between the clutch chambers is intentionally inserted to ensure a gentler pressure rise in the clutch outer chamber pressure and hence a smoother neutral-idle transition. Thus, the pressure drop across this orifice restriction is dominant. To simplify the model, therefore, the resistance effect of the orifice \( a_{ci} \) is ignored. In addition, the compliance associated with the inner
clutch chamber is also ignored because it is small as a result of negligible air entrainment. The fluid bulk modulus, which is what applies to the inner clutch chamber compliance, is $1.3 \times 10^6$ kPa, whereas the effective bulk modulus of the outer clutch chamber varies between $4.1 \times 10^4$ kPa and $1 \times 10^5$ kPa at $90^\circ$F. Therefore, even though the inner clutch chamber volume is approximately half of that of the outer clutch chamber volume, its capacitance is much lower than that of the outer clutch chamber. Therefore, equations (5.13) - (5.16) can be simplified as

$$Q_{co} = Q_1 - Q_{ate} = C_{co} \frac{dP_{co}}{dt}$$

$$Q_{co} = C_d \cdot a_{co} \cdot \sqrt{\frac{2\left(P_x - P_{co}\right)}{\rho}} \cdot \text{sgn}(P_x - P_{co})$$

where

- $P_x$ is an intermediate pressure as shown in Figure 3.1, and is calculated from

$$P_x = P_f - \left(\frac{Q_1}{C_d \cdot a_1}\right)^2 \frac{p}{2} \cdot \text{sgn}(P_f - P_x)$$

- $C_{co}$ is a lumped parameter representing the fluid compliance, and can be represented by

$$C_{co} = \frac{V_{co}}{\beta_e}$$

- $S_{co}$ is a lumped parameter representing the stiffness, and is defined by

$$S_{co} = \frac{\beta_e}{V_{co}}$$

- $\beta_e$ is the effective bulk modulus

Thus, the model of the power-stage dynamic response is third order and nonlinear. The three energy storage elements are spool inertia, the spool return spring, and the clutch outer chamber. The nonlinearities arise from the orifice flow equations and valve flow forces. Additional nonlinearities would arise if the port area limits in the spool valve are encountered during valve operation.
The equations as stated above assume that the flow rates are in the directions indicated and the equations would be different if the flow directions were different. These conditions were satisfied during all the experiments.

5.4 Model parameter determination

Some of the model parameters are initially estimated from part drawings and hydraulic fluid property data sheets. However, the model results differ greatly from experimental observations. Further analysis reveals the need to identify the following parameters through experimental results:

- Spool valve spring constant \( K_s \)
- Spool valve spring preload or installed height \( F_{k0}, x_{vp-0} \)
- Leakage flow rates at different locations \( Q_{ikv}, Q_{ikmt}, Q_{ikse} \)
- Orifice area \( a_1 \)

In order to identify the above parameters, several sets of tests are conducted. In these tests, the leakage flow rates at different locations are measured, and the following signals are recorded:

- Pilot pressure \( P_p \)
- Back pressure \( P_b \)
- Feeding chamber pressure \( P_f \)
- Clutch inner chamber pressure \( P_{ci} \)
- Clutch outer chamber pressure \( P_{co} \)
- Line pressure \( P_l \)
- Solenoid voltage \( V_s \)

The dSPACE input-output box has four A/D and D/A channels. Therefore, the above signals are recorded in two groups. The pilot pressure \( P_p \) is collected in both groups for purposes of synchronization and comparison.
Figure 5.5 – Figure 5.7 show steady state test results for different solenoid voltage inputs. The designations “110” and “130” in the data set label stand for line pressure settings of 110 psi (785.5kPa) and 130 psi (896.4kPa) respectively. The last 3 digits, “322” and “327”, indicate the test dates. Figure 5.5 shows pressure and force versus solenoid input voltage, corresponding to a line pressure setting of 110 psi (785.5kPa).

When the solenoid input voltage is below 2.3 volts, the pilot pressure is high, the spool is at its right extreme position, and the feeding chamber pressure reaches the line pressure. This means that if the pilot pressure changes in this range, the feeding chamber pressure would not respond to the change. Therefore, the pressure control range for this value of the line pressure involves solenoid voltages above 2.3 volts. Figure 5.5 also shows the computed forces acting on the pilot end and the back end of the spool, versus solenoid input voltage. The difference between the forces on the pilot end and the back end of the spool is the force due to the return spring and flow force. This is used to estimate the mechanical spring constant and the spring preload.

The other end of the control range in Figure 5.5 is when the input solenoid voltage exceeds 3.5 volts. In this case, the return spring in the back chamber of the spool pushes the spool to the left extreme position. The hydraulic force acting on the pilot end is not big enough to overcome the forces acting on the back end of the spool. So the exhaust port of the spool valve is open. This is the other extreme of the control range. Obviously, the control range depends on the spring constant and preload and the spool valve geometry, in addition to the line pressure.

Similar trends are also observed in Figure 5.6 and Figure 5.7. Comparing Figure 5.7 with Figure 5.5 and Figure 5.6 reveals that the pressure control range is wider at higher line pressures. Also, Figure 5.5 and Figure 5.6 show good data repeatability.
Figure 5.5 Data set 110322: Pressure and force data versus solenoid input voltage

Figure 5.6 Data set 110327: Pressure and force data versus solenoid input voltage
Figure 5.7 Data set 130328: Pressure and force data versus solenoid input voltage

Figure 5.8 shows the measured volumetric leakage flow rates versus the feeding chamber pressure, under the three testing conditions. There are three leakage paths: valve leakage $Q_{dv}$, mounting surface leakage $Q_{dmt}$, and shaft end leakage $Q_{dse}$. The leakage flow rates are measured manually by collecting the leakage fluid for a certain period of time. The flow measurements are approximate, and the valve leakage and mounting surface leakage cannot be separated. Therefore, these two paths are measured as one lumped leakage $Q_{dvm}$. $Q_{dse}$ and $Q_{dvm}$ are plotted in the figures. The two sets of leakage data for the 110 psi (758.5 kPa) line pressure setting coincide well when the feeding chamber pressure $P_f$ is above 55 psi (379.2 kPa). The difference between the two traces at lower feeding chamber pressures is contributed by several factors. One factor is due to the way the leakage is measured. Data set “110322” is obtained by changing the input solenoid voltage from 5V to 0V, i.e. the pilot pressure changes from low to high. The exhaust port is closed in this process. Data set “110327” is obtained by changing the input solenoid voltage from 0V to 5V, i.e. the pilot pressure changes from high to low. In
this case, the clutch chamber is already filled. When the pilot pressure decreases, the
clutch exhausts fluid and the valve exhaust port may open temporarily. If the system has
not reached steady state, the measured leakage rate will have an error. In addition, though
the leakage data was taken with caution, the accuracy of the measurement is still limited.
The slight rise of leakage with decrease of the feeding chamber pressure below 35 psi
(241.3 kPa) in data set "110327" can be explained partly by these factors.

Figure 5.9 shows the computed leakage resistance versus the feeding chamber
pressure at the three testing conditions. The leakage resistance is calculated by the ratio
of the feeding chamber pressure and the volumetric leakage flow rate \( Q_{\text{volum}} \) assuming
laminar flow.

In summary, the identified parameters are listed below,

\[
K_s = 1400 \quad N/m
\]

\[
K_e = K_s + 0.43 \cdot \pi \cdot (P_i - P_f) \cdot 0.8
\]

\[
a_i = 1.88 \times 10^{-6} \quad m^2
\]

\[
F_{ko} = K_s \cdot x_{\text{vp}_o} = 12.376 \quad N
\]

![Figure 5.8 Comparison of measured leakage flow rates at different feeding chamber pressures](image)

Figure 5.8 Comparison of measured leakage flow rates at different feeding chamber pressures

246
5.5 Power-stage model validation

The power-stage dynamic model is validated by using measured line pressure and pilot pressure data as inputs to the simulation. The simulation is programmed using SIMULINK, the details being given in Appendix B. The simulation results for the feeding chamber and clutch chamber pressures are then compared with experimental results, as shown below. Table 5.1 lists the variable names and the symbols used in experiment, simulation and derivation.

As has been mentioned before, when the check ball location in the clutch outer chamber changes, as it does between successive neutral-idle transitions, the amount of air trapped inside the outer chamber also changes. This means that the effective bulk modulus of the outer clutch chamber is not known and varies with the check ball location and the manner of chamber filling. The compliance of the outer clutch chamber is therefore determined by adjusting the fluid stiffness $S_{co}$ so that the simulation results match the experimental data.
Step response tests for different levels of step inputs of voltage to the solenoid are conducted experimentally under two conditions chosen to represent extremes of check ball location, and the model validation results are presented for the two check ball locations mentioned below.

1. The check ball is at the top location, which results in the least amount of air being trapped inside the outer clutch chamber, i.e. the effective bulk modulus of the fluid/air mixture is maximum

2. The check ball is at the bottom location, which results in the greatest amount of air being trapped inside the outer clutch chamber, i.e. the effective bulk modulus of the fluid/air mixture is at a minimum.

Figure 5.10—Figure 5.14 show the comparison of simulation and experimental results using line pressure and pilot pressure data recorded in the data set “1302530” as inputs to the simulation. The input voltage is a low frequency square wave with voltage level alternating between 2.5 and 3.0 volts. This data set corresponds to the check ball being at the top location. Figure 5.10 shows the comparison of experimental and simulation results for the feeding chamber pressure. Figure 5.11 shows the comparison of outer clutch chamber pressures. Figure 5.12 compares the back chamber pressures obtained from experiment and simulation. The simulation results compare well with the

<table>
<thead>
<tr>
<th>Variable name</th>
<th>Symbol used in experiment</th>
<th>Symbol used in simulation</th>
<th>Symbol used in Section 5.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feeding chamber pressure</td>
<td>$P_{cf_exp}$</td>
<td>$P_f$</td>
<td>$P_f$</td>
</tr>
<tr>
<td>Valve spool back chamber pressure</td>
<td>$P_{b_exp}$</td>
<td>$P_{back}$</td>
<td>$P_b$</td>
</tr>
<tr>
<td>Inner chamber clutch pressure</td>
<td>$P_{ci_exp}$</td>
<td>$P_{c_i}$</td>
<td>$P_{ci}$</td>
</tr>
<tr>
<td>Outer chamber clutch pressure</td>
<td>$P_{co_exp}$</td>
<td>$P_{c_o}$</td>
<td>$P_{co}$</td>
</tr>
</tbody>
</table>

Table 5.1 List of symbols used in the experiment and simulation

248
experimental results. Figure 5.13 shows the measured pressures at different locations. Notice that the clutch inner chamber pressure is almost the same as the outer chamber pressure. Figure 5.14 shows the simulation results for the valve spool opening.

Figure 5.15 – Figure 5.19 show the comparison of simulation and experimental results using line pressure and pilot pressure data recorded in the data set “13d2530” as inputs to the simulation. The input voltage is again a low frequency square wave with voltage level alternating between 2.5 and 3.0 volts. The difference is that this data set corresponds to the check ball being at the bottom location. Figure 5.15 shows the comparison of experimental and simulation results for the feeding chamber pressure. Figure 5.16 shows the comparison of outer clutch chamber pressures. Figure 5.17 compares the back chamber pressures obtained from experiment and simulation. The simulation results compare well with the experimental results. Figure 5.18 shows the measured pressures at different locations. The clutch outer chamber pressure is clearly lagging the clutch inner chamber pressure. This indicates that the compliance effect of the outer clutch chamber due to air entrainment is more significant comparing to the case when the check ball is at the top location. Figure 5.19 shows the simulation results of the valve spool opening.

The location of the outer chamber check ball does seem to affect the amount of air trapped inside the outer clutch chamber and has a small effect on the step response of the neutral-idle valve. Figure 5.10 -Figure 5.19 show the comparison of simulation results and experimental results for the two check ball locations, and yield different values of the outer chamber compliance term. The other testing conditions are the same.
Figure 5.10 Comparison of simulation and experimental results for feeding chamber pressure: check ball at top location

Figure 5.11 Comparison of simulation and experimental results for outer chamber pressure: check ball at top location
Figure 5.12 Comparison of simulation and experimental results for spool valve back chamber pressure: check ball at top location

Figure 5.13 Measured pressures at different locations: check ball at top location
Figure 5.14 Simulation results for spool valve opening for check ball at top location

Figure 5.15 Comparison of simulation and experimental results for feeding chamber pressure: check ball at bottom location
Figure 5.16 Comparison of simulation and experimental results for clutch outer chamber pressure: check ball at bottom location

Figure 5.17 Comparison of simulation and experimental results for spool valve back chamber pressure: check ball at bottom location
Figure 5.18 Measured pressures at different locations: check ball at bottom location

Figure 5.19 Simulation results for spool valve opening: check ball at bottom location
When the check ball is at the top location, the compliance $C_{co}$ is $5 \times 10^{-13} \text{m}^3/\text{Pa}$. The compliance $C_{co}$ is $1.25 \times 10^{-12} \text{m}^3/\text{Pa}$ when the check ball is at the bottom location. This comparison indicates that there is more air trapped inside the outer clutch chamber when the check ball location is lower, as we would expect. The effective bulk modulus $\beta_e$ can be estimated as follows (Merritt, 1967).

\[ \frac{1}{\beta_e} = \frac{1}{\beta_c} + \frac{1}{\beta_i} + \frac{V_g}{V_{co}} \left( \frac{1}{\beta_s} \right) \]  

(5.23)

where

- $\beta_c$ is the container bulk modulus
- $\beta_i$ is the fluid bulk modulus, $\beta_i = 1.298 \times 10^9 \text{Pa}$ for automatic transmission fluid
- $\beta_s$ is the adiabatic bulk modulus of air,
  \[ \beta_s = 1.4P_g = 6.4 \times 10^5 \sim 1.0 \times 10^6 \text{Pa} \]  

(5.24)

- $V_g$ is the volume of the air trapped inside the outer clutch chamber

Since container bulk modulus is much higher, it is ignored in the calculation. $\beta_e$ is calculated as

\[ \beta_e = C_{co}^{-1} \cdot V_{co} = \begin{cases} 1.0257 \times 10^8 \text{Pa} & \text{checkball at higher position} \\ 4.1028 \times 10^7 \text{Pa} & \text{checkball at lower position} \end{cases} \]  

(5.25)

Then the percent of air trapped inside the outer clutch chamber is estimated by

\[ \frac{V_g}{V_{co}} = \frac{1}{\beta_e} \frac{1}{\beta_i} + \frac{1}{\beta_s} \begin{cases} 1.5118 \sim 2.2938\% & \text{checkball at lower position} \\ 0.5751 \sim 0.9124\% & \text{checkball at upper position} \end{cases} \]  

(5.26)

The above estimates indicate that more air is trapped inside the outer clutch chamber when the check ball is at the lower position. Notice that these estimates are approximate.
5.6 Pilot stage dynamic response identification

Step response tests and sinusoidal input tests both show that the pilot stage dynamic behavior is close to that of a second order system. A transfer function representation of the pilot stage dynamic response relating the solenoid input voltage to the pilot pressure is given by equation (5.27).

\[ G_z(s) = \frac{K_{sol} \omega_{ns}^2}{s^2 + 2\zeta \omega_{ns} + \omega_{ns}^2} \]  

(5.27)

where

- \( K_{sol} \) is the gain, kPa/volt
- \( \omega_{ns} \) is the natural frequency of the solenoid or linear force motor
- \( \zeta \) is the damping ratio of the solenoid or linear force motor

The "minus" sign indicates that when the solenoid voltage increases, the pilot pressure decreases. \( K_{sol} \) depends on the operating equilibrium point since the pilot pressure versus solenoid voltage relationship is nonlinear as indicated by Figure 5.5 – Figure 5.7. The dynamics characteristics such as the natural frequency and damping ratio may also be functions of the operating points. Therefore, step inputs with different offsets and amplitudes are sent to the system to identify the model parameters.

In order to identify the pilot stage dynamic model parameters, sinusoidal input tests and step response tests are conducted. The data collected at different operating conditions are analyzed. The data shown in this section are collected under the following conditions:

1. The system is warmed up and the hydraulic oil temperature is maintained at 88 – 90 °F, or 31.1 – 32.2 °C.
2. Line pressures are set at 120 and 130 PSI or 827.4 kPa and 896.4 kPa (at the pressure relief valve of the Hydraulic Power Unit).

The dynamic response of the pilot stage is not independent of the load represented by the second stage. Therefore, what is measured here is the "loaded" transfer function of the first stage. If the dynamic response of the first stage were analytically derived, the
The effect of the load by the second stage on the first stage could be included in the analysis. However, there is not enough information on the solenoid design to do analytical modeling. Figure 5.20 and Figure 5.21 show the measured pilot pressure, clutch outer chamber pressure and back chamber pressure for the two check ball locations. As shown by the figures, the pilot pressure plot looks similar to the back chamber pressure and clutch outer chamber pressure except for a scaling factor. This indicates that the second stage has a much faster response time than the first stage, and the loading effect is small at the lower frequencies.

5.6.1 Frequency response tests

Sinusoidal input tests are conducted. The offset of the input sine wave is 2.8 volts, and the signal amplitude is 0.2 volts, i.e. the input signal varies between 2.6 volts and 3.0 volts. The frequency of the input signal varies from 0.1 Hz to 20 Hz. The supply pressure is set at 120 psi or 827.4 kPa. The pilot stage model parameters are identified as follows:

\[ K_{so} = 1.72 \times 10^5 - 2.2 \times 10^5 \text{ Pa/volt} \]
\[ \zeta = 0.4 - 0.6 \]
\[ \omega_n = 7.2 - 7.3 \text{ Hz} \]

The identified parameters in equation (5.28) indicate that a gain value ranging from \(1.72 \times 10^5\) to \(2.2 \times 10^5\) Pa/volt, and a natural frequency between 7.2 and 7.3 Hz give good comparison of the frequency responses of the identified model and the experimental results. Figure 5.22 shows a series of Bode plots for the identified model (without the minus sign) for the different gain values together with the experimental results and indicates a satisfactory agreement. The effective gain decreases at higher frequencies. This suggests that the solenoid response has dynamic nonlinearities in addition to the static nonlinearity shown in Figure 5.5 – Figure 5.7.
Figure 5.20 Measured pilot pressure, clutch outer chamber pressure and back chamber pressure: checkball at the top location

Figure 5.21 Measured pilot pressure, clutch outer chamber pressure and back chamber pressure: checkball at the bottom location
5.6.2 Step response tests

Step response tests are conducted with a low frequency square wave as input to the solenoid. The input solenoid voltage and output pilot pressure are recorded. Figure 5.23 - Figure 5.26 compare step responses of the identified model and the experimental results for different step input magnitudes. The model gain $K_{sol}$, damping ratio $\zeta$, and natural frequency $\omega_n$ together with the input voltage are indicated in each figure. The supply pressure is set at 130 psi or 896.4 kPa for these responses.

Figure 5.23 shows a comparison of the identified model behavior with the experimental results when the input solenoid voltage changes between 2.5 and 3.0 volts, i.e., the offset of the input square wave is 2.75 volts, and the amplitude is 0.25 volts.

Figure 5.23(a) shows the comparison for both rising and falling edges of the step input,
and Figure 5.23(b) shows the magnified plot for the rising edge of the pilot pressure, which corresponds to the falling edge of the input voltage, i.e. when the solenoid voltage input changes from 3.0 volt to 2.5 volt. The identified model with a gain of $2 \times 10^5$ Pa/volt, damping ratio of 0.4, and a natural frequency of 7.3 Hz shows good agreement with the experimental results.

Figure 5.24 shows a comparison of the step response of the identified model with the experimental results when the solenoid input voltage changes between 2.8 and 3.0 volts. The offset of the input square wave is 2.9 volts, and the amplitude of the input voltage is 0.1 volts. The identified model with a gain of $2.14 \times 10^5$ Pa/volt, damping ratio of 0.4, and a natural frequency of 7.2 Hz shows good agreement with the experimental results.

Figure 5.25 shows a comparison of the step response of the identified model with the experimental results when the solenoid input voltage changes between 2.2 and 2.4 volts. This condition corresponds to the high pilot pressure range. The offset of the input square wave is 2.3 volts, and the amplitude of the input voltage is 0.1 volts. The identified model has a gain of $1.72 \times 10^5$ Pa/volt, damping ratio of 0.6, and a natural frequency of 7.2 Hz. Comparing Figure 5.24 and Figure 5.25, the difference is in the offset value of the input voltage. The gain and damping ratio of the identified models vary from $2.14 \times 10^5$ Pa/volt to $1.72 \times 10^5$ Pa/volt, and 0.4 to 0.6 respectively.

Figure 5.26 shows a comparison of the step response of the identified model and the experimental results, when the solenoid input voltage changes between 3.0 and 3.5 volts. This input corresponds to a low pilot pressure range. The identified model has a gain of $2.14 \times 10^5$ Pa/volt, damping ratio of 0.45, and a natural frequency of 7.3 Hz.

From the above comparisons, the effective gain increases as the mean value of the solenoid input voltage increases. This is contributed by the static nonlinearity between solenoid input voltage and pilot pressure as shown in Figure 5.5 – Figure 5.7. However, there is no systematic trend discernible in the damping ratio change.
Figure 5.23 Step response comparison between simulation and experimental data

Input: $V_{in} = 2.5 - 3.0\, \text{volts}$, Model $K_{sol} = 200\, \text{kPa} / \text{v}$, $\zeta = 0.4$, $\omega_n = 7.3\, \text{Hz}$

(a) Overall comparison          (b) Magnified plots
Figure 5.24 Step response comparison between simulation and experimental data

Input: $V_{in} = 2.8 - 3.0$ volts, Model $K_{sol} = 213.7 kPa/v$, $\zeta = 0.4$, $\omega_n = 7.2$Hz

(a) Overall comparison (b) Magnified plots
Figure 5.25 Step response comparison between simulation and experimental data

Input: $V_{in} = 2.2 - 2.4\,\text{volts}$, Model $K_{sol} = 172.4\,\text{kPa/\,v}$, $\zeta = 0.6$, $\omega_n = 7.2\,\text{Hz}$

(a) Overall comparison  (b) Magnified plots
Figure 5.26 Step response comparison between simulation and experimental data

Input: $V_{in} = 3.0 - 3.5 \text{volts}$, Model $K_{sol} = 213.7 \text{kPa}/\text{v}$, $\zeta = 0.45$, $\omega_{ns} = 7.3 \text{Hz}$

(a) Overall comparison  (b) Magnified plots
5.7 Linearized dynamic model of power stage

The model equations derived in Section 5.3 are nonlinear. In order to characterize the system behavior analytically for purposes of controller design, a linear model is obtained around an operating point. The operating point is denoted by

\[ x_{vo}, P_{po}, P_{bo}, P_{fo} \]

**Force balance on spool:**

\[
m_{v_{sp}} \cdot \ddot{x}_{v_{sp}} + B_{v_{sp}} \cdot \dot{x}_{v_{sp}} + K_{e} \cdot x_{v_{sp}} = P_{p} \cdot a_{sp1} - P_{b} \cdot a_{sp2} - F_{ko}
\]  

(5.29)

Letting \( x_{v} \) be the valve opening, i.e.

\[ x_{v} = x_{v_{sp}} - 4.0 \text{mm} \]  

(5.30)

Then, equation (5.29) can be written as

\[
m_{sp} \cdot \ddot{x}_{v} + B_{sp} \cdot \dot{x}_{v} + K_{e} \cdot x_{v} = P_{p} \cdot a_{sp1} - P_{b} \cdot a_{sp2}
\]  

(5.31)

where the effective spring constant \( K_{e} \) is a function of pressure difference as well.

\[ K_{e} = K_{e} + 0.43 \pi \cdot d_{sp2} \cdot (P_{i} - P_{f}) \cdot 0.8 \]  

(5.32)

Linearizing equation (5.31) for small perturbations from the operating point and dropping the symbol \( \delta \) representing the perturbations, for simplicity of notation, the resulting perturbation model is,

\[
m_{v_{sp}} \cdot \ddot{x}_{v} + B_{v_{sp}} \cdot \dot{x}_{v} + \left( K_{e} + 0.43 \pi \cdot d_{sp2} \cdot (P_{i} - P_{f}) \cdot 0.8 \right) x_{v} - \left( 0.43 \pi \cdot d_{sp2} \cdot x_{vo} \right) P_{f} = a_{sp1} P_{p} - a_{sp2} P_{b}
\]  

(5.33)

**Summary of flow model:**

\[ Q_{s} = C_{d} \cdot \pi \cdot d_{sp2} \cdot x_{v} \cdot \sqrt{\frac{2(P_{i} - P_{f})}{\rho}} \]  

(5.34)

\[ Q_{1} = Q_{c} + Q_{2} - (Q_{k_{av}} (P_{f}) + Q_{k_{km}} (P_{f})) = Q_{co} + Q_{kse}
\]  

(5.35)
\[ Q_2 = a_{sp2} \cdot x_v = C_d \cdot a_f \cdot \sqrt{\frac{2(P_b - P_f)}{\rho}} \]  
\( \text{(5.36)} \)

\[ P_x = P_f - \left( \frac{Q_1}{C_d \cdot a_1} \right)^2 \frac{\rho}{2} \]  
\( \text{(5.37)} \)

\[ Q_{co} = C_d \cdot a_{co} \cdot \sqrt{\frac{2(P_x - P_{co})}{\rho}} \]  
\( \text{(5.38)} \)

\[ Q_{co} = C_{co} \cdot P_{co} \]  
\( \text{(5.39)} \)

Linearizing equation (5.38), the resulting perturbation model is,

\[ Q_{co} = K_{sp} (P_x - P_{co}) \]  
\( \text{(5.40)} \)

where \[ K_{sp} = \left[ \frac{\partial Q_{co}}{\partial (P_x - P_{co})} \right]_{op} \]  
\( \text{(5.41)} \)

Substituting equation (5.40) into (5.39) and rearranging, we get,

\[ \frac{C_{co}}{K_{sp}} P_{co} + P_{co} = P_x \]  
\( \text{(5.42)} \)

Linearizing equations (5.34), (5.35), and (5.37), we get

\[ Q_1 = K_{sv} \cdot x_v + K_{sp} \cdot P_f \]  
\( \text{(5.43)} \)

\[ Q_1 = Q_x + Q_2 - K_f \cdot P_f = Q_{co} + K_{kxx} \cdot P_x \]  
\( \text{(5.44)} \)

\[ P_x = P_f - K_{xx} \cdot Q_1 \]  
\( \text{(5.45)} \)

where

\[ K_{sv} = \left[ C_d \cdot \pi \cdot d_{sp2} \cdot \sqrt{\frac{2(P_i - P_f)}{\rho}} \right]_{op} \]  
\( \text{(5.46)} \)

\[ K_{sp} = \left[ C_d \cdot \pi \cdot d_{sp2} \cdot x_v \cdot \frac{1}{\sqrt{2\rho(P_i - P_f)}} \right]_{op} \]  
\( \text{(5.47)} \)

\[ K_f = \left[ \frac{\partial Q_{dv} (P_f) + Q_{aam} (P_f)}{\partial P_f} \right]_{op} \]  
\( \text{(5.48)} \)
Substituting equations (5.43) - (5.49) into (5.42), we get,
\[
\frac{C_{co}}{K_{xp}} \dot{P}_{co} + P_{co} = \left(1 + K_{sq} (K_{yf} - K_{sp})\right) P_f - K_{sq} \left(K_{sv} x_v + a_{sp2} \cdot x_v\right) \tag{5.51}
\]

Linearizing equation (5.36) and denoting,
\[
K_{bp} = \left[C_d \cdot a_f \cdot \frac{1}{\sqrt{2 \rho (P_b - P_f)}}\right]_{op} \tag{5.52}
\]

we get,
\[
P_b = P_f + \frac{a_{sp2} \cdot x_v}{K_{bp}} \tag{5.53}
\]

Thus far, there are four linearized equations: (5.33), (5.44), (5.51) and (5.53), and there are five unknowns: \(x_v, P_p, P_b, P_f, P_{co}\) \(P_x\) is related to \(P_f\) and \(x_v\). Next, we will solve for \(P_{co}\) and \(P_b\) assuming \(P_p\) as input. Equation (5.44) is rewritten as:
\[
K_{sv} x_v + K_{sp} P_f + a_{sp2} x_v - K_{if} P_f = K_{sp} (P_x - P_{co}) + K_{laxe} P_x \tag{5.54}
\]

Rearranging equation (5.54), we get
\[
K_{sv} x_v + a_{sp2} x_v + (K_{sp} - K_{if}) P_f = (K_{sp} + K_{laxe}) P_x - K_{sp} P_{co} \tag{5.55}
\]

Substituting equation (5.45) into equation (5.55) and rearranging, we get,
\[
K_{sv} x_v + a_{sp2} \cdot x_v = \left[\frac{K_{sp} + K_{laxe}}{1 + K_{sq} (K_{sp} + K_{laxe})} - (K_{sp} - K_{if})\right] P_f - \frac{K_{sp}}{1 + K_{sq} (K_{sp} + K_{laxe})} P_{co} \tag{5.56}
\]

Substituting equation (5.56) into equation (5.51) and rearranging, we get,
Defining
\[ K_{fc0} = \frac{1}{1 + K_{sq} (K_{xp} + K_{kse})} \] (5.58)
and Laplace transforming equation (5.57), we get,
\[ \frac{P_{co}}{P_f} (s) = \frac{K_{fc0}}{K_{xp}} \frac{C_{co}}{s + (1 - K_{sq} K_{xp} K_{fc0})} \] (5.59)
Substituting equation (5.59) into equation (5.56), and defining
\[ m_1 = K_{fx} S_{co} K_{xp} (1 - K_{sq} K_{xp} K_{fc0}) - K_{fc0}^2 S_{co} K_{xp}^2 \] (5.61)
\[ m_2 = S_{co} K_{xp} (1 - K_{sq} K_{xp} K_{fc0}) \] (5.62)
we get,
\[ \frac{x_v}{P_f} (s) = \frac{K_{fx} s + m_1}{(a_{sp2} s + K_{sv}) (s + m_2)} \] (5.63)
Substituting equations (5.63) and (5.53) into equation (5.33), and eliminating \( P_b \), we get,
\[ \frac{P_f}{P_p} (s) = \frac{a_{sp1} (a_{sp2} s + K_{sv}) (s + m_2)}{(K_{fx} s + m_1) \left[ m_{vp} s^2 + \left( B_{vp} + \frac{a_{sp2}^2}{K_{bp}} \right) s + K_{co} \right] - m_3 (a_{sp2} s + K_{sv}) (s + m_2)} \] (5.64)
where \( m_3 = 0.43 \pi d_{sp2} x_v - a_{sp2} \) (5.65)
Combining equations (5.64) and (5.59), we get,
\[ \frac{P_{co}}{P_p} (s) = \frac{K_{fc0} S_{co} a_{sp1} (a_{sp2} s + K_{sv})}{(K_{fx} s + m_1) \left[ m_{vp} s^2 + \left( B_{vp} + \frac{a_{sp2}^2}{K_{bp}} \right) s + K_{co} \right] - m_3 (a_{sp2} s + K_{sv}) (s + m_2)} \]
\[ = \frac{n_1 (n_2 s + 1)}{n_3 s^3 + n_4 s^2 + n_5 s + n_6} \] (5.66)
268
Similarly, we get the transfer function relating $P_b$ to $P_p$

$$\frac{P_b}{P_p}(s) = \frac{a_{sp1}a_{sp2}\left(1 + \frac{K_{fv}}{K_{bp}}\right)s^2 + \left(\frac{K_{sv}}{a_{sp2}} + m_2 + \frac{m_1}{K_{bp}}\right)s + \frac{K_{sv}m_2}{a_{sp2}}}{(K_{fv}s + m_1)\left[m_{vp}s^2 + \left(B_{vp} + \frac{a_{sp2}^2}{K_{bp}}\right)s + K_{eo}\right] - m_3(a_{sp2}s + K_{sv})(s + m_2)}$$

$$= \frac{n_7s^2 + n_8s + n_9}{n_3s^3 + n_4s^2 + n_5s + n_6}$$

(5.67)

where $K_{eo} = [K_{eo}]_{op}$

(5.68)

$$n_1 = K_{fv}S_{co} K_{sp} a_{sp1} K_{sv}$$

(5.69)

$$n_2 = \frac{a_{sp2}}{K_{sv}}$$

(5.70)

$$n_3 = K_{fv} m_{vp}$$

(5.71)

$$n_4 = m_1 m_{vp} + K_{fv} \left( B_{vp} + \frac{a_{sp2}^2}{K_{bp}} \right) - m_3 a_{sp2}$$

(5.72)

$$n_5 = m_1 \left( B_{vp} + \frac{a_{sp2}^2}{K_{bp}} \right) + K_{eo} K_{fv} - m_3 m_2 a_{sp2} - m_3 K_{sv}$$

(5.73)

$$n_6 = m_1 K_{eo} - m_3 m_2 K_{sv}$$

(5.74)

$$n_7 = a_{sp1} a_{sp2} \left(1 + \frac{K_{fv}}{K_{bp}}\right)$$

(5.75)

$$n_8 = a_{sp1} a_{sp2} \left(\frac{K_{sv}}{a_{sp2}} + m_2 + \frac{m_1}{K_{bp}}\right)$$

(5.76)

$$n_9 = a_{sp1} K_{sv} m_2$$

Table 5.2 lists two sets of operating conditions based on simulation and the experimental data set "13d2530", in which the check ball is at the bottom location. The two operating points are selected to correspond to the instants before and after the pilot
pressure starts rising as shown in Figure 5.27. The energy stored in each element corresponding to the operating points are computed, in order to identify the dominant energy storage elements.

Substituting numerical values at the operating conditions shown in Table 5.2, we get the poles and zeros for the transfer functions (5.66) and (5.67), and are listed in Table 5.3. Bode plots of these two transfer functions at point A are shown in Figure 5.28.
The Bode plot of $\frac{P_{co}}{P_p}(s)$ shows dominant first order dynamic behavior at frequencies below 1000 rad/sec. The relative significance of the different dynamic effects is indicated by the energy stored in the valve spool mass and spring, and clutch chamber compliance, at the operating condition B, these values being $5.5599 \times 10^{-9}$ J, $3.9848 \times 10^{-5}$ J, and 0.1072 J respectively. Therefore, the clutch pressure dynamic response is dominant and we can ignore valve spool dynamic effects, i.e. $m_{vp} = 0, K_e = 0$. With this simplification, the reduced order model of $\frac{P_{co}}{P_p}(s)$ is:

$$\frac{P_{co}}{P_p}(s) = \frac{K_{fco}S_{co}K_x a_{sp1}(a_{sp2}s + K_{sv})}{(K_{fvp}s + m_1)\left[B_{vp} + \frac{a_{sp2}^2}{K_{vp}}\right]s + a_{sp2}(a_{sp2}s + K_{sv})(s + m_2)}$$ (5.77)

Figure 5.29 shows the Bode plots of the full order and the reduced order transfer function forms of $\frac{P_{co}}{P_p}(s)$ at point A. The result confirms the above analysis that the clutch chamber dynamics is dominant. Then,

$$\tau_{co} \cdot P_{co} + P_{co} = K_{co}P_p$$ (5.78)

Equation (5.78) describes the simplified dynamic response of the power stage spool valve. It is a first order, type 0 system with pilot pressure as the input and the outer clutch chamber pressure as output. If there was no leakage in the system, the DC gain of the system would be higher.
<table>
<thead>
<tr>
<th>Variable</th>
<th>Illustration</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho$</td>
<td>ATF density</td>
<td>858.7</td>
<td>Kg/m$^3$</td>
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<tr>
<td>$x_{va}$</td>
<td>Valve spool opening</td>
<td>8.03x10^{-5}</td>
<td>m</td>
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<tr>
<td>$P_l$</td>
<td>Line pressure</td>
<td>848.1</td>
<td>kPa</td>
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<td>$P_{fo}$</td>
<td>Feeding chamber pressure</td>
<td>419.57</td>
<td>kPa</td>
</tr>
<tr>
<td>$P_{xo}$</td>
<td>Intermediate pressure</td>
<td>418.74</td>
<td>kPa</td>
</tr>
<tr>
<td>$P_{bo}$</td>
<td>Back chamber pressure</td>
<td>419.58</td>
<td>kPa</td>
</tr>
<tr>
<td>$P_{clo}$</td>
<td>Inner clutch chamber pressure</td>
<td>407.52</td>
<td>kPa</td>
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<td>$P_{co-o}$</td>
<td>Outer chamber clutch pressure</td>
<td>414.11</td>
<td>kPa</td>
</tr>
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<td>$x_{vp-o}$</td>
<td>Valve spool velocity</td>
<td>9.63x10^{-4}</td>
<td>m/s</td>
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<td>$Q_{zo}$</td>
<td>Back chamber flow</td>
<td>8.03x10^{-8}</td>
<td>m$^3$/s</td>
</tr>
<tr>
<td>$Q_{lvm-o}$</td>
<td>Valve and mounting leakage</td>
<td>4.54x10^{-6}</td>
<td>m$^3$/s</td>
</tr>
<tr>
<td>$S_{co}$</td>
<td>Outer clutch chamber compliance coefficient</td>
<td>8x10^{11}</td>
<td>Pa/m$^3$</td>
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<td>$K_{sp}$</td>
<td>Flow gain</td>
<td>2.2x10^{-10}</td>
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<td>$K_{sv}$</td>
<td>Flow gain</td>
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<td>$K_{sp}$</td>
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<td>$K_{sq}$</td>
<td>Flow gain</td>
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<td>Pa·s/m$^3$</td>
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<td>$K_{bp}$</td>
<td>Flow gain</td>
<td>4.53x10^{-9}</td>
<td>m$^3$/Pa·s</td>
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<tr>
<td>$K_{lke}$</td>
<td>Leakage gain</td>
<td>8x10^{-12}</td>
<td>m$^3$/Pa·s</td>
</tr>
<tr>
<td>$K_{lf}$</td>
<td>Leakage gain</td>
<td>8.70x10^{-12}</td>
<td>m$^3$/Pa·s</td>
</tr>
</tbody>
</table>

Table 5.2 Model parameter at the operating conditions chosen

272
\[ \prod_{i} \frac{(s - z_i)}{P_{co}} \prod_{j} \frac{1}{(s - p_j)} \]

<table>
<thead>
<tr>
<th></th>
<th>Point A</th>
<th>Point B</th>
<th>Point A</th>
<th>Point B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pole ( p_j ) (rad/sec)</td>
<td>((-2.0 + 3.63i) \times 10^3)</td>
<td>((-2.62 + 4.22i) \times 10^3)</td>
<td>((-2.0 + 3.63i) \times 10^3)</td>
<td>((-2.62 + 4.22i) \times 10^3)</td>
</tr>
<tr>
<td>Zero ( z_i ) (rad/sec)</td>
<td>(-7.35 \times 10^2)</td>
<td>(-7.35 \times 10^3)</td>
<td>(-7.08 \times 10^3)</td>
<td>(-6.94 \times 10^3)</td>
</tr>
<tr>
<td>Gain ( k ) (Dimension-less)</td>
<td>(6.92 \times 10^5)</td>
<td>(5.46 \times 10^3)</td>
<td>(5.12 \times 10^3)</td>
<td>(7.48 \times 10^3)</td>
</tr>
</tbody>
</table>

Table 5.3 Poles and zeros of transfer functions \( \frac{P_{co}}{P_p}(s), \frac{P_b}{P_p}(s) \)

Figure 5.28 Bode plots of \( \frac{P_{co}}{P_p}(s) \)(solid line) and \( \frac{P_b}{P_p}(s) \)(dashed line)

273
Figure 5.29 Bode plots of \[ \frac{P_{co}}{\tau_p} (s) \]: full order (solid line) and reduced order (dashed line)

Figure 5.30 shows the simulated step response of the power stage spool valve. The input to the nonlinear simulation is a step change of pilot pressure from 294.4 to 310.3 kPa. The line pressure is 841.2 kPa, and the check ball is assumed to be at the bottom location. The computed time constant \( \tau_{co} \) is 7.0 and 12.9 ms for the two operating conditions given in Table 5.2 respectively, which is compatible with the step response data shown in Figure 5.30.

Combining the pilot stage model and the power stage model, the neutral-idle valve dynamic model (perturbation model) is given by equation (5.79).

\[
\frac{P_{co}}{V_{sol}} = \frac{K_{sol} \cdot K_{co}}{s^2 + \frac{2\zeta_s}{\omega_n} + 1} \left( \frac{s^2}{\omega_n^2} + \frac{2\zeta_s}{\omega_n} + 1 \right) \tau_{co} (s + 1) \tag{5.79}
\]

Comparing the pilot stage dynamic response with the power stage dynamic response, it is seen that the pilot stage dynamic response is slower. The equivalent linear
characteristics of both stages change with operating conditions. For example, when the pressure $P_{co}$ decreases in the outer clutch chamber, the effective bulk modulus decreases. The outer clutch chamber compliance coefficient $C_{co}$ increases with the decrease of effective bulk modulus, and the power stage time constant $\tau_{co}$ will increase. This discussion is only qualitative. Quantitative characterization of this change is hard to do, due to uncertainty in determining the amount of air trapped inside the outer clutch chamber. A more detailed discussion of the variation of model parameters will be given in the next section.

5.8 Closed-loop pressure control strategy

For the neutral-idle control application, fast and repeatable increase of clutch pressure is necessary for good coordination between engine and transmission.
manipulation. In the previous sections, the dynamic behavior of a neutral-idle pressure control system has been examined.

In an open loop control scheme, to ensure satisfactory tracking for fast inputs, the plant dynamics have to be taken into account. Furthermore, to compensate for the plant dynamics under open loop control, the plant dynamics have to be inverted. This is impractical in practice. In addition, the dynamic characteristics of the pressure control system change with operating conditions. As discussed before, for example, the amount of the air trapped inside the outer clutch chamber varies with the check-ball location, which changes the dynamic characteristics of the system. A pre-calculated static open loop control will not be able to provide accurate and repeatable tracking performance in the presence of these changes. Therefore, closed-loop control is desired to reduce this effect and make system more robust to these changes in order to achieve satisfactory tracking performance.

Figure 5.31 shows the proposed controller structure. The variable names used in the figure are defined below.

![Diagram of Two degree-of-freedom controller structure]

Figure 5.31 Two degree-of-freedom controller structure
\( P_{b, \text{desired}} \) is the desired back chamber pressure trajectory

\( P_{b, \text{measured}} \) is the measured back chamber pressure

\( P_{b, \text{IC}} \) is the initial condition of the desired trajectory

\( \Delta P_{b} \) is the tracking error

\( G_{e, \text{ff}}(s) \) is the feedforward controller transfer function

\( G_{e, \text{fb}}(s) \) is the cascade controller transfer function

\( G_{p}(s) \) is the linearized plant model

\( V_{sol} \) is the command solenoid voltage

\( V_{sol, \text{IC}} \) is the initial solenoid voltage

\( K_{scale} \) is the scaling factor needed to convert the pressure sensor signal to the equivalent pressure value

The proposed controller includes feedforward and feedback controllers. It is said to have two degrees of design freedom (Horowitz, 1963). In theory, the feedback controller can be chosen to yield a desired level of stability robustness or disturbance rejection. The feedforward controller can then be chosen to yield a desired reference input tracking performance. Thus, use of dynamic controllers in the feedforward and cascade locations allow achievement of reference input tracking performance independent of stability robustness or disturbance rejection.

There are five pressure sensors in the experimental setup. The back chamber pressure is chosen as the feedback signal, since it is easier to install a pressure sensor in the valve in a production system as compared to instrumenting the clutch. This pressure control loop is going to be an inner loop in the neutral-idle application and the outer loops will be closed by measuring the turbine and engine speeds and controlling them closed loop.

The feedforward controller uses the desired trajectory to compute the required solenoid input voltage based on the nominal neutral-idle valve characteristics. This is the essence of any open loop control strategy and is sensitive to modeling error. The closed
loop controller compensates for modeling error and reduces the tracking error encountered. The advantage of feedforward control is that it brings the tracking error within a reasonable range without relying on a large loop gain and the attendant loss of stability robustness. The closed-loop control is intended to compensate for the much smaller amount of error remaining despite the control action. Without feedforward control, the feedback control effort required to reduce the tracking error is large and may cause stability robustness problems.

The static relationship between the desired clutch pressure and solenoid input voltage is nearly linear in the range of interest for operation. Therefore, a pre-calibrated gain is used for the feedforward control. Two forms of the cascade controller are considered for feedback control. One of the controllers involves an integrator together with a first-order phase lead controller, referred to here as "lead + integration" control, the other one is a "PID" controller. The design of these two controllers and experimental results for the implemented closed-loop control scheme are described below.

5.8.1 Controller design

The plant transfer function relating the back chamber pressure to pilot pressure identified in section 5.7 is faster than the solenoid dynamics and is omitted here for convenience. The plant model relating the back chamber pressure to solenoid input voltage is given by

$$G_p(s) = \frac{K_{sol} \cdot K_{bpl}}{s^2 + \frac{2\zeta\omega_n}{\omega_n^2} + 1}$$  \hspace{1cm} (5.80)$$

where $K_{bpl} = 2.1196$ (dimensionless) is obtained at operating point A shown in Table 5.2 and Table 5.3.

The feedforward controller design is based on the static experimental mappings relating the pilot pressure and the back chamber pressure to the solenoid input voltage and shown in Figure 5.32. "110" and "130" stand for line pressure settings of 110 psi (785.5 kPa) and 130 psi (896.4 kPa) respectively. The steady state gain from the solenoid
Figure 5.32 Relationship of neutral-idle valve pressures to solenoid input

Voltage input to pilot pressure and back chamber pressure are listed in the figure, and the gains at 130 psi (896.4 kPa) are:

\[
K_{Pb\_Vin} = -414.3914 \text{ kPa/V} \\
K_{Pp\_Vin} = -205.3549 \text{ kPa/V}
\] (5.81)

The desired pressure trajectory \( P_{b\_desired} \) is a terminated ramp signal. In order to track this type of signal with zero steady state error, a free "s" needs to be incorporated into the controller. So the modified plant with an integrator becomes,

\[
G_{ps}(s) = \frac{K_{sol} \cdot K_{bpilot}}{s^2 + \frac{2\zeta_s}{\omega_n} + 1} \] (5.82)
The range of solenoid model parameters are given in equation (5.28), and the following set of parameters are chosen as nominal plant parameters to design controllers.

\[ K_{sol} = 206.85 \text{ kPa/V} \]
\[ \zeta = 0.4 \]
\[ \omega_{nr} = 7.3 \text{ Hz} = 45.8673 \text{ rad/s} \]  

(5.83)

Figure 5.33 shows the Bode plot for the transfer function shown in equation (5.83) (Note that the “minus” sign is not included). The plant-integrator combination is unstable.

The design objective is to achieve a velocity error coefficient of 10, and a phase margin of 50°. A lead controller is cascaded with the integrator as follows:

\[ G_{c-ud}(s) = K_c \frac{T_s + 1}{s(\alpha T_s + 1)} \quad \alpha < 1 \]

(5.84)

where

\[ \alpha = 0.1 \]
\[ K_c = -0.1667 \text{ v/(psi} \cdot s) = -2.4172e-5 \text{ v/(Pa} \cdot s) \]
\[ T = 0.02 \text{ sec} \]

(5.85)

Figure 5.33 Bode plot of \( G_{pu}(s) \)
Figure 5.34 shows the Bode plot of the plant with the controller: $G_{c_{-id}}(s) \cdot G_p(s)$. The phase margin and gain margin are 89.6° and 16.5 dB respectively. The velocity error coefficient is 20.5 dB or 10.6, which satisfies the velocity error design specification.

The second cascade controller is a conventional PID controller. The controller has the following form,

$$G_{c_{-pid}}(s) = k \left( K_p + \frac{K_i}{s} + \frac{K_d N \cdot s}{s + N} \right)$$

(5.86)

where

$$k = \frac{1}{K_{Pb_{-vel}}} = -2.4 \times 10^{-6} \quad v/Pa$$

$$K_p = 1 \quad \text{dimensionless}$$

$$K_i = 10 \quad \text{(rad/s)}$$

$$K_d = 0.075 \quad \text{dimensionless}$$

$$N = 100 \quad \text{dimensionless}$$

Figure 5.35 shows the Bode plot of the plant with the PID controller: $G_{c_{-pid}}(s) \cdot G_p(s)$. The phase margin is 50.2 degrees, and the gain margin is infinity in the

---

Figure 5.34 Bode plot of $G_{c_{-id}}(s) \cdot G_p(s)$
frequency range of interest. The velocity error coefficient is 20.6 dB, or 10.7, which satisfies the design specification. The Bode plot of the closed loop system with “feedforward+Integral+Lead” and “feedforward+PID” controllers is shown in Figure 5.36, and has bandwidths of 10 rad/sec and 100 rad/sec respectively. This indicates that the “feedforward+PID” controller renders better performance.

5.8.2 Implementation and experimental results

In this part, four pressure trajectories are given as shown in Figure 5.37 and Table 5.4. The control objective is to ensure that the back chamber pressure accurately tracks the given pressure trajectories. Three control schemes: feedforward control alone, feedforward plus “lead+integration” feedback, and feedforward plus PID feedback are implemented in the experimental setup and the tracking results are compared for each given trajectory. Among these three trajectories, trajectory B is the most severe case, as it involves the largest slope.

The dSPACE DS1102 Floating-Point Controller Board has two 16-bit 250 KHz sampling A/D converters and two 12-bit 800 KHz sampling A/D converters (dSPACE, 1996). The sampling time used for the controller implementation in the experimental setup is 0.1 ms (10 KHz). The sampling frequency is much higher than the system frequency. Therefore the discrete time control implementation can duplicate the behavior of the continuous time compensation.
Figure 5.35 Bode plot of $G_{e-pid}(s) \cdot G_p(s)$

Figure 5.36 Bode plot of the designed closed loop system with “Lead+integration” (solid line) and “PID” (dashed line) controllers
Figure 5.37 Pressure trace

<table>
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<tr>
<th>Trace parameters</th>
<th>Type A</th>
<th>Type B</th>
<th>Type C</th>
<th>Type D</th>
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<td>Initial P (kPa) – $P_0$</td>
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<td>137.9</td>
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<td>275.8</td>
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<td>Step (kPa) – $\Delta P$</td>
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<td>1.8961e3</td>
<td>517.125</td>
<td>861.875</td>
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</table>

Table 5.4 Pressure trace

(a) Trajectory A

Figure 5.38 – Figure 5.43 show the experimental results corresponding to trajectory A. Figure 5.38 shows the comparison of experimental results with “feedforward”, “feedforward + lead+integral”, and “feedforward+PID” control respectively. When there is only feedforward control, there is a steady state error as the model used for computing controller gain is approximate.

The tracking error is shown in the magnified view in Figure 5.38(b). The addition of PID control to feedforward control gives the best performance. The tracking error is smaller than those for the other two control methods. The overshoot is also the lowest among the three methods compared.
Figure 5.39 shows the measured control effort, i.e. the solenoid input voltage, for the three control algorithms. The control action is within the operating range of the device (0-5V). Figure 5.40 shows the comparison of tracking error using the three control methods. It is clear that “feedforward+PID” control has the smallest tracking error. The error also converges to zero faster than for the other controllers. Figure 5.41 – Figure 5.43 show the back chamber pressure, feeding chamber pressure, inner and outer clutch chamber pressures under these three control algorithms. The gap between back chamber pressure and inner clutch chamber pressure is around 20.7 – 24.1 kPa. The lag between the inner and outer clutch chamber pressures is large compared to that shown in Figure 5.10 – Figure 5.17. This is probably due to the setting of low initial pressure, in this case 137.9 kPa. At this pressure, the clutch starts exhausting through the partially open exhaust port 4 in the power-stage valve (Figure 5.3), which requires clutch filling during the pressure rise. For the Type C and Type D trajectories to be shown later in this section, the lag between the inner and outer clutch chamber pressures is much smaller as the pressure levels are higher. In the neutral-idle application, the clutch remains filled during the neutral-idle cycle. The lowest clutch pressure that can be maintained in the forward clutch assembly without resulting in poor dynamic response depends on valve parameters. If a lower clutch pressure is desired, a lower valve spool spring preload is needed.

(b) Trajectory B

Figure 5.44 – Figure 5.49 show the experimental results for trajectory B using “feedforward”, “feedforward+integral+lead”, and “feedforward+PID” control.

Figure 5.44 shows the comparison of the three control methods. As was the case for the results presented for trajectory A, “feedforward+PID” gives the best tracking performance. Figure 5.45 shows the control effort. Figure 5.46 shows the comparison of the tracking error using different control schemes. With feedforward control only, the steady state tracking error is 51.7 kPa, which is higher than that corresponding to trajectory A. However, the tracking errors with the controllers “feedforward+integral+lead”, and “feedforward+PID” go to zero. “Feedforward+PID”
control gives the lowest overshoot and fastest convergence to steady state. Figure 5.47 — Figure 5.49 show the back chamber pressure, feeding chamber pressure, inner and outer clutch chamber pressures using these three control methods. The difference between the back chamber pressure and the inner clutch chamber pressure is about 20.7 kPa, which is the same as that shown for the results corresponding to trajectory A.

(c) Trajectory C

Figure 5.50 — Figure 5.55 show the experimental results corresponding to trajectory C. As was the case for the above two trajectories, the controller “feedforward+PID” gives the best tracking performance.

Figure 5.52 shows the tracking error. With feedforward control alone, the tracking error is negative instead of being positive as shown in Figure 5.40. This indicates that with feedforward control alone, the tracking accuracy in steady state can not be guaranteed. However, “feedforward +integral+lead” and “feedforward+PID” control both give zero steady state error. “Feedforward+PID” control gives the best tracking performance. The lag between the inner and outer clutch chamber pressures is smaller than that corresponding to Type A and B trajectories. The difference is due to the higher initial pressure setting at 206.85 kPa. In this case, the clutch chamber remains filled. The lag caused by clutch filling is reduced.

(d) Trajectory D

Figure 5.56 — Figure 5.61 show the experimental results corresponding to trajectory D. As was the case for the above two trajectories, the controller “feedforward+PID” gives the best tracking performance. Figure 5.56 shows the tracking error. With feedforward control alone, the tracking error is largest. However, the “feedfoward +integral+lead” and “feedforward+PID” controllers show oscillatory behavior indicating the need for gain adjustment. “Feedforward+PID” control gives the best tracking performance. As was the case for the Type C trajectory, the lag between the inner and outer clutch chamber pressures is smaller than that corresponding to Type A and B trajectories. The difference is due to the higher initial pressure setting at 275.8 kPa.
In this case, the clutch chamber remains filled. The lag caused by clutch filling is reduced.

Experimental results are presented for trajectory tracking using the three control schemes. The four command trajectories used for controller evaluation have different slopes. Comparing the three control algorithms mentioned above, "feedforward+PID" control works best. It gives the lowest tracking error and smallest overshoot and fastest response. Feedward plus feedback control can achieve satisfactory tracking. Combination of feedforward and feedback control recommended achieves good tracking accuracy without relying excessively upon feedback control.
Figure 5.38 Comparison of back chamber pressure under open loop and closed-loop control results for trajectory A: (a) Overview, (b) Magnified view
Figure 5.39 Comparison of control effort: solenoid input voltages under different control algorithms for trajectory A

Figure 5.40 Comparison of error under different control algorithms for trajectory A
Figure 5.41 Pressures under feedforward control for trajectory A

(a) Overview    (b) Magnified view
Figure 5.42 Pressures under “Feedforward+Integral+Lead” control for trajectory A

(a) Overview  (b) Magnified view
Figure 5.43 Pressures under "Feedforward+PID" control for trajectory A

(a) Overview  (b) Magnified view
Figure 5.44 Comparison of back chamber pressure under different control algorithms for trajectory B: (a) Overview, (b) Magnified view
Figure 5.45 Comparison of control effort: solenoid input voltage under different control algorithms for trajectory B

Figure 5.46 Comparison of error under different control algorithms for trajectory B
Figure 5.47 Pressures under "Feedforward" control for trajectory B

(a) Overview  (b) Magnified view
Figure 5.48 Pressures under “Feedward+Integral+Lead” control for trajectory B

(a) Overview  (b) Magnified view
Figure 5.49 Pressures under "Feedforward+PID" control for trajectory B
(a) Overview  (b) Magnified view
Figure 5.50 Comparison of back chamber pressure under different control algorithms for trajectory C

(a) Overview    (b) Magnified view
Figure 5.51 Comparison of control effort: solenoid input voltage under different control algorithms for trajectory C

Figure 5.52 Comparison of error under different control algorithms for trajectory C
Figure 5.53 Pressures under “Feedforward” control for trajectory C

(a) Overview  (b) Magnified view

300
Figure 5.54 Pressures under "Feedforward+Integral+Lead" control for trajectory C

(a) Overview  (b) Magnified view
Figure 5.55 Pressures under “Feedforward+PID” control for trajectory C

(a) Overview  (b) Magnified view
Figure 5.56 Comparison of back chamber pressure under different control algorithms for trajectory D

(a) Overview  (b) Magnified view
Control effort: solenoid input voltage

Figure 5.57 Comparison of control effort: solenoid input voltage under different control algorithms for trajectory D

Error ($\Delta P_b$)

Figure 5.58 Comparison of error under different control algorithms for trajectory D
Figure 5.59 Pressures under "Feedforward" control for trajectory D
(a) Overview  (b) Magnified view
Figure 5.60 Pressures under "Feedforward+Integral+Lead" control for trajectory D
(a) Overview  (b) Magnified view
Figure 5.61 Pressures under "Feedforward+PID" control for trajectory D
(a) Overview  (b) Magnified view
5.9 Sensitivity study

The controller design in section 5.8 is based on a nominal linearized dynamic model of the plant. Closed-loop control is implemented in the test setup and shown to achieve satisfactory performance. However, it is inevitable that the model we use is an approximation of the true dynamic behavior under given operating conditions. The objective of controller design is to have stable performance despite substantial variations in plant dynamics under all operating conditions. In this section, the stability robustness of controller design is examined.

The model uncertainty is defined as:

\[ t(s) = \frac{G(s) - G_o(s)}{G_o(s)} \]  

(5.88)

where

\[ G_o(s) \] is the nominal plant transfer function

\[ G(s) \] is the true plant transfer function

The sensitivity function is defined as (Franklin et al., 1994):

\[ S(s) = (1 + G_c \cdot G)^{-1} \]  

(5.89)

The complementary sensitivity function is defined as

\[ T(s) = (1 + G_c \cdot G)^{-1} G \cdot G_c \]  

(5.90)

Notice that,

\[ S(s) + T(s) = 1 \]  

(5.91)

The stability robustness measure is defined as (Franklin et al., 1994):

\[ \delta_{SR} = \left| 1 + (G_c \cdot G_o)^{-1} \right| \]  

(5.92)

\[ \delta_{SR} \] is simply the inverse of the closed-loop magnitude frequency response in the absence of feedback compensation and feedforward control. For the system to be stable under all operating conditions, if the system is stable under the nominal condition,

\[ |T(s)^{-1} > |t(s)| \]  

(5.93)
For the system considered, the following changes are likely to significantly affect the plant dynamics.

(1) Temperature change: 0°C – 150°C
(2) Checkball location: up - bottom
(3) Line pressure change: 620.55 – 1103.2 kPa
(4) Operating point change: 137.9 – 827.4 kPa.

Temperature change will affect fluid properties such as fluid density and viscosity. These changes will in turn affect the valve leakage characteristics. The effect of checkball location is discussed in the previous sections. Its main effect is on the amount of air trapped inside the outer clutch chamber, which affects the compliance of the outer clutch chamber. When line pressure and operating conditions change, the model parameters also change.

The variation of the solenoid dynamics depends on operating conditions. Due to the limited access to design structure and parameters of the solenoid, the solenoid model and parameters are identified in section 5.6 under the nominal testing conditions. In this section, consideration of solenoid dynamics variation is based on the testing results shown in section 5.6.

From the measured leakage data given in Figure 5.8 and Figure 5.9, the following approximation of the total leakage characteristics as a function of line pressure and feeding chamber pressure is obtained:

\[ Q_{\text{leak}} = (Q_{\text{lvem}} + Q_{\text{lve}}) \cdot \frac{\mu_{90\,\text{F}}}{\mu_{90\,\text{F}}} \]

\[ = \left(1 \times 10^{-7} \cdot \left(\frac{P_f}{6895 - 30} \right) + 6 \times 10^{-6} + \frac{P_l}{6895 - 110} \times 10^{-6} \right) \cdot \frac{\mu_{90\,\text{F}}}{\mu_{90\,\text{F}}} \]

where

- \( Q_{\text{leak}} \) is the total leakage, \( m^3/s \)
- \( P_f \) is the feeding chamber pressure, \( Pa \)
- \( P_l \) is the line pressure, \( Pa \)
- \( \mu_{90\,\text{F}} \) is the fluid viscosity at 90°F (32.2°C), \( Pa \cdot sec \)
\( \mu_{at} \) is the fluid viscosity at \( T_c \) (°C), \( Pa \cdot sec \)
\[
\log_{10}\left( \frac{\mu_{at}}{\rho_{at}} + 0.7 \right) = 10^{(-3.146 \log_{10}(273.15+T_c)+8.044)} \tag{5.95}
\]

\( \rho_{at} \) is the fluid density at \( T_c \) (°C), \( Kg/m^3 \)
\[
\rho_{at} = (-6.1e^-4 \times T_c + 0.8784) * 1000 \tag{5.96}
\]

Equations (5.95) and (5.96) are given by Kemp and Linden (1990).

Figure 5.62 shows the sensitivity function of the system with PID and “Integration+Lead” controllers. Figure 5.63 and Figure 5.64 show the stability robustness measure and model uncertainty when the fluid temperature is changed to 0°C and 150°C respectively. Other conditions are unchanged. As shown in the figure, the stability robustness measure is always greater that the model uncertainty. Therefore, the controlled systems with the PID and “lead+integration” controllers are stable for the temperature variations considered.

![Sensitivity functions of system with PID and lead(with I) controllers](image)

Figure 5.62 Sensitivity function of the system with PID and “Lead+Integration” controllers
Figure 5.63 Stability robustness measure of PID and "Lead+Integrator" controller when temperature is changed to 0°C

Figure 5.64 Stability robustness measure of PID and "Lead+Integrator" controllers when the temperature changes to 150°C
Figure 5.65 shows the stability robustness measure and the model uncertainty caused by solenoid dynamics variation identified in section 5.6. In this case, the rest of the operation conditions remain unchanged. The two model uncertainty curves $l_{o_{\text{max}}}$ and $l_{o_{\text{min}}}$ correspond to the maximum and minimum model variations for the parameters $K_{\text{sol}}$, $\zeta$ and $\omega_n$. As shown in the figure, the controlled systems with the PID and "lead+integration" controllers are stable under the solenoid model parameter variations considered. The solenoid dynamics may vary more with operating conditions. However, due to the lack of information on solenoid structure and design parameters, the examination of robustness measure in this section is limited to the test data available.

Figure 5.66 shows the variation of the static gain $K_{b_{\text{pilot}}}$ obtained at various operating points based on the experimental data shown in Figure 5.27. $K_{b_{\text{pilot}}}$ changes between 2.118 ~ 2.128. This variation is combined with the solenoid model parameter variation to obtain model uncertainty under different operating conditions, while the other testing conditions remain unchanged. Figure 5.67 shows the stability robustness measure and the two worst cases of model uncertainty labeled as $l_{o_{1}}$ and $l_{o_{2}}$. As shown in the figure, the stability robustness measure is greater than the model uncertainty. The controlled systems with PID and "lead+integration" controllers are stable.

The above analysis indicates that the controlled systems with the PID and "lead+integration" controllers are stable. The examination of the model uncertainty is carried out by comparing the nominal model with the plant model for specified changes in temperature, operating point and solenoid model parameters. The analysis is limited by the available information on the solenoid design and testing conditions. Experimental results are obtained at several conditions, which do not cover the full operating range encountered in automobile operation. The designed controllers have to be tested under a wider range of operating conditions to fully determine their robustness.
Figure 5.65 Stability robustness measure of PID and “Lead+Integrator” controllers with solenoid model parameter change

Figure 5.66 Variation of $K_{spilo}$ at different operating points
Figure 5.67 Stability robustness measure of PID and "Lead+Integrator" controllers with solenoid model parameter change and $K_{spilot}$ change

The nature of the controller design problem here is different from that of more conventional high performance hydraulic control problems. In the latter case, high performance two-stage servovalves are used. It is then generally the case that the load dynamics of the system are more significant than the first stage valve dynamics. In addition, the load resonance is usually lightly damped, resulting in the control involving control of a lightly damped resonance. Furthermore, pressure or force control in the latter case is more difficult than position control because of the presence of plant zeros, which are lightly damped and determined by load characteristics. In the current application, the system is moderately damped because of the high leakages. Also, load resonance is not significant, but load dynamics are highly variable because of the variability in the way the clutch chamber fills from one application to another. If the forward clutch never empties, it is appropriate to say that for a given unit, the load dynamics vary little. If this were not the case, the load dynamics may be different depending on the conditions existing at the time the clutch was filled. So, it would be necessary to develop a better
understanding of how to determine the actual load dynamic behavior in service, and then incorporate robustness issues in controller design.

The pilot stage dynamics are significant for the frequency range of interest. This is a consequence of the fact that the need for good dynamic response of the pilot stage has not been appreciated. Better dynamic performance need not necessarily mean a big increase in cost, because of the economies of scale possible with the application.

5.10 Conclusion

This chapter describes the detailed analysis of the dynamic response of a neutral-idle pressure control system. A nonlinear simulation model is presented and verified by experiments. The simulation results correlate well with experimental data. Analysis of the system reveals that the pilot stage dynamics are dominant in the neutral-idle pressure control system, for the equipment studied. The power-stage spool valve responds much faster. The analytical model can be used to improve system design as it expresses clearly the dependence of the system dynamic response on the physical system parameters.

The desired pressure trajectory is a terminated ramp. Open loop control will work reasonably well in steady state conditions provided that an accurate model is available or, equivalently, extensive calibration is performed. If the desired pressure trajectory involves slow transients compared to the plant dynamics, then this scheme may work reasonably fine as well. However, if the desired trajectory involves rapid changes as in neutral-idle control applications, then the system dynamics will be important.

Combined feed forward and feedback control scheme is preferred for more accurate tracking of a wider variety of desired pressure trajectories. Feedforward control provides a prediction of the control effort needed to track a given trajectory. Feedback control compensates for any steady-state tracking error and improves transient tracking performance as well. Controller design is carried out based on the linearized model derived. Conventional “lead+integrator”, and PID controllers are shown to provide satisfactory tracking performance. After satisfactory pressure control is achieved, the pressure control system is incorporated into the neutral-idle control system, which is discussed in Chapter 6.
6.1 Introduction

Torque and inertia phase control of a stepped automatic transmission is discussed in Chapter 4. In this chapter, integrated engine-transmission control of neutral idle shifts is considered. Recalling the review of neutral-idle control given in Chapter 2, the objective in neutral idle control is to put the automatic transmission in the neutral state by fully or partially disengaging the input clutch when the vehicle is stalled with the brake on and the manual shift valve is in “D”. This would occur quite frequently in urban driving. With neutral idle control, fuel economy during urban driving can be improved due to the resulting decrease in torque converter slip. In addition, with the input clutch fully or partially disengaged, the engine vibrations transmitted to the passenger compartment would be reduced.

Neutral idle control process can be divided into four phases:
(1) Forward clutch partially disengaged
(2) Neutral state
(3) Forward clutch application phase, and
(4) Forward clutch lockup.

The first phase is the forward clutch disengaging phase, the goal of which is to partially disengage the forward clutch so as to maintain the clutch pressure at a desired
value. This is done to ensure that the clutch piston is fully stroked, enabling it to respond quickly without the delay associated with clutch filling when the clutch application command occurs. The tradeoff in disengaging of the forward clutch is increased frequency of the forward clutch usage, which increases clutch friction and wear. This requires reinforced forward clutch structure and accurate pressure control (Hayabuchi et al., 1996).

In the second phase, the engine is idling and the forward clutch is partially disengaged. The control problem in this phase is to maintain the idle speed constant in the presence of changes in the accessory loads which would result from the air conditioning being turned on and off. The manipulated variable in this case is the airflow rate, usually involving the idle bypass valve operation. In the context of neutral idle shifts, the torque converter slip also needs to be maintained close to a desired low value, typically 100 to 200 rpm. Engine idle speed control has been studied extensively in the literature, though the torque load on the torque converter pump has not received much attention. Maintaining the torque converter slip speed at the desired level, on the other hand, can be achieved by controlling the forward clutch pressure. There is some interaction between engine and transmission control functions in this phase. However, the control problems in this phase involve regulation of the controlled variables, a task which is easier than tracking changing reference inputs. Coordinated tracking of time-varying engine and transmission variables is more difficult and is required in the third phase, which is the forward clutch application phase.

In the third phase, accurate coordination of engine and transmission functions is critical. If the forward clutch engages too slowly, the engine speed may race. On the other hand, if the forward clutch engages too fast, the engine speed may drop too abruptly, resulting in the engine stalling. Therefore, among the four phases in the neutral idle shift described above, the most critical phase is the third phase. When the driver steps on the gas pedal, he or she expects the car to accelerate immediately. To ensure satisfactory performance, the forward clutch has to engage quickly and smoothly.
The fourth phase of neutral idle shift is forward clutch lockup. The closed loop control in the third phase ends when the forward clutch slip speed decreases below a threshold value. Then open loop control is used for clutch lockup.

In the published literature, neutral idle control is done by controlling transmission functions alone to achieve tracking of the desired torque converter slip trajectory (Hayabuchi et al., 1996; Brown and Kraska, 1993). In the two references cited, the torque converter slip speed is controlled by manipulating the forward clutch pressure through a linear solenoid. Conventional PID control is used in both cases. However, there are no analytical details given by Hayabuchi et al. (1996). Brown and Kraska use PID control to correct the torque converter slip error. The static nonlinear characteristics of the linear pressure control solenoid are inverted to obtain the feedforward component of the required solenoid current. Details of their approach are given in Chapter 2.

Neither of the above approaches mentions the effect of engine dynamics. In fact, the coordination of engine and transmission functions is the key to a successful forward clutch application process. This is because different drivers may request different acceleration trajectories by varying their operation of the gas pedal. With manual throttle control, the burden of engine control falls on the driver. With electronic throttle control, incorporating explicit control of engine functions into neutral idle control design allows one to capture the driver's intent by specifying an appropriate desired engine speed trajectory. Details of this technique will be given in the following sections.

A new coordinated engine-transmission control approach for the neutral idle forward clutch application phase is proposed in this chapter. In this phase, the clutch pressure is directly manipulated by a solenoid controlled spool valve. The engine function to be controlled is the throttle angle command to the electronic throttle control loop. The transmission function to be controlled is the voltage input to the neutral-idle pressure control solenoid. The objective here is to design a model based neutral idle control strategy. The Linear Quadratic Regulation (LQR) formulation is used to design the controller. This design is presented in the following sections.
6.2 System dynamic model

The whole system is separated into three subsystems, viz. the powertrain subsystem, the electronic throttle control (ETC) subsystem, and the clutch pressure control subsystem. The model of each subsystem is described below.

6.2.1 Powertrain dynamic model

6.2.1.1 Engine dynamic model

The following assumptions are made while deriving the engine model:

a. Accessory load: \( T_a = 0 \);

b. Throttle angle (driver input): \( \alpha \leq 79.46^\circ \);

c. Air-fuel ratio remains stoichiometric, and

d. Spark-to-torque production delay \( \Delta t_s = 0 \).

Air-fuel ratio control is a complex issue in its own right. It has been extensively studied in the literature. The most common approach used for the transient air-fuel ratio control is open-loop control. According to Hendricks and Sorenson (1991), air-fuel ratio transients resulting from the fast application of throttle last about 30-40 milliseconds with open loop control. Closed-loop air-fuel ratio control strategies reported (Cho and Hedrick, 1988; Onder and Geering, 1993; Turin and Geering, 1994; Kim and Rizzoni, 1998) show time constant and delay of the order of 1 to 2 seconds. Hendricks et al. (1993) also show that fast and accurate air-fuel ratio control can be achieved by the use of an open-loop observer. In the neutral-idle control process, we are interested in the shift durations of less than 0.5 second and typically a few tenths of a second. Therefore, fast open-loop control of air-fuel ratio is assumed, and the air-fuel ratio is assumed to be at stoichiometric value in the formulation of the neutral-idle control problem presented in this chapter. The tradeoff is that air-fuel ratio may not be controlled as precisely, but the air-fuel ratio control does not need to be integrated into the neutral idle control formulation. If the air-fuel ratio were controlled closed loop, it would have been
controlled more precisely but its control has to be integrated with neutral idle control due to the time constant of air-fuel ratio control loop.

Based on the engine model derived in Chapter 3 and the above assumptions, the engine model equations are given below. The symbols used here are the same as those used in Chapter 3 if not accompanied by their definitions.

\[ m_{ai} = \text{MAX} \cdot [1 - \cos(1.14459\alpha - 1.06)] \cdot \left[ 1 - e^{-9(\theta_{\text{atm}} - 1)} \right] \]  

(6.1)

\[ \frac{dP_m}{dt} + \frac{\eta_{\text{vol}} \cdot V_t}{4\pi \cdot V_m} \cdot \omega_e \cdot P_m = m_{ai} \cdot \frac{R \cdot T_m}{V_m} \]  

(6.2)

\[ \eta_{\text{vol}} = (24.5\omega_e - 3.10 \times 10^4) \left( \frac{P_m V_m}{R \cdot T_m} \right)^2 + (-0.167\omega_e + 222) \left( \frac{P_m V_m}{R \cdot T_m} \right) + (8.10 \times 10^{-4}\omega_e + 0.352) \]  

(6.3)

Define

\[ P_{mn} = \frac{P_m}{P_{\text{atm}}} \]  

(6.4)

where \( P_{\text{atm}} = 101330\text{ Pa} \).

\[ k_1 = \frac{\eta_{\text{vol}} \cdot V_t}{4\pi \cdot V_m} \]  

(6.5)

\[ k_2 = \frac{R \cdot T_m}{V_m} \]  

(6.6)

\[ k_3 = \frac{k_2}{P_{\text{atm}}} \]  

(6.7)

Rewriting equation (6.2) in terms of \( P_{mn} \):

\[ \frac{dP_{mn}}{dt} + k_1 \cdot \omega_e \cdot P_{mn} = k_3 \cdot m_{ai} \]  

(6.8)

The engine indicated torque is given by

\[ T_i = c_r \cdot c_1 \cdot \eta_{\text{vol}} \cdot \frac{V_m}{R \cdot T_m} \cdot P_m (t - \Delta t) \cdot SI \]  

(6.9)

where

320
where \( \Delta t_d \) is the intake to torque production delay, \( = \frac{5.48}{\omega_e} \) seconds. While the engine is idling, the delay \( \Delta t_e \) is maximum. Letting \( \omega_e(0) = 800 \text{rpm} \), and using the first order Padé approximation for the delay, we get
\[
e^{-at} \approx \frac{1 - 0.5t_d s}{1 + 0.5t_d s}
\]
where \( s \) is Laplace operator, \( t_d = 0.0654 \text{second at 800 rpm} \). Then equation (6.9) becomes,
\[
T_i = k_4 \cdot SI \cdot P_{mn}(s) \cdot \frac{1 - 0.5t_d s}{1 + 0.5t_d s}
\]
(6.13)

\[
I_e \cdot \omega_e = T_i - T_f - T_p
\]
(6.14)

\[
T_f = B_e \cdot \omega_e + 15.10
\]
(6.15)

where \( B_e = 0.1056 \text{ N-rad/sec} \)

\[
k_4 = c_T \cdot c_i \cdot \eta_{vol} \cdot \frac{V_m}{R \cdot T_m} - P_{am}
\]
(6.16)

Substituting equation (6.15) into equation (6.14) and rearranging, we get
\[
I_e \cdot \omega_e + B_e \cdot \omega_e = T_i - 15.1 - T_p(\omega_e, \omega_f)
\]
(6.17)

Comparing with the engine model in Chapter 3, the main difference is in the indicated engine torque given in equation (6.13). First, fueling dynamics are ignored in the problem formulation due to the assumption of air-fuel ratio being at stoichiometric value. Second, the intake-to-torque production delay is assumed to be constant based on the engine idle speed. The engine dynamic model is simplified as a consequence. However, the drawback is a less accurate representation of engine torque generation. In addition, these simplifications place an extra burden on controller robustness requirements by introducing model uncertainty into the system. Though the engine is
simplified in the problem formulation, the designed controller is evaluated using the complete nonlinear simulation. The effect of air-fuel ratio is examined in the simulation, and will be presented later.

6.2.1.2 Torque converter dynamic model

The pump and turbine torques for the torque converter are repeated below:

\[ T_p(\omega_p, \omega_t) = a_0 \omega_p^2 + a_1 \omega_p \omega_t + a_2 \omega_t^2 \]  
\[ T_t(\omega_p, \omega_t) = b_0 \omega_p^2 + b_1 \omega_p \omega_t + b_2 \omega_t^2 \]

The torque converter model used here is a nonlinear static model originally developed by Kotwicki (1982). Pump and turbine inertial effects are accounted for appropriately in the engine and turbine equations of motion. Dynamic torque converter models with fluid inertial effects are reviewed in Chapter 2. There are two components of fluid inertial effects: fluid inertial effects in the plane of the torque converter (annular direction) and fluid inertial effects associated with the circulatory flow or torus flow. Fujita and Inukai (1990) have studied the transient characteristics of torque converters. They evaluate three types of torque converter models. The first one includes the fluid inertial effects in both the annular direction and the circulatory flow direction. The second one includes fluid inertial effects in the circulatory flow direction alone. The third one does not include fluid inertial effects but does include the turbine and pump inertial effects. According to their study, the first two types of models give similar results indicating that the fluid inertial effects in the annular direction can be ignored. In addition, when the torque converter speed ratio is above 0.1, the model ignoring fluid effects gives good results compared to the complete dynamic model (the first type). When the speed ratio falls below 0.1, the model ignoring fluid inertial effects deviates from the complete dynamic model. This result is reasonable since at low speed ratios, the fluid deceleration is greater and fluid inertial effects are likely to be more significant.

In the clutch application phase of neutral-idle control, the torque converter speed ratio is close to 1 initially, and decreases when the forward clutch is applied. The torque
converter speed ratio of 0.1 is encountered when the forward clutch is close to lockup and when the vehicle starts to accelerate after reaching the first gear. The closed-loop control strategy proposed in this chapter is ended before lockup occurs. Therefore, a torque converter model which ignores fluid inertial effects should be sufficient for our controller design needs.

Define the torque ratio of the torque converter and the \( K \) – factor as

\[
T_i \left( \frac{\omega_i}{\omega_e} \right) \\
T_p \left( \frac{\omega_i}{\omega_e} \right)
\]

\[ r_T = \frac{T_i \left( \frac{\omega_i}{\omega_e} \right) \cdot \frac{T_i \left( \frac{\omega_i}{\omega_e} \right)}{T_p \left( \frac{\omega_i}{\omega_e} \right)} }{ T_p \left( \frac{\omega_i}{\omega_e} \right) } \]  \hspace{1cm} (6.20)

\[
K = \frac{\omega_e}{\sqrt{T_p}} \hspace{1cm} (6.21)
\]

Figure 6.1 and Figure 6.2 show typical performance data for a torque converter. From these figures, we can curve-fit the two performance characteristics as follows:

\[
r_T = \begin{cases} 
-1.2352 \left( \frac{\omega_i}{\omega_e} \right)^3 + 1.4710 \left( \frac{\omega_i}{\omega_e} \right)^2 - 1.0978 \left( \frac{\omega_i}{\omega_e} \right) + 1.6854 & \text{if } \left( \frac{\omega_i}{\omega_e} \right) \leq 0.8832 \\
1 & \text{if } \left( \frac{\omega_i}{\omega_e} \right) > 0.8832 
\end{cases} \]  \hspace{1cm} (6.22)

\[
K = 18.0515 \left( \frac{\omega_i}{\omega_e} \right)^2 - 9.8150 \left( \frac{\omega_i}{\omega_e} \right) + 17.4827 \hspace{1cm} \text{if } \left( \frac{\omega_i}{\omega_e} \right) \leq 0.8832 \]  \hspace{1cm} (6.23)

During the clutch application phase, the speed ratio will be less than 0.8832, i.e. the torque converter will be operating in the torque multiplication mode.

6.2.1.3 Gear box and vehicle dynamics

The dynamic equations of the transmission mechanical system are repeated below.
Figure 6.1 Torque ratio versus speed ratio for the torque converter

Figure 6.2 K-factor ($rad/sec/\sqrt{Nm}$) versus speed ratio for the torque converter
\[ I_t \cdot \omega_t = T_t - T_{C_1} \tag{6.24} \]
\[ I_{eq} \cdot \omega_{st} = T_{C_1} - d_{21} \cdot R_D \cdot T_s \tag{6.25} \]
\[ I_{eq} = I_{st} + I_{r_r} \cdot d_{11}^2 + I_{c_r} \cdot d_{21}^2 \tag{6.26} \]
\[ T_{C_1} = P_{C_1} \cdot k_C \cdot \mu(\omega_t - \omega_{st}) \tag{6.27} \]

where
\[ k_C = A_{C_1} \cdot R_{C_1} \cdot n_{C_1} \tag{6.28} \]
\[ \mu(\omega_t - \omega_{st}) = C_{w_0} + C_1 \cdot e^{-C_{w_1} |\omega_t - \omega_{st}|} \tag{6.29} \]

\[ \omega_t > \omega_{st} \] during this phase, therefore, \[ |\omega_t - \omega_{st}| = \omega_t - \omega_{st}. \]
\[ I_v \cdot \omega_v = T_s - T_L \tag{6.30} \]
\[ \dot{\omega}_v = d_{21} \cdot R_D \cdot \omega_{st} \tag{6.31} \]
\[ T_L = C_r \cdot M_v \cdot g + B_{\omega} \cdot \omega_{st}^2 \tag{6.32} \]

The gear box model given here is for the neutral-to-first gear phase only for the formulation of neutral idle control. The models for other phases of shifts are given in Chapter 3. The vehicle dynamics model here is simplified by assuming a stiff output shaft. Therefore, the effect of forward clutch application on vehicle vibration caused by drivetrain compliance is not accurately captured. This is a limitation when evaluating drivers' perception of neutral idle shift quality. It also places an additional burden on the robustness requirements on the controller by introducing extra model uncertainty into the system.
6.2.1.4 State space model

Let

\[
\begin{aligned}
    x_1 &= P_{mn} \\
    x_2 &= T_f \\
    x_3 &= \omega_e \\
    x_4 &= \omega_t \\
    x_5 &= \omega_{si}
\end{aligned}
\]  

(6.33)

Denote

\[
\alpha_1 = 0.1843 \left[ 1 - \cos \left( \frac{(1.14459\alpha - 1.06)\pi}{180} \right) \right] = \text{MAX} \cdot \varphi(\alpha)
\]  

(6.34)

\[
I_v = M_v \cdot r^2
\]  

(6.35)

\[
I_{eq1} = I_{eq} + d_{21} R_s^2 I_v
\]  

(6.36)

Then the model equations can be written in the state-space form as:

\[
\begin{aligned}
    \dot{x}_1 &= -k_1 \cdot x_1 \cdot x_3 + k_3 \cdot \alpha_1 \cdot \left( 1 - e^{9(x_1-1)} \right) \\
    \dot{x}_2 &= -\frac{1}{0.5 t_d} x_2 + \frac{k_4 \cdot S I}{0.5 t_d} x_1 + k_1 \cdot k_4 \cdot S I \cdot x_1 x_3 - k_3 \cdot k_4 \cdot S I \cdot \alpha_1 \cdot \left( 1 - e^{9(x_1-1)} \right) \\
    \dot{x}_3 &= -\frac{B_e}{I_e} x_3 + \frac{1}{I_e} x_2 - \frac{15.1}{I_e} \frac{T_p(x_3, x_4)}{I_e} \\
    \dot{x}_4 &= \frac{T_p(x_3, x_4)}{I_t} \cdot \frac{k_c}{I_t} \cdot \mu(\omega_t - \omega_{si}) \cdot P_{cl} \\
    \dot{x}_5 &= \frac{k_c}{I_{eq1}} \cdot \mu(\omega_t - \omega_{si}) \cdot P_{cl} - \frac{d_{21} R_{D} C_{r} M_{s} g}{I_{eq1}} - \frac{d_{21} R_{D} B_{sc}}{I_{eq1}} x_5^2
\end{aligned}
\]  

(6.37) \quad (6.38) \quad (6.39) \quad (6.40) \quad (6.41)

Define

\[
\begin{aligned}
    k_5 &= \frac{k_c}{I_{eq1}} = \frac{A_{cl} R_{cl} n_{cl}}{I_{eq1}} \\
    k_6 &= \frac{d_{21} R_{D} C_{r} M_{s} g}{I_{eq1}}
\end{aligned}
\]  

(6.42) \quad (6.43)
Then the state space equations can be rewritten as

\[
\begin{align*}
  x_1 &= -k_1 \cdot x_1 \cdot x_3 + k_3 \cdot \alpha_1 \cdot (1 - e^{\alpha_i \cdot (t - t_{ci})}) \\
  x_2 &= -k_8 x_2 + k_4 k_8 \cdot SI \cdot x_1 + k_1 \cdot k_4 \cdot SI \cdot x_1 x_3 - k_3 \cdot k_4 \cdot SI \cdot \alpha_1 \cdot (1 - e^{\alpha_i \cdot (t - t_{ci})}) \\
  x_3 &= -k_9 x_3 + k_{10} x_2 - 15.1 k_{10} - k_{10} \cdot T_p(x_3, x_4) \\
  x_4 &= k_{11} \cdot T_i(x_3, x_4) - k_{12} \cdot \mu(x_4 - x_3) \cdot P_{CI} \\
  x_5 &= -k_6 - k_7 x_5^2 + k_5 \cdot \mu(x_4 - x_5) \cdot P_{CI}
\end{align*}
\] (6.50)

6.2.2 Linearized powertrain dynamic model

Let us consider the system given by equation (6.50) first, and add the ETC and pressure control valve dynamics later. We can write the system equations in a compact form as follows,

\[
\begin{align*}
  \dot{x} &= f(x) + g_1(x, SI, \alpha_i) + g_2(x) P_{CI} \\
  y &= h(x)
\end{align*}
\] (6.51)

where
The system has three inputs \( \alpha_i \) (function of \( \alpha \)), SI and \( P_{cl} \), and two outputs. \( \alpha_i \) and \( P_{cl} \) are chosen as control inputs, and SI is set to 1. The control objective is to let the engine and turbine speeds track specified trajectories. Linearizing the system equations, we get,

\[
\Delta x = \left[ \left( \frac{\partial f(x)}{\partial x} \right)_0 + \left( \frac{\partial g_1(x, SI, \alpha_i)}{\partial x} \right)_0 + \left( \frac{\partial g_2(x)}{\partial x} \right)_0 (P_{cl})_0 \right] \Delta x \\
+ \left( \frac{\partial (\bar{g}_{i}(x, \alpha_i))}{\partial (\alpha_i)} \right)_0 \delta(\alpha_i) + g_2(x_0) \delta P_{cl}
\]

(6.56)

\[
\delta y = \begin{bmatrix} \delta x_3 \\ \delta x_4 \end{bmatrix}
\]

(6.57)
where \( \frac{\partial f(x)}{\partial x} \), \( \frac{\partial g_1(x, SI, \alpha_1)}{\partial x} \), \( \frac{\partial g_2(x)}{\partial x} \), \( \frac{\partial \tilde{g}_1(x, SI, \alpha_1)}{\partial (\alpha_1)} \), \( \frac{\partial \tilde{g}_2(x, SI, \alpha_1)}{\partial (\alpha_1)} \) are Jacobian matrices evaluated at the operating point. Denoting:

\[
F_i = \left[ \left( \frac{\partial f(x)}{\partial x} \right) + \left( \frac{\partial g_1(x, SI, \alpha_1)}{\partial x} \right) + \left( \frac{\partial g_2(x)}{\partial x} \right) (P_{ci})_0 \right] \tag{6.58}
\]

\[
G_i = \left[ \left( \frac{\partial (\tilde{g}_1(x, SI, \alpha_1))}{\partial (\alpha_1)} \right) \right] g_2(x_0) \tag{6.59}
\]

\[
H_i = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \end{bmatrix} \tag{6.60}
\]

\[
J_i = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix} \tag{6.61}
\]

\[
\delta u = \begin{bmatrix} \delta (\alpha_1) \\ \delta (P_{ci}) \end{bmatrix} \tag{6.62}
\]

where the subscript \( i \) indicates the system matrices obtained at the \( i^{th} \) operating point. The system model becomes,

\[
\begin{align*}
\dot{\delta x} &= F_i \delta x + G_i \delta u \\
\delta y &= H_i \delta x + J_i \delta u
\end{align*} \tag{6.63}
\]

For convenience, we will drop \( \delta \) and note that we are working with the linearized perturbation model. So we have

\[
\begin{align*}
\dot{x} &= F_i x + G_i u \\
y &= H_i x + J_i u
\end{align*} \tag{6.64}
\]

For compactness in representation, we will also use \((F_i, G_i, H_i, J_i)\) to represent the above system.

Next we need to linearize the system model around several operating points to examine variations in the behavior of the linearized system. For the first operating point, assume that the engine is idling, and that the engine idle speed is 800 rpm. The forward clutch is partially disengaged, and the clutch piston is fully stroked. It is also assumed
that the speed ratio across the torque converter is 0.85. Figure 6.1 shows that the operating point is at the starting portion of the torque multiplication phase. In practice, the slip speed across the torque converter is less than 200 rpm. Since the vehicle barely moves during the neutral-to-first gear engaging phase, we will assume \( \omega_s = 0 \), i.e. \( \omega_{si} = 0 \). Based on these assumptions, we can calculate the following conditions corresponding to the operating point.

\[
\begin{align*}
\omega_{e0} &= 800 \text{rpm} = \frac{800 \times 2\pi}{60} \text{ rad/ sec} \\
r_{sp0} &= 0.85 \\
\omega_{i0} &= r_{sp0}\omega_{e0} = 680 \text{rpm} \\
r_{r0} &= -1.2352r_{sp0}^3 + 1.4710r_{sp0}^2 - 1.0978r_{sp0} + 1.6854 \\
K_0 &= 18.0515r_{r0}^2 - 9.8150r_{r0} + 17.4827 \\
T_{p0} &= \left(\frac{\omega_{e0}}{K_0}\right)^2 \\
T_{i0} &= T_{p0}r_{r0} \\
\omega_{si0} &= 0 \\
\omega_{v0} &= 0 \\
P_{C10} &= \frac{T_{i0}}{k_c \left(C_{m0} + C_{m1}e^{-C_{m2}(\omega_{e0} - \omega_{i0})}\right)} Pa \\
\end{align*}
\]

The minimum brake torque holding the vehicle is:

\[
T_{\text{brake}} = k_5 \left(C_{m0} + C_{m1}e^{-C_{m2}(\omega_{e0} - \omega_{i0})}\right)P_{C10} - k_6 \\
\]

The forward clutch torque capacity is

\[
T_{C10} = k_c \left(C_{m0} + C_{m1}e^{-C_{m2}(\omega_{e0} - \omega_{i0})}\right)P_{C10} \\
\]

The engine states at the operating conditions are:

\[
egin{align*}
T_{f0} &= B_e \omega_{e0} + 15.1 \quad (Nm) \\
T_{i0} &= T_{f0} + T_{p0}
\end{align*}
\]
\( P_{mn0} \) is obtained from equation (6.3), and then

\[
\alpha_0 = \frac{1}{1.14459} \left( 1.06 + \cos^{-1} \left( 1 - \frac{k_1 \omega_{e0} P_{mn0}}{0.1843 k_3 (1 - e^{9(p_{mn0})})} \right) \right) \frac{180}{\pi} \text{ deg (6.78)}
\]

Table 6.1 lists the nominal conditions at several operating points, including the one described above.

The linearized matrices at the first operating point are given by equations (6.79)-(6.82). Table 6.2 - Table 6.8 list the pole/zero locations of the system equations obtained at the seven operating points.

![Table 6.1 List of operating points chosen for deriving the linearized models](image)

Table 6.1 List of operating points chosen for deriving the linearized models

331
\[
F_1 = \begin{bmatrix}
-6.6855 & 0 & -.02049 & 0 & 0 \\
4.599 \times 10^3 & -30.5751 & 1.7461 & 0 & 0 \\
0 & 6.8776 & -4.9413 & 2.2037 & 0 \\
0 & 0 & 14.7951 & -9.2529 & -.6260 \\
0 & 0 & 0 & -.01424 & .01424 \\
\end{bmatrix}
\]

\[
G_1 = \begin{bmatrix}
306.5384 & 0 \\
-3.0539e4 & 0 \\
0 & 0 \\
0 & -3.517 \times 10^{-3} \\
0 & -8.0 \times 10^{-5} \\
\end{bmatrix}
\]

\[
H_1 = \begin{bmatrix}
0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 \\
\end{bmatrix}
\]

\[
J_1 = \begin{bmatrix}
0 & 0 \\
0 & 0 \\
\end{bmatrix}
\]

\[
k_p \frac{(s-z_1)(s-z_2)\cdots(s-z_m)}{(s-p_1)(s-p_2)\cdots(s-p_n)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>(\frac{y_1(s)}{u_1})</th>
<th>(\frac{y_1(s)}{u_2})</th>
<th>(\frac{y_2(s)}{u_1})</th>
<th>(\frac{y_2(s)}{u_2})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poles: (p_i)</td>
<td>-32.0389</td>
<td>-32.0389</td>
<td>-32.0389</td>
<td>-32.0389</td>
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<td>-3.9375+2.4420i</td>
<td>-3.9375+2.4420i</td>
<td>-3.9375+2.4420i</td>
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<td>-3.9375-2.4420i</td>
</tr>
<tr>
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<td>0.0160</td>
<td>0.0160</td>
<td>0.0160</td>
</tr>
<tr>
<td>Zeros: (z_j)</td>
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<td>39.4809</td>
<td>-0.0000</td>
</tr>
<tr>
<td></td>
<td>-9.2539</td>
<td>-6.6855</td>
<td>0.0142</td>
<td>-5.1170+4.8139i</td>
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<tr>
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<td>39.4809</td>
<td>-0.0000</td>
<td>-5.1170-4.8139i</td>
<td>-31.9679</td>
</tr>
<tr>
<td>Gain: (k_p)</td>
<td>-2.1 \times 10^4</td>
<td>-7.749 \times 10^{-3}</td>
<td>-3.11 \times 10^3</td>
<td>-3.517 \times 10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.2 Poles and zeros of system \((F_1, G_1, H_1, J_1)\)
\[ \frac{k_p \frac{(s-z_1)(s-z_2)\cdots(s-z_m)}{(s-p_1)(s-p_2)\cdots(s-p_n)}}{\frac{Y_1(s)}{u_1}} \]

Table 6.3 Poles and zeros of system \((F_2, G_2, H_2, J_2)\)

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>(\frac{Y_1(s)}{u_1})</th>
<th>(\frac{Y_1(s)}{u_2})</th>
<th>(\frac{Y_2(s)}{u_1})</th>
<th>(\frac{Y_2(s)}{u_2})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poles: (p_i)</td>
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<td>-32.2117</td>
<td>-32.2117</td>
<td>-32.2117</td>
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<tr>
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<td>-12.3409</td>
<td>-12.3409</td>
<td>-12.3409</td>
<td>-12.3409</td>
</tr>
<tr>
<td></td>
<td>-3.9269+2.4675i</td>
<td>-3.9269+2.4675i</td>
<td>-3.9269+2.4675i</td>
<td>-3.9269+2.4675i</td>
</tr>
<tr>
<td></td>
<td>0.0295</td>
<td>0.0295</td>
<td>0.0295</td>
<td>0.0295</td>
</tr>
<tr>
<td>Zeros: (z_j)</td>
<td>39.6017</td>
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<tr>
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<td>0.0270</td>
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<td>0.0000</td>
<td>0.0000</td>
</tr>
<tr>
<td>Gain: (k_p)</td>
<td>-2.134x10^3</td>
<td>-8.582x10^-3</td>
<td>-3.527x10^6</td>
<td>-3.596x10^-3</td>
</tr>
</tbody>
</table>

\[ \frac{k_p \frac{(s-z_1)(s-z_2)\cdots(s-z_m)}{(s-p_1)(s-p_2)\cdots(s-p_n)}}{\frac{Y_1(s)}{u_1}} \]

Table 6.4 Poles and zeros of system \((F_3, G_3, H_3, J_3)\)

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>(\frac{Y_1(s)}{u_1})</th>
<th>(\frac{Y_1(s)}{u_2})</th>
<th>(\frac{Y_2(s)}{u_1})</th>
<th>(\frac{Y_2(s)}{u_2})</th>
</tr>
</thead>
<tbody>
<tr>
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<td>-3.9519+2.4121i</td>
<td>-3.9519+2.4121i</td>
<td>-3.9519+2.4121i</td>
</tr>
<tr>
<td></td>
<td>0.0132</td>
<td>0.0132</td>
<td>0.0132</td>
<td>0.0132</td>
</tr>
<tr>
<td>Zeros: (z_j)</td>
<td>39.4415</td>
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<tr>
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<td>-6.6439</td>
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<td>-5.0517+4.7323i</td>
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<td>0.0127</td>
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<td>0.0000</td>
</tr>
<tr>
<td>Gain: (k_p)</td>
<td>-2.09x10^3</td>
<td>-7.44x10^-3</td>
<td>-2.97x10^4</td>
<td>-3.497x10^-3</td>
</tr>
</tbody>
</table>

333
\[
\frac{k_p \left( \frac{s-z_1}{s-p_1} \right) \left( \frac{s-z_2}{s-p_2} \right) \cdots \left( \frac{s-z_m}{s-p_n} \right)}{y_1(s)} = \frac{y_2(s)}{u_2(s)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>( \frac{y_1}{u_1} )</th>
<th>( \frac{y_1}{u_2} )</th>
<th>( \frac{y_2}{u_1} )</th>
<th>( \frac{y_2}{u_2} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poles: ( p_i )</td>
<td>-32.4120</td>
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<tr>
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<td>-4.0590 +2.131i</td>
<td>-4.0590 +2.131i</td>
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<td>0.0637</td>
<td>0.0637</td>
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<tr>
<td>Zeros: ( z_j )</td>
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<td>-8.9546</td>
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</tr>
<tr>
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<td>0.0000</td>
<td>-5.4427 - 5.3926i</td>
</tr>
<tr>
<td>Gain: ( k_p )</td>
<td>-2.176x10^3</td>
<td>-8.8x10^{-3}</td>
<td>-3.914x10^6</td>
<td>-3.731x10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.5 Poles and zeros of system \((F_4, G_4, H_4, J_4)\)

\[
\frac{k_p \left( \frac{s-z_1}{s-p_1} \right) \left( \frac{s-z_2}{s-p_2} \right) \cdots \left( \frac{s-z_m}{s-p_n} \right)}{y_1(s)} = \frac{y_2(s)}{u_2(s)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>( \frac{y_1}{u_1} )</th>
<th>( \frac{y_1}{u_2} )</th>
<th>( \frac{y_2}{u_1} )</th>
<th>( \frac{y_2}{u_2} )</th>
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</thead>
<tbody>
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<td>-32.5446</td>
<td>-32.5446</td>
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<td>-7.2979+ 2.5098i</td>
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<tr>
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<td>-5.4514 + 5.6313i</td>
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</tr>
<tr>
<td>Gain: ( k_p )</td>
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<td>-7.462x10^{-3}</td>
<td>-4.091x10^6</td>
<td>-3.923x10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.6 Poles and zeros of system \((F_5, G_5, H_5, J_5)\)

334
\[
k_p \frac{(s-z_1)(s-z_2) \cdots (s-z_m)}{(s-p_1)(s-p_2) \cdots (s-p_n)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>( \frac{y_1(s)}{u_1} )</th>
<th>( \frac{y_1(s)}{u_2} )</th>
<th>( \frac{y_2(s)}{u_1} )</th>
<th>( \frac{y_2(s)}{u_2} )</th>
</tr>
</thead>
<tbody>
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<td>-32.5859</td>
<td>-32.5859</td>
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</tr>
<tr>
<td>-6.5661+ 4.8082i</td>
<td>-6.5661- 4.8082i</td>
<td>-6.5661+ 4.8082i</td>
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<td></td>
</tr>
<tr>
<td>-6.5661- 4.8082i</td>
<td>-6.5661+ 4.8082i</td>
<td>-6.5661- 4.8082i</td>
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</tr>
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</tr>
<tr>
<td><strong>Gain: ( k_p )</strong></td>
<td>-2.23x10^3</td>
<td>-4.592x10^{-3}</td>
<td>-4.087x10^8</td>
<td>-4.131x10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.7 Poles and zeros of system \( (F_6, G_6, H_6, J_6) \)

\[
k_p \frac{(s-z_1)(s-z_2) \cdots (s-z_m)}{(s-p_1)(s-p_2) \cdots (s-p_n)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>( \frac{y_1(s)}{u_1} )</th>
<th>( \frac{y_1(s)}{u_2} )</th>
<th>( \frac{y_2(s)}{u_1} )</th>
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<tr>
<td>-10.019- 3.2129i</td>
<td>-10.019+ 3.2129i</td>
<td>-10.019- 3.2129i</td>
<td>-10.019+ 3.2129i</td>
<td></td>
</tr>
<tr>
<td>-2.5491</td>
<td>0.1855</td>
<td>-2.5491</td>
<td>0.1855</td>
<td></td>
</tr>
<tr>
<td><strong>Zeros: ( z_j )</strong></td>
<td>49.6176</td>
<td>-38.2189</td>
<td>49.6176</td>
<td>-40.3044</td>
</tr>
<tr>
<td>-8.1680</td>
<td>-9.6934</td>
<td>0.0841</td>
<td>-7.2280+ 6.5761i</td>
<td></td>
</tr>
<tr>
<td>0.1218</td>
<td>-0.0000</td>
<td>0.1218</td>
<td>-0.0000</td>
<td></td>
</tr>
<tr>
<td><strong>Gain: ( k_p )</strong></td>
<td>-2.322x10^3</td>
<td>-8.872x10^{-3}</td>
<td>-5.3723x10^8</td>
<td>-3.731x10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.8 Poles and zeros of system \( (F_7, G_7, H_7, J_7) \)
According to Table 6.2 – Table 6.8, the pole and zero locations of system 
\( (F_i, G_i, H_i, J_i) \) do change at the different operating points. The analysis on 
\( (F_i, G_i, H_i, J_i) \), i.e. the system obtained at the first operating point is carried out below.

The transfer functions given in Table 6.2 are factorized using the symbolic 
mathematics program Maple. The result is shown below.

\[
\frac{\omega_c(s)}{\alpha_1} = \frac{2.0987 \times 10^4 - 1.8756 \times 10^4 + 7.7202 + (228.25 - 2.2392s) \times 10^3}{s + 32.0389 - s + 11.5428 - s - 0.016 - s^2 + 7.8749s + 21.4668} 
\tag{6.83}
\]

\[
\frac{\omega_c(s)}{P_{Cl}} = \frac{1.763 \times 10^5 - 5.471 \times 10^4 - 3.179 \times 10^{-6} - (52.92 + 5.615s) \times 10^{-4}}{s + 32.0389 - s + 11.5428 - s - 0.016 - s^2 + 7.8749s + 21.4668} 
\tag{6.84}
\]

\[
\frac{\omega_c(s)}{\alpha_1} = \frac{-1.3627 \times 10^4 + 1.2122 \times 10^5 + 27.067 + (1.1513 - 1.0762e5s) \times 10^5}{s + 32.0389 - s + 11.5428 - s - 0.016 - s^2 + 7.8749s + 21.4668} 
\tag{6.85}
\]

\[
\frac{\omega_c(s)}{P_{Cl}} = \frac{-1.144 \times 10^5 - 3.536 \times 10^{-3} - 1.114 \times 10^{-5} - (836.2 - 4.177s) \times 10^{-5}}{s + 32.0389 - s + 11.5428 - s - 0.016 - s^2 + 7.8749s + 21.4668} 
\tag{6.86}
\]

From the above factorization, it is clear that the mode corresponding to the pole 0.016 rad/sec has little effect if we are interested in transient phenomena lasting less than 1 second. In other words, if \( s = j\omega \) and the frequency is a few Hz or higher, the term corresponding to this pole has a low magnitude compared to the other poles and can be neglected. In addition, there is a zero that is close to this pole, hence the effect of this pole is canceled. We take the approach here that, since we are interested in short duration phenomena during neutral idle shifts, we can neglect the effect of this very slow pole. Our motivation in doing so is to simplify the controller design computations. However, as we shall state later, there is a penalty paid for this simplification in the controller design.
The slow mode is related to the state $x_2$, which is associated with the vehicle speed. In this neutral-to-first gear engagement phase, the vehicle barely moves. Let us examine the reduced-order system model obtained by ignoring the state $x_5$, i.e. only considering the states $x_1, x_2, x_3, x_4$. The reduced-order system model equations are as follows:

$$
x_r = [x_1, x_2, x_3, x_4] = f_r(x) + g_{1r}(x, SI, \alpha_1) + g_{2r}(x)P_{C1}
$$

(6.87)

where

$$f_r(x) = \begin{bmatrix}
-k_1 \cdot x_1 \cdot x_3 \\
-k_8 x_2 \\
-k_9 x_3 + k_{10} x_2 - 15.1 k_{10} - k_{10} \cdot T_p(x_3, x_4) \\
k_{11} \cdot T_p(x_3, x_4)
\end{bmatrix}
$$

(6.88)

$$g_{1r}(x, SI, \alpha_1) = \begin{bmatrix}
k_3 \cdot \alpha_1 \cdot (1 - e^{g(x_1-1)}) \\
(k_4 k_9 x_1 + k_5 k_4 x_1 x_3) SI - k_7 k_4 (1 - e^{g(x_1-1)}) SI \cdot \alpha_1 \\
0 \\
0
\end{bmatrix}
$$

(6.89)

$$g_{2r}(x) = \begin{bmatrix}
0 \\
0 \\
0 \\
-k_{12} \mu (x_4 - x_{50})
\end{bmatrix}
$$

(6.90)

The linearized model can be easily obtained from the “full-order” linearized model $(F, G, H, J)$. The reduced-order linear model $(F_{1r}, G_{1r}, H_{1r}, J_{1r})$ is given by
equations (6.91) - (6.94). The pole and zero locations of the reduced-order linear model are listed in Table 6.9.

\[
F_{ir} = \begin{bmatrix}
-6.6855 & 0 & -0.0205 & 0 \\
4.5993 \times 10^3 & -30.5751 & 1.7461 & 0 \\
0 & 6.8776 & -4.9413 & 2.2037 \\
0 & 0 & 14.7951 & -9.2530 \\
\end{bmatrix}
\]  \hspace{1cm} (6.91)

\[
G_{ir} = \begin{bmatrix}
306.5384 & 0 \\
-3.0539 \times 10^4 & 0 \\
0 & 0 \\
0 & -3.517 \times 10^{-3} \\
\end{bmatrix}
\]  \hspace{1cm} (6.92)

\[
H_{ir} = \begin{bmatrix}
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1 \\
\end{bmatrix}
\]  \hspace{1cm} (6.93)

\[
J_{ir} = \begin{bmatrix}
0 & 0 \\
0 & 0 \\
\end{bmatrix}
\]  \hspace{1cm} (6.94)

\[
k_p \frac{(s - z_1)(s - z_2) \cdots (s - z_m)}{(s - p_1)(s - p_2) \cdots (s - p_n)}
\]

<table>
<thead>
<tr>
<th>Transfer function</th>
<th>( \frac{y_1(s)}{u_1} )</th>
<th>( \frac{y_1(s)}{u_2} )</th>
<th>( \frac{y_2(s)}{u_1} )</th>
<th>( \frac{y_2(s)}{u_2} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Poles: ( p_i )</td>
<td>-32.0389</td>
<td>-32.0389</td>
<td>-32.0389</td>
<td>-32.0389</td>
</tr>
<tr>
<td></td>
<td>-3.9370 + 2.4411i</td>
<td>-3.9370 + 2.4411i</td>
<td>-3.9370 + 2.4411i</td>
<td>-3.9370 + 2.4411i</td>
</tr>
<tr>
<td></td>
<td>-9.2530</td>
<td>-6.6855</td>
<td>-5.1170 + 4.8139i</td>
<td>-5.1170 - 4.8139i</td>
</tr>
<tr>
<td>Gain: ( k_p )</td>
<td>-2.1 \times 10^4</td>
<td>-7.750 \times 10^{-3}</td>
<td>-3.1075 \times 10^{-4}</td>
<td>-3.517 \times 10^{-3}</td>
</tr>
</tbody>
</table>

Table 6.9 Poles and zeros of reduced order system \((F_{ir}, G_{ir}, H_{ir}, J_{ir})\)
Comparing the pole and zero locations given in Table 6.2 and Table 6.9, it is clear that the mode corresponding to the eigenvalue 0.016 rad/sec is contributed by the state \( x_5 \). The reduced order model represents the original system well for short duration phenomena. However, higher order derivatives of the neglected state, such as jerk, do affect perceived shift quality (Naruse et al., 1993). Thus, omitting the vehicle velocity from the state prevents us from directly incorporating this information in the index of performance to be optimized by controller design. We will evaluate shift quality as indicated by vehicle jerk, a posteriori, for proposed neutral-idle controller designs.

6.2.3 Actuator models

6.2.3.1 Electronic throttle control (ETC) actuator

The throttle angle is controlled by an electronic-throttle-control (ETC) system. The dynamics of the ETC is assumed to be a first order system with a time constant of \( \tau_{ETC} \) as shown below:

\[
\alpha = \frac{1}{\tau_{ETC} s + 1} \alpha_{com}
\]

(6.95)

where \( \alpha_{com} \) is the throttle angle command, 
\( \alpha \) is the throttle angle, and 
\( \tau_{ETC} \) is the time constant.

Defining an additional state associated with the throttle

\[
x_a = \alpha
\]

(6.96)

the state space representation of (6.95) becomes,

\[
x_a = -\frac{1}{\tau_{ETC}} x_a + \frac{1}{\tau_{ETC}} \alpha_{com}
\]

(6.97)

The value of \( \tau_{ETC} \) varies from 25 – 30 ms (Lenz and Shroeder, 1997; Hendricks et al., 1996) to 100 ms (Nichols, 1997). In our application, \( \tau_{ETC} \) is chosen to be 100 ms.
based on Delphi hardware specification (Nichols, 1997). The relationship between \( x_\alpha \) and the control input \( \alpha_1 \) is given by equation (6.34), and is repeated here for convenience.

\[
\alpha_1 = 0.1843 \left[ 1 - \cos \left( (1.14459x_\alpha - 1.06) \frac{\pi}{180} \right) \right] = \text{MAX} \cdot \varphi(x_\alpha)
\]  

(6.98)

6.2.3.2 Neutral idle pressure control valve model

The neutral idle pressure control valve model derived from Chapter 5 is repeated below:

\[
\frac{P_C}{V_{in}} = \frac{-k_v \omega_{nv}^2}{s^2 + 2\zeta_v \omega_{nv} s + 1}
\]

(6.99)

where \( V_{in} \) is the solenoid input voltage,

\( k_v \) is the static gain,

\( \zeta_v \) is the damping ratio, and

\( \omega_{nv} \) is the natural frequency.

Let

\[
\begin{align*}
\dot{x}_6 &= P_{C1} \\
\dot{x}_7 &= P_{C1}
\end{align*}
\]

(6.100)

the state space representation of the pressure control valve model is as follows.

\[
\begin{bmatrix}
\dot{x}_6 \\
\dot{x}_7
\end{bmatrix} =
\begin{bmatrix}
0 & 1 \\
-\omega_{nv}^2 & -2\zeta_v \omega_{nv}
\end{bmatrix}
\begin{bmatrix}
x_6 \\
x_7
\end{bmatrix} +
\begin{bmatrix}
0 \\
-k_v \omega_{nv}^2
\end{bmatrix} V_{in}
\]

(6.101)

\[
y_v = P_{C1} = x_6
\]

(6.102)

The numerical values for neutral idle pressure control valve model are given below,

\[
k_v = 463.34 \text{ kPa/volt}
\]

(6.103)

\[
\zeta_v = 0.5
\]

(6.104)

\[
\omega_{nv} = 7.3 \text{ Hz}
\]

(6.105)
6.2.4 Complete system model with actuator dynamics

Combining the powertrain system model and the actuator models, the total system representation is given by equation (6.106).

\[
\begin{align*}
\dot{x}_\alpha &= -\frac{1}{\tau_{ETC}} x_\alpha + \frac{1}{\tau_{ETC}} \alpha_{com} \\
\dot{x}_1 &= -k_1 \cdot x_1 \cdot x_3 + k_3 \cdot M \cdot \varphi(x_\alpha) \cdot \left(1 - e^{\varphi x_\alpha}ight) \\
\dot{x}_2 &= -k_4 x_2 + S_l \left(k_4 k_8 \cdot x_1 + k_1 \cdot k_4 \cdot x_1 x_3 - k_3 \cdot k_4 \cdot M \cdot \varphi(x_\alpha) \cdot \left(1 - e^{\varphi x_\alpha}ight)\right) \\
\dot{x}_3 &= -k_9 x_3 + k_{10} x_2 - 15.1 k_{10} - k_{10} \cdot T_p (x_3, x_4) \\
\dot{x}_4 &= k_{11} \cdot T_f (x_3, x_4) - k_{12} \cdot \mu(x_4 - x_5) \cdot x_6 \\
\dot{x}_5 &= -k_6 - k_7 x_2^2 + k_3 \cdot \mu(x_4 - x_5) \cdot x_6 \\
\dot{x}_6 &= x_7 \\
\dot{x}_7 &= -\omega_{nv}^2 x_6 - 2 \zeta \omega_{nv} x_7 - k_s \omega_{nv}^2 P_{pilot}
\end{align*}
\] (6.106)

The linear model of equations (6.106) is constructed by combining equations (6.56) - (6.57) with the actuator dynamic equations. The linearized total system representation is given by equations (6.107) - (6.108).

\[
\begin{bmatrix}
\dot{x}_\alpha \\
x \\
x_6 \\
x_7
\end{bmatrix} =
\begin{bmatrix}
-\frac{1}{\tau_{ETC}} & 0 & 0 & 0 \\
0 & \frac{\partial(g_1(x, S_l, \alpha_1))}{\partial(\alpha_1)} & 0 & 0 \\
0 & 0 & \frac{\partial(g_2(x_0))}{\partial(\alpha_1)} & 0 \\
0 & 0 & 0 & \frac{1}{\tau_{ETC}}
\end{bmatrix}
\begin{bmatrix}
x_\alpha \\
x \\
x_6 \\
x_7
\end{bmatrix}
\]

\[
\begin{bmatrix}
\dot{\alpha}_{com} \\
V_{in}
\end{bmatrix}
\]

\[
y =
\begin{bmatrix}
x_3 \\
x_4
\end{bmatrix}
\] (6.108)
where \( x = [x_1 \ x_2 \ x_3 \ x_4 \ x_5]^T \). Denoting,

\[
F_{\text{sys}} = \begin{bmatrix}
-\frac{1}{\tau_{\text{ERC}}} & 0 & 0 & 0 \\
\left(\frac{\partial g_1(x, S_l, \alpha_l)}{\partial (\alpha_l)}\right)\left(\frac{\partial \alpha_l}{\partial x_{\alpha}}\right) & F_i & g_2(x_0) & 0 \\
0 & 0 & 1 \\
0 & -\omega_{\text{av}}^2 & -2\zeta_{\omega_{\text{av}}} \\
\end{bmatrix}
\]

(6.109)

\[
G_{\text{sys}} = \begin{bmatrix}
1 \\
\tau_{\text{ERC}} \\
0 \\
0 \\
0 \\
-\kappa, \omega_{\text{av}}^2 \\
\end{bmatrix}
\]

(6.110)

\[
H_{\text{sys}} = \begin{bmatrix}
0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
\end{bmatrix}
\]

(6.111)

\[
J_{\text{sys}} = \begin{bmatrix}
0 & 0 \\
0 & 0 \\
\end{bmatrix}
\]

(6.112)

\[
u_{\text{sys}} = \begin{bmatrix}
\alpha_{\text{com}} \\
V_{\text{in}} \\
\end{bmatrix}
\]

(6.113)

the system equations can be written as

\[
\dot{X} = F_{\text{sys}} X + G_{\text{sys}} u_{\text{sys}} \\
y_{\text{sys}} = H_{\text{sys}} X + J_{\text{sys}} u_{\text{sys}}
\]

(6.114)

The linearized reduced order powertrain model including the actuator dynamics is obtained similarly, and is listed below. This mode is used to design the optimal controller, which will be described in the next section.
6.3 Controller design

6.3.1 Control objective

The control objective is to coordinate engine and transmission functions to achieve a fast and smooth transition from neutral to the first gear. This involves ramping up engine speed smoothly and simultaneously engaging the forward clutch smoothly. Effective coordination can be realized by controlling the engine and turbine speeds closed loop so that they follow specified trajectories. Therefore, the controller design objective is to realize optimal trajectory tracking by synthesizing the control inputs so that the engine and turbine speeds track the specified desired trajectories closely.

The control system has two manipulated inputs and two outputs, namely, the controlled speeds. One characteristic feature of MIMO (multi-input, multi-output) control problems is the significance of process interaction whereby each manipulated variable

\[
\begin{bmatrix}
  x_1 \\
  x_2 \\
  x_3 \\
  x_4 \\
  x_5 \\
  x_6 \\
  x_7 \\
  x_r
\end{bmatrix}
= 
\begin{bmatrix}
  \frac{1}{\tau_{\text{ETC}}} & 0 & 0 & 0 \\
  \left( \frac{\partial g_{1c}(x, S_f, \alpha)}{\partial (\alpha)} \right) & 0 & 0 & 0 \\
  0 & 0 & 0 & 1 \\
  0 & 0 & 0 & 1 \\
  0 & 0 & 0 & 1 \\
  0 & 0 & 0 & 1 \\
  0 & 0 & 0 & 1 \\
  0 & 0 & 0 & 1 \\
\end{bmatrix}
\begin{bmatrix}
  x_\alpha \\
  x_{r4} \\
  x_6 \\
  x_7 \\
  x_r
\end{bmatrix}
\]

\[
+ \begin{bmatrix}
  \frac{1}{\tau_{\text{ETC}}} & 0 & 0 & 0 \\
  0 & 0 & 0 & 0 \\
  0 & 0 & 0 & 0 \\
  0 & 0 & 0 & 0 \\
\end{bmatrix}
\begin{bmatrix}
  \alpha_{\text{com}} \\
  V_{in}
\end{bmatrix}
\]

\[
y = \begin{bmatrix}
  x_3 \\
  x_4
\end{bmatrix} = \begin{bmatrix}
  0 & 0 & 0 & 1 & 0 & 0 & 0 \\
  0 & 0 & 0 & 0 & 1 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
  x_\alpha \\
  x_1 \\
  x_2 \\
  x_3 \\
  x_4 \\
  x_6 \\
  x_7
\end{bmatrix}^T
\]

where \( x_{r4} = [x_1 \ x_2 \ x_3 \ x_4]^T \) and \( x_r = [x_\alpha \ x_1 \ x_2 \ x_3 \ x_4 \ x_6 \ x_7]^T \).
(input) can affect both controlled variables (outputs). In the current problem formulation, the throttle input command affects both the engine and turbine speeds. Similarly, changing the solenoid input voltage also affects both the engine and turbine speeds. As a first step, we need to examine whether the MIMO system can be decoupled into two SISO systems with little interaction. The Relative Gain Array (RGA) method is useful in determining the level and nature of the interaction in the steady state response of multivariable systems.

6.3.2 Relative Gain Array (RGA) analysis

The RGA method developed by Bristol (Seborg et al., 1989) is used here to analyze control loop interactions. Bristol's approach is based on the concept of a relative gain. For a process with \( n \) inputs and \( n \) outputs, the relative gain \( \lambda_{ij} \) between a controlled variable \( C_i \), and a manipulated variable \( M_j \) is defined to be the dimensionless ratio of two steady-state gains:

\[
\lambda_{ij} = \frac{\left( \frac{\partial C_i}{\partial M_j} \right)_M}{\left( \frac{\partial C_i}{\partial M_j} \right)_C} = \frac{\text{open-loop gain}}{\text{closed-loop gain}}
\]

for \( i = 1, 2, \cdots, n \) and \( j = 1, 2, \cdots, n \). In equation (6.117), \( \left( \frac{\partial C_i}{\partial M_j} \right)_M \) denotes the partial derivative of the \( i^{th} \) output with respect to the \( j^{th} \) input, evaluated with all of the inputs except \( M_j \) held constant, i.e. the open-loop gain between \( C_i \) and \( M_j \). Similarly, \( \left( \frac{\partial C_i}{\partial M_j} \right)_C \) is evaluated with all the outputs except \( C_i \) held constant. \( \left( \frac{\partial C_i}{\partial M_j} \right)_C \) can therefore be interpreted as a closed-loop gain indicating the effect of \( M_j \) on \( C_i \) when all of the other feedback control loops are ideally closed.

The RGA for a linear 2 input - 2 output system is calculated as follows. For such a system, there are four process transfer functions characterizing the process dynamics:
Applying the Principle of Superposition, the outputs of the above system are given by

\[
C_1(s) = G_{p11}(s)M_1(s) + G_{p12}(s)M_2(s) \\
C_2(s) = G_{p21}(s)M_1(s) + G_{p22}(s)M_2(s)
\]  

(6.119)

The steady-state model is:

\[
C_1 = K_{11}M_1 + K_{12}M_2 \\
C_2 = K_{21}M_1 + K_{22}M_2
\]  

(6.120)

where \( K_{ij} \) is the steady-state gain relating the output \( C_i \) and the input \( M_j \).

\[
\left( \frac{\partial C_1}{\partial M_1} \right)_{M_2} = K_{11}
\]  

(6.121)

To calculate \( \left( \frac{\partial C_1}{\partial M_1} \right)_{M_2} \), let \( C_2 = 0 \), and we get from equation (6.120),

\[
M_2 = -\frac{K_{21}}{K_{22}}M_1
\]  

(6.122)

Substituting equation (6.122) into (6.120), we get

\[
C_1 = K_{11} \left( 1 - \frac{K_{12}K_{21}}{K_{11}K_{22}} \right)M_1
\]  

(6.123)

So,

\[
\left( \frac{\partial C_1}{\partial M_1} \right)_{C_2} = K_{11} \left( 1 - \frac{K_{12}K_{21}}{K_{11}K_{22}} \right)
\]  

(6.124)

and

345
We can show that (Seborg et al., 1989)
\[ \lambda_{12} = \lambda_{21} = 1 - \lambda_{11} \]
\[ \lambda_{22} = \lambda_{11} \] (6.126)

The relative gains are arranged in the following RGA:
\[ \Lambda = \begin{bmatrix} \lambda_{11} & \lambda_{12} \\ \lambda_{21} & \lambda_{22} \end{bmatrix} \] (6.127)

Applying the above calculation to the system equations (6.115) - (6.116) at the first operating point in Table 6.1, the steady-state relationship between the inputs and the outputs is:
\[ \omega_e = K_{11} \cdot \alpha_{com} + K_{12} \cdot V_{in} \]
\[ \omega_i = K_{21} \cdot \alpha_{com} + K_{22} \cdot V_{in} \] (6.128)

where
\[ K_{11} = 8.812 \text{ rad/sec} \]
\[ K_{12} = 92.4962 \text{ rad/sec volt} \] (6.129)
\[ K_{21} = 14.09 \text{ rad/sec deg} \]
\[ K_{22} = 324 \text{ rad/sec volt} \]

The RGA is calculated as:
\[ \Lambda = \begin{bmatrix} 1.8398 & -0.8398 \\ -0.8398 & 1.8398 \end{bmatrix} \] (6.130)

In equation (6.130), \( \lambda = 1.8398 > 1 \). According to the definition of relative gain values given by equation (6.117), for this situation, closing the loop around \( \omega_i \) reduces
The steady state gain between the manipulated variable $\alpha_{\text{com}}$ and the controlled variable $\omega_e$. Thus, the control loops interact significantly in the steady state. The negative value of $\lambda_{12}$ and $\lambda_{21}$ indicates that closing the loop around $\omega_e$ inverts the polarity of the steady state relationship between the manipulated variable $V_{in}$ and the controlled variable $\omega_e$. A value of $\lambda_{11}$ of unity and of $\lambda_{12}$ of zero would indicate the absence of any steady state interaction. The presence of steady state interaction by itself does not indicate that SISO controller design techniques are not useful. In fact, if the level of steady state interaction does not vary dramatically with the operating point or with frequency, static decoupling control would reduce the MIMO system to multiple SISO systems which interact weakly. In this case, SISO techniques would be appropriate for controller design.

Calculated RGA for the different operating points given in Table 6.1 are given below.

At the 2\textsuperscript{nd} point:

$$\Lambda_2 = \begin{bmatrix} 1.9645 & -0.9645 \\ -0.9645 & 1.9645 \end{bmatrix}$$

(6.131)

At the 5\textsuperscript{th} point:

$$\Lambda_5 = \begin{bmatrix} 2.5995 & -1.5995 \\ -1.5995 & 2.5995 \end{bmatrix}$$

(6.132)

At the 7\textsuperscript{th} point:

$$\Lambda_7 = \begin{bmatrix} 2.8669 & -1.8669 \\ -1.8669 & 2.8669 \end{bmatrix}$$

(6.133)

Thus, the RGA varies significantly with the operating points, and thus varies during the neutral idle engagement process. Even in such a case, reliance on single loop controllers with high disturbance rejection characteristics may be adequate. For example, incorporation of integral control might achieve this objective. However, we see in this case that the interaction varies with signal frequency as well, as shown by equations (6.83) – (6.86). Figure 6.3 shows the magnitude frequency response for the reduced order system $(F_{ir}, G_{ir}, H_{ir}, J_{ir})$ at the seven operating conditions given in Table 6.1. In the
In the frequency range of interest for neutral idle control (1 – 100 rad/sec), the magnitude frequency response curves do not have the same relationship to each other. Therefore, static decoupling control is not feasible, and a simple static decoupling controller would not be effective.

Recalling the previous work of Brown and Kraska (1993) and Hayabuchi et al. (1996), Brown and Kraska's approach relied on manipulating the solenoid input voltage in order to follow a desired torque converter slip trajectory, and Hayabuchi et al.'s approach relied on manipulating the solenoid input voltage in order to follow a desired turbine speed trajectory. In both cases, the only manipulated variable is the solenoid input.
voltage. Both approaches essentially relied on the fast response of clutch control loop to engage the forward clutch before the engine speed started to race.

It is clear, however, that variation of the solenoid input voltage results in variation of both the engine and turbine speeds. Also, variation of the throttle by the driver has a significant impact on both the engine and turbine speeds and creates a disturbance that places a greater demand for disturbance rejection by the single clutch control loop. Since the clutch control loop is faster responding than the engine speed dynamics, it probably does a reasonable job of rejecting this disturbance. However, we note that the effect of the clutch speed control loop on the engine speed is not compensated.

Based on the above analysis, an MIMO design technique is necessary to achieve the control objective of simultaneous control of engine and turbine speeds so that they follow specified trajectories.

6.3.3 Explicit Model Following (EMF) control system design

The linear quadratic optimal control formulation is selected to handle the MIMO design. Specifically, Explicit Model Following using the linear quadratic optimal control system design is employed. It is reasonable for the desired response of a closed-loop control system to follow the step response of a model, especially if the model provides the ability and flexibility to effectively specify desired performance. In the forward clutch application phase of neutral idle control, the turbine speed decreases and the engine speed increases. The simplest engine and turbine speed trajectories can be terminated ramp signals with different slopes and step sizes. However, in this case, the derivative of the terminated ramp signal is discontinuous, which is not desirable since vehicle acceleration (1st derivative of speed) and jerk (2nd derivative of speed) are related to the passengers' perception of shift quality. Similarly, a first order system step response also has discontinuity in its derivative. Since a lower model order simplifies controller design computations, we choose the model to be a second order system. The step response of a second order system provides a convenient mechanism for specifying the speed of
response, damping, and response magnitude. These attributes of the response are easily parameterized in such a formulation.

When the model characterizing the desired response is added explicitly to the original state equation, the linear quadratic optimal solution includes feedback of system states as well as dependence on the model states and the reference inputs to the models. Since model states depend entirely on the model and the reference inputs, the control law dependence on these states is equivalent to feedforward control. Below, the Explicit Model Following formulation with linear quadratic optimal control system design is described.

The desired response is written as:

\[
x_d(t) = A_d x_d(t) + B_d r(t)
\]

\[
y_d(t) = C_d x_d(t)
\]

The plant model is given below:

\[
x(t) = Ax(t) + Bu(t)
\]

\[
y(t) = Cx(t)
\]

Defining

\[
x = \begin{bmatrix} x \\ x_d \end{bmatrix}
\]

\[
A = \begin{bmatrix} A & 0 \\ 0 & A_d \end{bmatrix}
\]

\[
B = \begin{bmatrix} B \\ 0 \end{bmatrix}
\]

\[
B_r = \begin{bmatrix} 0 \\ B_d \end{bmatrix}
\]

\[
C = [C - C_d]
\]

the expanded state space equations become
\[ \dot{x}(t) = Ax(t) + Bu(t) + B_r r(t) \]  
\[ e(t) = Cx(t) \]  
\[ (6.141) \]

It is desired to have the tracking error \( e(t) \) as small as possible. This is a typical Linear Quadratic Regulator (LQR) problem. To achieve this objective, the cost function to be minimized is selected as

\[ J = 0.5 \int_0^t (e^T Q e + u^T R u) \, dt \]  
\[ (6.142) \]

with the weighting matrices \( Q \geq 0 \) and \( R > 0 \). The solution is given by (Lin, 1994):

\[ u = -R^{-1} B^T S x + K_{FF} r = Fx + K_{FF} r \]  
\[ (6.143) \]

\[ K_{FF} = R^{-1} B^T \left( A^T S B R^{-1} B^T \right)^{-1} S B_r \]  
\[ (6.144) \]

where \( S \) is the solution to the algebraic Riccati equation:

\[ S A + A^T S - S B R^{-1} B^T S + C^T Q C = 0 \]  
\[ (6.145) \]

Partitioning \( F \) as \( F = [F_1 : F_2] \), the optimal control input \( u(t) \) is expressed as

\[ u(t) = F_1 x(t) + F_2 x_d(t) + K_{FF} r(t) \]  
\[ (6.146) \]

Substituting equation (6.146) into the system equation (6.141), the closed-loop system equation is expressed as

\[ \begin{bmatrix} \dot{x} \\ \dot{x}_d \end{bmatrix} = \begin{bmatrix} A + BF_1 & BF_2 \\ 0 & A_d \end{bmatrix} \begin{bmatrix} x \\ x_d \end{bmatrix} + \begin{bmatrix} BK_{FF} \\ B_d \end{bmatrix} r(t) \]  
\[ (6.147) \]

In equations (6.146) and (6.147), \( F_1 \) is the new full-state feedback gain matrix, \( F_2 \) is the feedforward gain matrix on the desired model states, and \( K_{FF} \) is the feedforward gain matrix on the reference inputs. The feedback portion of the controller \( F_1 x(t) \) determines system stability and robustness. The feedforward portion of the controller \( F_2 x_d(t) + K_{FF} r(t) \) provides the tracking capability. The structure of the Explicit Model Following design is shown in Figure 6.4.
6.3.4 Application of EMF control to neutral idle controller design

We use numerical values for the system matrix elements in equations (6.115) and (6.116). The numerical values correspond to the operating point no. 1 in Table 6.1. The unit for clutch pressure is $10^4 Pa$. The resulting system matrices are:

$$A = F_r = \begin{bmatrix}
-10 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0.2794 & -6.6855 & 0 & -0.0205 & 0 & 0 & 0 & 0 \\
-27.832 & 4599.3 & -30.575 & 1.7461 & 0 & 0 & 0 & 0 \\
0 & 0 & 6.8776 & -4.941 & 2.2037 & 0 & 0 & 0 \\
0 & 0 & 0 & 14.795 & -9.253 & -35.167 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & -2103.8 & -45.867 & 0
\end{bmatrix} \quad (6.148)
In the control implementation, the seven states need to be available for feedback. In most production vehicles, ETC throttle angle sensor, manifold pressure sensor, engine speed and turbine speed sensors are readily available. The engine indicated torque could be obtained by engine mapping data, which are available for production vehicles. However, the clutch pressure sensing is problematic, especially for rotating clutches. If an alternative pressure signal is sensed such as the pressure control valve back chamber pressure $P_b$ discussed in Chapter 5, that raises the need for state estimation. Another state that is needed is the derivative of clutch pressure, which could be obtained by differentiating clutch pressure signal, or through state estimation. In addition, the computing power available in current production vehicles is limited, and the control computations have to be optimized and integrated with other vehicle functions. In the current formulation, we assume the clutch pressure and the derivation of clutch pressure are available for feedback. State estimations of clutch pressure and the derivative of clutch pressure remained to be explored.

Care has to be taken into account when selecting the desired response model parameters. In previous approaches to neutral idle control, the desired torque converter slip trajectory is chosen as a terminated ramp signal. Engine and turbine speeds are not individually controlled. The forward clutch pressure is manipulated to achieve the desired slip trajectory. However, in such an approach, the clutch application control does not react to engine response changes or to the nature of the driver's throttle actuation.
As discussed in section 6.3.3, a good candidate for the desired response is the step response of a second order system. The desired engine and turbine speed trajectories are, therefore, so specified. The desired response model is given below:

\[ x_d = A_d x_d + B_d r \]
\[ y_d = C_d x_d \]  

(6.151)

where

\[ x_d = \begin{bmatrix} \omega_{ed} \\ \omega_{ed} \\ \omega_{td} \\ \omega_{td} \end{bmatrix} \]  

(6.152)

\[ A_d = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\omega_{ne}^2 & -2\zeta_e \omega_{ne} & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -\omega_{nt}^2 & -2\zeta_t \omega_{nt} \end{bmatrix} \]  

(6.153)

\[ B_d = \begin{bmatrix} K_e \omega_{ne}^2 & 0 \\ 0 & 0 \\ 0 & K_t \omega_{nt}^2 \end{bmatrix} \]  

(6.154)

\[ C_d = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \]  

(6.155)

In equations (6.152) – (6.155), \( K_e, \zeta_e, \omega_{ne}, K_t, \zeta_t, \omega_{nt} \) specify the desired engine speed and turbine speed trajectories. The parameters \( K_e, \zeta_e, \omega_{ne} \) are chosen based on the nature of the driver’s manipulation of the gas pedal. For example, if the driver steps hard on the gas pedal, he/she is indicating the need for a fast neutral idle shift, and the mode parameters should be chosen accordingly.

The driver’s request in terms of the speed of response is captured by specifying the natural frequencies of the desired responses appropriately. However, the model natural frequencies for the desired engine and turbine speed trajectories can not be selected independently. The natural frequency of the desired turbine speed trajectory is 354.
selected first to be high or low, depending on the driver's request. The natural frequency of the desired engine speed trajectory is selected subsequently so that it is compatible with the turbine speed trajectory desired.

The magnitude of the change in the desired turbine speed is relatively fixed, being the difference in the turbine speed prior to application of the forward clutch and that after clutch lockup. The latter value being zero for the vehicle as it starts from rest, the magnitude of the change in turbine speed is equal to its value prior to clutch application. The desired engine speed change is not fixed, however. A larger desired engine speed change would result in higher engine speed at clutch lockup and higher vehicle acceleration. There is, thus, flexibility in the specification for achieving different vehicle acceleration characteristics during the neutral idle shift, within the limits of actuator capability.

The model system response can be chosen to be moderately damped, critically damped or over-damped. Lightly damped responses are clearly not desirable since the speeds would be fluctuating. The model damping ratio can be selected in conjunction with the natural frequency and the magnitude of the desired speed change to improve passenger perception of shift quality, as indicated below. Hayabuchi et al. (1996) show that passenger perception of a shift is good when the rate of turbine speed change is between 2600 - 4000 rpm/sec as shown in Figure 6.5. The best shift quality occurs in the range 3000 - 3600 rpm/sec. The rationale for relating shift quality to turbine rate of speed change is a reasonable one in the absence of direct access to final drive shaft speed. Clearly, passenger perception of shift quality would be more directly related to the rate of change of the final drive shaft. When the forward clutch is slipping, the rate of final drive shaft speed change is independent of the rate of the turbine speed change. However, in our current problem formulation, we omitted the vehicle speed from the state and, consequently, have to rely on the indirect link with turbine speed.

Figure 6.6 and Figure 6.7 show a series of 2\textsuperscript{nd} order step responses and the rate of speed changes assuming a natural frequency of 20 rad/sec, and a step change in desired speed of 680 rpm. In Figure 6.6, the four terminated ramp signals yield rates of speed change corresponding to passenger perception of good shift quality, i.e. 2600 - 4000
rpm/sec. The trajectories corresponding to damping ratios of 1, 1.2 and 1.5 result largely in speed rates of change in this range, for this natural frequency and speed change magnitude, and would be perceived as good shifts. Figure 6.7 shows furthermore that the rate of change of speed corresponding to a damping ratio of 1.2 lies within the 'optimum' range for a longer time interval and may be viewed as better in that respect. It is fair to state, however, that damping ratios of 1 and 1.5 are nearly as good. We should note also that if the natural frequency changes, the appropriate damping ratio would change if the step change in the turbine speed remained the same.

Choosing the engine model response parameters is dictated by several factors, such as the desired duration of engine speed rise, control effort constraints, desired vehicle acceleration at clutch lockup, and permitted level of engine induced vibrations. After the turbine speed response parameters are selected, the engine speed trajectory is chosen here to have a similar natural frequency and damping ratio. There is flexibility in the specification of these parameters that we are not exploiting here. That is, the engine response natural frequency and damping ratio can be different from the corresponding values for turbine response. The consequences of specifying different values for the resulting neutral idle shift remain to be explored.
According to Naruse et al. (1993), human sensitivity to shift shock varies with frequency as shown in Figure 6.8. Human response is more sensitive to fore and aft vibration in the low frequency range involving frequencies between 1.1 – 2.2 Hz approximately, and to vertical vibration in the high frequency ranges involving frequencies between 5 and 15 Hz. Neutral idle shifts could result in for and aft vibration if the compliance of transmission elements is significant. Vertical vibrations could result from interaction with the suspension system, or from transmitted engine vibrations via engine mounts. Better understanding of the relationship between engine and turbine response during neutral idle shifts and for and aft and vertical vehicle vibrations would enable more rational specification of the model parameters. Such understanding may come either from quantitative modeling of these relationships or empirical observations obtained during tests. In the following sections, different combinations of parameters characterizing the desired engine and turbine responses are presented. These combinations represent fast and slow clutch applications, and the corresponding controller designs and evaluations are presented.

Figure 6.6 Speed trajectories at different damping ratios, step input: 680 rpm, $\omega_n = 20$ rad/sec.
Rate of speed change with step of 680 rpm

Figure 6.7 Rate of speed change at different damping ratios, step input: 680 rpm, $\omega_n = 20$ rad/sec.

Figure 6.8 Human response sensitivity to frequency of vibration (Naruse et al., 1993)
6.3.4.1 Fast desired response

In this case, the desired engine and turbine speed trajectories have a faster response as compared to the slow response case to be presented in section 6.3.4.2. The following parameters corresponding to the desired response are selected:

\[
\begin{align*}
\omega_{\text{ne}} &= 20 \text{ rad/sec} \\
\omega_{\text{ns}} &= 20 \text{ rad/sec} \\
\zeta_e &= 1.2 \\
\zeta_t &= 1.2 \\
K_e &= 1 \\
K_t &= -1 \\
\omega_{\text{ce}} &= 500 \text{ rpm} \\
\omega_{\text{ct}} &= 680 \text{ rpm}
\end{align*}
\]

(6.156)

The performance index is given by equation (6.142), i.e.

\[
J = 0.5 \int_0^{t_f} (e^T Q e + u^T R u) dt
\]

(6.157)

where

\[
e = \begin{bmatrix} \omega_e - \omega_{ed} \\ \omega_t - \omega_{td} \end{bmatrix} = [C - C_d] \begin{bmatrix} x_r \\ x_d \end{bmatrix}
\]

(6.158)

\[
u = \begin{bmatrix} \alpha_{\text{com}} \\ V_{\text{in}} \end{bmatrix}
\]

(6.159)

In the linear quadratic optimal design, a trade-off must be made between control effort and output performance. The choice of the weighting matrices \(Q\) and \(R\) can be related to factors such as the expected magnitudes of key variables, desired closed-loop eigenvalues etc. (Anderson and Moore, 1990; Lin, 1994). It has been suggested that choosing \(R\) to be diagonal improves controller robustness properties. According to Anderson and Moore (1990), if \(R\) is diagonal in the multivariable case, the gain margin in
each loop is $\left(\frac{1}{2}, \infty\right)$, and the phase margin is 60 degrees or more. We choose both $Q$ and $R$ to be diagonal here.

Initial designs of $Q$ and $R$ are selected as follows to trade off between the tracking energy and control energy (Anderson and Moore, 1990; Lin, 1994). Trade-off studies are performed by varying parameters in the cost function.

\begin{align*}
Q &= \begin{bmatrix} q_1 & 0 \\ 0 & q_2 \end{bmatrix} \\
R &= \begin{bmatrix} r_1 & 0 \\ 0 & r_2 \end{bmatrix}
\end{align*}

where

\begin{align*}
q_i &= \frac{1}{[\left(e_i\right)_{\text{max}}]^2} \\
r_i &= \frac{1}{[\left(u_i\right)_{\text{max}}]^2}
\end{align*}

$(e_i)_{\text{max}}$ is the maximum expected value of the $i^{th}$ tracking error, and $(r_i)_{\text{max}}$ is the maximum expected value of the $i^{th}$ control input. This initial design is suggested by Lin (1994) based on Bryson and Ho (1975). The rationale here is to implicitly trade off between tracking and control energy performance. After some iterations, the following numerical values are selected,

\begin{align*}
Q &= \begin{bmatrix} 0.9119 & 0 \\ 0 & 0.9119 \end{bmatrix} \\
R &= \begin{bmatrix} 0.1 & 0 \\ 0 & 2.5 \end{bmatrix}
\end{align*}
The clutch pressure unit has to be scaled properly when solving the feedback gain matrices. The following gain matrices are corresponding to clutch pressure unit being $10^4 Pa$. The units of the other states are the same as those described in the model. The resulting feedback gain matrix is:

$$F_1 = \begin{bmatrix} -1.962 & -171.2087 & -0.3206 & -1.141 & -0.0383 & 0.011 & 0 \\ -0.018 & -4.5134 & -0.0246 & -0.2154 & -0.5281 & 0.2875 & 0.002 \end{bmatrix}$$

(6.166)

The model response state gain matrix is:

$$F_2 = \begin{bmatrix} 0.7513 & 0.0201 & -0.3266 & -0.0074 \\ 0.0811 & 0.0017 & 0.5697 & 0.0062 \end{bmatrix}$$

(6.167)

The feedforward gain matrix is:

$$K_{FF} = \begin{bmatrix} 2.1871 & 0.3332 \\ 0.0514 & -0.0196 \end{bmatrix}$$

(6.168)

The eigenvalues of the closed-loop system are listed in (6.169).


(6.169)
Figure 6.9 – Figure 6.11 show the simulation results for fast trajectory tracking with air-fuel ratio at stoichiometric (14.7). When the forward clutch slip is smaller than $5 \text{ rad/sec}$, the control loop is opened. The forward clutch pressure is ramped up to the supply pressure and clutch lockup occurs under open loop control. As shown in Figure 6.9, the engine speed tracks the desired trajectory and rises smoothly. The lag in the engine speed trajectory is caused by the intake-to-production delay $\Delta t_a$, which is 65 ms at idle. The turbine speed decreases smoothly as well until the forward clutch is locked up. The engaging process lasts about 0.21 seconds, which is fairly fast. The maximum engine speed tracking error is about 250 rpm due to the delay, and is about 80 rpm if compared to $\omega_d \cdot e^{-\Delta t_s}$. The maximum turbine speed tracking error is about 45 rpm. The vehicle speed is below 0.15 km/h when the forward clutch engages. Figure 6.10 shows vehicle acceleration, the output shaft torque and jerk of the vehicle during the clutch application phase. The sharp rise in the output shaft torque is due to the change of friction coefficient when lockup occurs. The maximum value of the jerk during the neutral idle shift is 0.65 g/sec and the vibration frequency is about 18 Hz, which differs significantly from the frequency range of maximum human response sensitivity for fore and aft vibration. Note that the magnitude of the jerk at lock up is not controlled by the closed loop action. Figure 6.11 shows the throttle response and forward clutch pressure during the clutch application phase.
Figure 6.9 Fast tracking: Engine, turbine and vehicle speed trajectories
Figure 6.10 Fast tracking: Vehicle acceleration, output shaft torque and jerk
In order to improve the engine speed quickness of response, a more aggressive design is implemented. The parameters chosen for the desired model are:

\[
\begin{align*}
\omega_{ae} &= 30 \text{ rad/sec} \\
\omega_{at} &= 20 \text{ rad/sec} \\
\zeta_e &= 1.2 \\
\zeta_r &= 1.2 \\
K_e &= 1 \\
K_r &= -1 \\
\omega_{er} &= 500 \text{ rpm} \\
\omega_{ir} &= 680 \text{ rpm}
\end{align*}
\] (6.170)

Figure 6.11 Fast tracking: Throttle angle and forward clutch pressure
The weighting matrices are selected to have more weighting on the tracking performance. The penalty on the throttle angle command is also decreased, allowing aggressive throttle response.

\[ Q = \begin{bmatrix} 9.1189 & 0 \\ 0 & 9.1189 \end{bmatrix} \]  \hspace{1cm} (6.171)

\[ R = \begin{bmatrix} 0.0063 & 0 \\ 0 & 2.5 \end{bmatrix} \]  \hspace{1cm} (6.172)

The feedback and feedforward gain matrices are:

\[ F_1 = \begin{bmatrix} -6.8126 & -1.6571e3 & -5.8444 & -22.5676 & -0.5140 & 0.0974 & 0 \\ -0.0063 & -6.3400 & -0.0497 & -0.5103 & -1.7394 & 0.6605 & 0.0032 \end{bmatrix} \]  \hspace{1cm} (6.173)

The model response state gain matrix is:

\[ F_2 = \begin{bmatrix} 10.7466 & 0.2051 & -4.2493 & -0.0843 \\ 0.1831 & 0.0026 & 1.8533 & 0.0154 \end{bmatrix} \]  \hspace{1cm} (6.174)

\[ K_{FF} = \begin{bmatrix} 26.9872 & 1.6758 \\ 0.1132 & -0.0335 \end{bmatrix} \]  \hspace{1cm} (6.175)

The closed-loop eigenvalues are:

\[ L = \begin{bmatrix} -92.746 + 163.52i \\ -92.746 - 163.52i \\ -185.2518 \\ -20.9795 + 37.2221i \\ -20.9795 - 37.2221i \\ -39.3255 + 5.8932i \\ -39.3255 - 5.8932i \\ -55.8997 \\ -37.2665 \\ -10.7335 \\ -16.1003 \end{bmatrix} \]  \hspace{1cm} (6.176)
The simulation results are shown in Figure 6.12 - Figure 6.14. The maximum turbine speed tracking error is about 40 rpm. The maximum engine speed tracking error is about 300 rpm due to delay $\Delta t_e$ and 40 rpm if compared to $\omega_{ed} e^{-\Delta t_e}$, which is lower than in the previous design. If we examine the throttle angle trace shown in Figure 6.14, the throttle is wide open until close to 0.1 second. Figure 6.13 vehicle acceleration, output shaft torque and jerk of the vehicle during the clutch application phase. The vibration frequency is about 25 Hz, which is higher than that shown in the previous fast tracking case. This is also acceptable in terms of its corresponding to a frequency region of lower human response sensitivity to fore and aft vibration. But the magnitude of the jerk signal during the neutral idle shift is higher, being a little over 1g/sec. Compared to the results given in Figure 6.9 - Figure 6.11, this design results in faster response and more accurate tracking of the desired response.

With greater $\omega_{ne}$, the desired engine speed trajectory has a steeper slope. This, in turn, drives the throttle to wide open. In order to see its effect clearly, we compare the simulation results for $\omega_{ne} = 30$ rad/sec and $\omega_{ne} = 8$ rad/sec shown in Figure 6.15 - Figure 6.17. The other conditions are the same as those shown in Figure 6.12 - Figure 6.14. Figure 6.15 shows engine and turbine speed traces. During neutral-to-first gear clutch application phase, the engine speed rise is lower with $\omega_{ne} = 8$ rad/sec. The engine speed tracking error is also smaller. The turbine speed tracking error is about 30 rpm, which is greater than that corresponding to $\omega_{ne} = 30$ rad/sec. Figure 6.16 shows the comparisons of output shaft torque and jerk. In the case of $\omega_{ne} = 8$ rad/sec, the jerk is lower, about 0.8 g/s, and the output shaft torque at lockup is also lower, about 600 Nm. These comparisons show that a greater engine speed natural frequency causes higher output shaft torque at lockup, which is less desirable. Figure 6.17 shows the throttle angle and clutch pressure responses corresponding to the two natural frequencies. The throttle angle response has much lower amplitude at the lower natural frequency. This indicates that a higher engine speed natural frequency $\omega_{ne}$ places a greater demand on the throttle actuator. The demand on the pressure actuator is also greater.
Figure 6.12 Engine, turbine, slip trajectory and vehicle speed for $\omega_{ke} = 30 \text{ rad/sec}$
Figure 6.13 Vehicle acceleration, output shaft torque and jerk for $\omega_{ne} = 30 \text{ rad/sec}$
Figure 6.14 Throttle angle and forward clutch pressure response $\omega_{ne} = 30 \text{ rad/sec}$

Figure 6.18 - Figure 6.20 compare the simulation results with two different engine speed reference step input, i.e. 500 rpm and 100 rpm. Figure 6.18 shows the engine and turbine speed traces with $\omega_{e}$ being 100 rpm. The maximum engine speed tracking error is about 80 rpm, and the maximum turbine speed tracking error is about 40 rpm. Figure 6.19 shows the comparison of output shaft torque and jerk. The output shaft torque is lower for the smaller step input case. The shaft torque at lockup is about 480 Nm for 100 rpm step input, and about 880 Nm for 500 rpm input. Jerk is also lower at smaller step input. Figure 6.19 compares the throttle angle and clutch pressure response for the two cases. Higher step input places greater demand on throttle and clutch actuators. These comparisons suggest that lower step input and lower natural frequency of desired engine speed result in better performance, and the less demand on throttle and clutch pressure actuators.
Figure 6.15 Comparison of engine and turbine speed traces with different \( \omega_{ne} \),

\[ \omega_{er} = 500 \text{rpm} \]
Figure 6.16 Output shaft torque and jerk with different $\omega_{ne}$, $\omega_{er} = 500$rpm

Figure 6.17 Throttle and clutch pressure with different $\omega_{ne}$, $\omega_{er} = 500$rpm
Figure 6.18 Engine and turbine speed trajectories: $\omega_{er} = 100\ rpm$, $\omega_{ne} = 30\ rad/sec$

Figure 6.19 Comparison of output shaft torque and jerk with different engine speed steps
Some aspects of the performance of the two controllers for the two cases investigated can also be investigated by using singular values (also called principal gains) of the return ratio at the input to the plant (Maciejowski, 1989; Anderson and Moore, 1990). Since the stability margins are guaranteed to be good for the linear quadratic designs and the nominal stability of the controlled systems is not in question, we can compare these designs on the basis of the principal gain plots. The return ratio is

$$\text{Return\_ratio} = -F_i (sI - A)^{-1} B$$ (6.177)

The singular values of the system with these two controllers are plotted in Figure 6.21. The singular values of the sensitivity function $S(j\omega)$ and the complementary sensitivity function $T(j\omega)$ are plotted in Figure 6.22 and Figure 6.23 respectively.
For good tracking, we need (Anderson and Moore, 1990),

$$\bar{\sigma}[S(j\omega)] << 1$$

(6.178)

where $\bar{\sigma}$ denotes the largest singular value, and $S(j\omega)$ is the sensitivity function

$$S(j\omega) = (I + F_i(sI - A)^{-1} B)^{-1}$$

(6.179)

and $F_i(sI - A)^{-1} B$ is the return ratio. For good disturbance rejection as well, $S(j\omega)$ should be small in the same sense. For good noise suppression and robustness to plant parameter variations, the complementary sensitivity function $T(j\omega)$ should be small, i.e.,

$$\bar{\sigma}[T(j\omega)] << 1$$

(6.180)

where

$$T = I - S$$

(6.181)

As shown in Figure 6.21, the loop gains corresponding to the second design are higher than those corresponding to the first design as could be expected from the faster engine response speed specification. The singular values at low frequencies are also higher for the second design, which explains the smaller tracking errors in the second design. Of course, given the larger weighting matrix elements in $Q$ for the second design, we should expect this behavior as well. Figure 6.22 also shows that the singular values of the sensitivity function corresponding to the second design are smaller than those corresponding to the first design, which indicates better tracking accuracy and disturbance rejection. Figure 6.23 shows the singular values of the complementary sensitivity functions corresponding to the two designs. The second design has poorer stability robustness as its singular values are higher. Again, this is what would be expected given the higher speed of response demanded in the form of the specifications. An issue that is not addressed here is the level of modeling uncertainty characteristic of the application. We had identified some of the simplifications associated with the model used for controller design earlier. The consequences of these simplifications for modeling uncertainty in terms of its frequency domain characterization remain to be quantified.
Here, we will only examine the effect of air-fuel ratio disturbance on the controller design by simulation.

Figure 6.24 and Figure 6.25 show the comparisons of the second fast desired response given above with the dynamic responses under air-fuel ratio disturbance of a pulse with frequency of 20 Hz, and magnitude of 4. Figure 6.24 shows the engine and turbine speed trajectories. With air-fuel ratio disturbance, the maximum engine speed tracking error is about the same as that without disturbance, but has lag. The output shaft torque is also about the same. With disturbance, the jerk is more oscillatory than that without disturbance. The frequency of the oscillation is about the same for the two cases. This comparison indicates that the controller performance is acceptable under air-fuel ratio disturbance.

Figure 6.21 Singular values of return_ratio for the two designs
Figure 6.22 Singular values of $S(j\omega)$ for the two designs

Figure 6.23 Singular values of $T(j\omega)$ for the two designs
Figure 6.24 Comparison of engine and turbine speeds with and without AFR disturbance:

\[ \omega_{ne} = 30 \text{ rad/sec} \]
Figure 6.25 Comparison of output shaft torque and jerk with and without AFR disturbance; $\omega_{ne} = 30 \ rad/sec$
6.3.4.2 Slow desired response

A slow desired response is specified here. The parameters for the slow desired response trajectory are specified as follows:

$$\omega_{ne} = 8 \text{ rad/sec}$$
$$\omega_{nu} = 8 \text{ rad/sec}$$
$$\zeta_e = 1.2$$
$$\zeta_t = 1.2$$
$$K_e = 1$$
$$K_t = -1$$
$$\omega_{er} = 500 \text{ rpm}$$
$$\omega_{r} = 680 \text{ rpm}$$

The $Q$ and $R$ matrices are the same as those given by equations (6.164) and (6.165). The computed feedback and feedforward gain matrices are:

$$F_1 = \begin{bmatrix}
-1.962 & -171.2087 & -0.3206 & -1.141 & -0.0383 & 0.011 & 0 \\
-0.018 & -4.5134 & -0.0246 & -0.2154 & -0.5281 & 0.2875 & 0.0020
\end{bmatrix}$$

(6.183)

$$F_2 = \begin{bmatrix}
1.9466 & 0.1115 & -0.5366 & -0.0239 \\
0.1157 & 0.0045 & 0.5860 & 0.0076
\end{bmatrix}$$

(6.184)

$$K_{ff} = \begin{bmatrix}
0.9918 & 0.1232 \\
0.0168 & -0.0032
\end{bmatrix}$$

(6.185)
The eigenvalues of the closed loop system are:

\[
P = \begin{bmatrix}
-62.705 + 112.6i \\
-62.705 - 112.6i \\
-124.6473 \\
-31.6479 \\
-9.8047 + 15.7929i \\
-9.8047 - 15.7929i \\
-20.9236 \\
-14.9066 \\
-4.2934 \\
-14.9066 \\
-4.2934
\end{bmatrix}
\] (6.186)

Figure 6.26 – Figure 6.28 show the simulation results for trajectory tracking for the slow desired response. As shown in Figure 6.26, the engine speed rises smoothly and tracks the desired engine speed trajectory well. The maximum engine speed tracking error is about 110 rpm. The turbine speed also decreases smoothly until lockup occurs. The maximum turbine speed tracking error is about 190 rpm. The engaging process lasts about 0.48 seconds, which is longer as compared to the fast tracking cases considered earlier and is as expected. When the lockup occurs, the vehicle speed is around 0.32 km/h. Figure 6.27 shows the output shaft torque, vehicle acceleration and jerk traces during the clutch application phase. Compared to the faster response cases earlier, the output shaft torque is lower in the application phase, and torque at lockup is slightly higher. This is also as expected. Though the magnitude of the jerk during the closed loop control is less than 0.2 g/sec and is much lower as compared to the previous two cases, the frequency of the jerk signal is about 4 Hz. This falls within the range for frequencies for which human response sensitivity of fore and aft vibration is high and hence the shift may be perceived as a bad one. In addition, the longer period of clutch engagement could mean more clutch wear for the forward clutch. These comments suggest that specification of the desired engine and turbine speed responses need to be based on implications for
jerk and clutch wear. We note here that specification of the desired turbine response alone does not determine the vehicle jerk. This finding is an argument for including the vehicle velocity, acceleration, and jerk as states in the system model formulation so that limits on these variables can be accommodated more explicitly and naturally in the optimal control formulation. For the present, comparing the fast desired response specifications considered earlier with the slow desired response specification here, the fast response cases are more appropriate. Figure 6.28 shows the throttle angle response and the forward clutch pressure. The pressure trace is fairly smooth and the throttle levels are lower than in the earlier cases. Recall that the control loop is opened when the clutch slip drops below 5 rad/sec.

6.4 Conclusion

In this chapter, the neutral idle control strategy for the forward clutch application phase is developed. Linear analysis of the powertrain model is carried out. The linearized powertrain model is then combined with actuator models to form a complete linear system model, for purposes of controller design. Due to the MIMO nature of the control problem, an optimal linear quadratic regulator design technique is employed. The Explicit Model Following (EMF) method is used to allow the system dynamic response to track two given desired trajectories for engine and turbine speeds, simultaneously. The optimal LQR design for the system with EMF is carried out in a straightforward fashion. The desired response specifications are quite meaningful physically and flexible in terms how they allow the control system designer to specify system performance. Designs corresponding to the tracking of fast and slow desired trajectories are completed. The designed controllers are implemented on the nonlinear system simulation. The results show that the proposed control strategy can achieve satisfactory results. Improvements in the design procedure are also suggested. The systematic approach developed here can be extended to other applications as well.
Figure 6.26 Slow response case: Engine, turbine speed and vehicle speed trajectories
Figure 6.27 Slow response case: Vehicle acceleration, output shaft torque and jerk
Figure 6.28 Slow response case: Throttle angle and forward clutch pressure
CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Summary

This dissertation addresses the modeling and control of automotive powertrains with emphasis on stepped automatic transmissions. An extensive literature survey covers the state-of-the-art of modeling and control issues related to automatic transmissions. Although many applications of electronic control of transmission functions are reported in the literature, applications of integrated engine-transmission control, especially closed loop control based on quantitative models of powertrain response, are few in number. One such application identified is powertrain neutral idle control, which is an important aspect of this dissertation work.

A powertrain simulator is developed in a modular form using Matlab/Simulink. Models of automatic transmission dynamic response, of varying levels of complexity, are implemented. The simulation can be used to study transmission shift dynamics, and provide a simulation environment to test certain design configurations and shift control strategies. Based on this powertrain model, a first-second gear power-on upshift controller is designed and implemented in the simulation. The advantage of model-based controller design is demonstrated. The model-based approach identifies the dominant factors in the transmission shift response and leads to a straightforward controller design. It is also demonstrated that improved dynamic models of transmission behavior can lead
to improved control of shifts and better understanding of the dependence of transmission
dynamic behavior on its design.

Shifts involving clutch-to-clutch load transfer or rapid pressurizing of the input
clutch as in neutral-idle transitions require direct manipulation of the clutch pressure.
Understanding of the dynamic response of clutch pressure control systems is essential for
such applications. A clutch pressure control system is studied via theoretical analysis,
simulation and experiments. Closed-loop control of the clutch pressure is implemented in
the test setup, which employs combined feedforward and feedback control. The study
reveals dominant dynamics in the clutch pressure control system, and demonstrates the
effectiveness of the combined feedforward and feedback control approach.

The powertrain neutral-idle application requires integrated engine-transmission
control. The critical phase of the neutral-idle application is the application phase of the
forward clutch during neutral-to-first gear transition, which involves coordination of
engine and transmission functions. An MIMO linear quadratic regulation method is used
to design the controller for neutral-idle shifts. The controller is implemented in the
nonlinear simulator developed in this dissertation and demonstrates the effectiveness of
integrated powertrain control.

7.2 Contributions of the research

This dissertation addresses the modeling and control of automotive powertrains
with emphasis on automatic transmissions. The contributions of the dissertation are in the
following areas:

(i) An extensive survey of the state-of-the-art of powertrain modeling and control
issues with emphasis on automatic transmissions is provided, along with the
identification of the limitations and the need for improved modeling and
control of powertrain systems, especially the need for integrated powertrain
control.

(ii) Simulation models of automatic transmission shift processes, of varying levels
of complexity, suitable for dynamic response simulation, controller design and
validation, are presented, along with derivation procedures to analyze other transmission systems.

(iii) Linearized models of transmission dynamic response during the inertia phase (speed adjustment phase) of a first-second gear shift are determined from the above transmission simulation model. Procedures for obtaining such models directly from the Simulink block diagram, along with the modal analysis techniques, are presented.

(iv) Closed-loop shift control algorithms for the inertia phase of a first-second gear upshift are developed based on these linearized models, and shown to be effective on the simulation model of the powertrain.

(v) An open-loop control strategy for the first-second gear upshift, based on the powertrain model, is presented and shown to be effective in reducing the torque drop in the load transfer phase, and the torque overshoot in the inertia phase. The analysis reveals the need for integrated engine and transmission control to achieve satisfactory shift quality.

(vi) A detailed analysis of the dynamic response of the neutral-idle clutch pressure control system, along with a nonlinear simulation model verified by experiments, is presented. The critical dynamics in the clutch pressure control system are identified. A closed-loop controller design based on the identified model is implemented.

(vii) Formulation of the powertrain neutral-idle control problem is presented, along with an MIMO controller design for the powertrain neutral-idle control process.

The improved dynamic models of automatic transmissions and the first-second gear inertia phase controller design and the neutral-idle control strategies claimed here are analytical in nature and have been verified by means of simulation. Due to the unavailability of suitable experimental facilities allowing for full powertrain tests, experimental verification is limited to the component level, i.e. the neutral-idle pressure control system. The engine model in the powertrain simulation is based on a well-
established model in the literature. The automatic transmission configuration and parameters also come from the literature. However, the Simulink implementation of the full powertrain model is unique, and provides an analytical testing environment for control algorithm development. In addition, the enhanced transmission model with the hydraulic network uses data consistent with the physical setup, and is useful in studying transmission shift hydraulic system designs. While the simulations are carried out for upshifts, the ideas can be easily adapted to other shift scenarios. Furthermore, the MIMO design approach for the neutral-idle process can be applied to other powertrain control applications.

7.3 Recommendations for future research

The benefit of integrated engine and transmission control based on theoretical analysis is demonstrated in this work. The problems examined in this dissertation represent a limited scope of powertrain applications. Integrated engine and transmission control can be extended to more applications such as idle speed control, clutch-to-clutch shift control, closed loop torque phase control and powertrain power control. These problems are conventionally viewed either as engine control problem or transmission control problems, but not as powertrain system control problems. More benefits can clearly be gained via integrated engine and transmission control. Therefore, a systems perspective needs to be adopted to solve a broad range of powertrain control problems more effectively.

The first recommendation for future work is to validate the inertia phase and neutral-idle shift control strategies in an application environment, such as in vehicles or powertrain test cells. Experimental validation of these control strategies can lead to further refinement of these strategies.

The second recommendation is to extend the neutral-idle control strategy to the idle phase. The neutral-idle control strategy developed in this dissertation focuses on the forward clutch application phase. In practice, the neutral-idle control strategy needs to be extended to the idling phase where engine idle speed control and clutch pressure
regulation need to be performed. In addition, air-fuel ratio control needs to be considered together with neutral-idle control, if the air-lead approach is used in air-fuel ratio control. If fuel-lead approach is used in air-fuel ratio control, then air-fuel ratio control does not need to be considered together with neutral-idle control. Furthermore, the consequences of model simplifications for modeling uncertainty in terms of its frequency domain characterization remain to be quantified to examine issues of controller robustness.

The third recommendation is to validate the pressure control system modeling and control strategy in a wider operating range, such as in a real vehicle or powertrain test cell. The current test setup is limited by the temperature range and pressure range that the hydraulic power unit can operate in. In practice, the pressure control system has to operate over broader operating ranges appropriate for vehicles in service. In addition, state estimation of the clutch pressure remains to be explored based on sensed information at other valve locations, and incorporated into the neutral-idle controller design.

Finally, the integrated engine-transmission control strategy presented here should be extended to closed loop output shaft torque control in both the torque phase and inertia phase for one-way clutch to clutch and clutch-to-clutch shifts.
BIBLIOGRAPHY


APPENDIX A

TRANSMISSION DYNAMICS SIMULATOR

A.1 Transmission-vehicle model parameters

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<tr>
<th>Inertia Values (kg·m²)</th>
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<td>Numerical Value</td>
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<tr>
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<tr>
<td>Iₜ</td>
<td>Engine inertia</td>
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<td>Iₚ</td>
<td>Pump inertia</td>
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<td>Reaction sun lumped inertia</td>
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<td>Iᵣᵣᵟ</td>
<td>Reaction carrier-input ring lumped inertia</td>
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<tr>
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<tr>
<td>Nₛᵢ</td>
<td>No. of teeth in input sun gear</td>
<td>26</td>
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<td>Nᵣᵟᵢ</td>
<td>No. of teeth in input ring gear</td>
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<td>Nₛᵣ</td>
<td>No. of teeth in reaction sun gear</td>
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<td>Nᵣᵣᵟ</td>
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<td>Rₛᵢ</td>
<td>Input sun gear transformer modulus</td>
<td>.2955</td>
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<td>Rᵣᵟᵢ</td>
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<tr>
<td>$R_{Sr}$</td>
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<td>$R_{Rr}$</td>
<td>Reaction ring gear transformer modulus</td>
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<td>$R_{D}$</td>
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### Plate Clutch Parameters

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<tr>
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<td>.056 m$^2$</td>
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<td>Rc1</td>
<td>Effective radius - clutch 1</td>
<td>.052 m</td>
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<td>Ac2</td>
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### Band Clutch Parameters

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<td>$B_{12}$ band piston area</td>
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<td>$B_{12}$ drum radius</td>
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<td>$A_{BR}$</td>
<td>$B_R$ band piston area</td>
<td>.0044 m$^2$</td>
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<td>$R_{BR}$</td>
<td>$B_R$ drum radius</td>
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### Drivetrain-Vehicle Parameters

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<td>$K_s$</td>
<td>Combined axle shaft stiffness</td>
<td>6742 Nm/rad</td>
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<td>$I_{wf}$</td>
<td>Combined front wheel inertia</td>
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<td>$h_f$</td>
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<td>$T_{tf}$</td>
<td>Tire reaction torque, front</td>
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<td>$I_{wr}$</td>
<td>Combined rear wheel inertia</td>
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Table A.1 (continued)

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<th>h_r</th>
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<td>T_{rr}</td>
<td>Tire reaction torque, rear</td>
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<td>M</td>
<td>Vehicle mass</td>
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<td>K_i</td>
<td>Tire force-slip gain</td>
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<td>C_a</td>
<td>Aerodynamic drag coefficient</td>
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Table A.1 Transmission-vehicle model parameters

A.2 Overview of “Transmission Dynamics Simulator”

Figure A.1 Overview of “Transmission Dynamics Simulator”
Figure A.2 Engine subsystem

Figure A.3 Torque converter subsystem
Figure A.5 Shift hydraulic system model overview
Figure A.6 Transmission mechanical system model overview
A.3 Simulink block diagram for 1-2 clutch accumulator assembly

![Simulink block diagram](image)

Figure A.7 Overview of clutch-accumulator assembly model

A.4 Gross vehicle dynamics model

![Gross vehicle dynamics model](image)

Figure A.8 Gross vehicle dynamics model
APPENDIX B

NEUTRAL IDLE PRESSURE CONTROL VALVE MODEL

B.1 Overview of neutral-idle pressure control valve model

Figure B.1 Overview of neutral idle pressure control system model
APPENDIX C

SIMULINK BLOCK DIAGRAMS FOR NEUTRAL IDLE CONTROL SYSTEM

C.1 Neutral idle control system model

Figure C.1 Simulink block diagram of neutral-idle control system
C.2 Neutral idle controller

Figure C.2 Neutral idle controller