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Investigation into multiple-speed transmissions for electric vehicles

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Abstract

The aim of the research is to investigate multiple-speed transmissions for electric vehicles. This research is driven by the requirement to reduce emissions within the automotive industry increasing the demand for electric vehicles. The typical torque characteristics of an electric motor allow a clutchless single-speed transmission to be used, yet it is suggested by literature that the adoption of multiple-speed transmissions can benefit the energy consumption and vehicle performance. However, the published research up to date is limited in this field and lacks credible quantifiable evidence and as such motivates this research. The author developed complex non-linear models in Matlab/Simulink of case study vehicles with single and multiple-speed transmissions to analyse vehicle performance and simulate driving cycles to calculate energy consumption. The main focus of the research was based around a single and two-speed transmission developed by Vocis Drivelines and Oerlikon Graziano. The two-speed transmission has a novel mechanical layout comprising a friction clutch and sprag clutch allowing seamless gearshifts, a gearshift controller was developed as part of the research. The two transmissions were modelled in simulation with the gearshift dynamics of the two-speed transmission being simulated and considered with multiple controllers. In addition, a hardware-in-the-loop test rig was built at the University of Surrey by the author to test the prototype single and two-speed transmissions. The vehicle models were validated using the hardware-in-the-loop test rig whilst allowing performance tests and driving cycles to be carried out. The research showed that the adoption of the two-speed transmission over the single-speed transmission gave rise to reductions in energy consumption over numerous driving cycles of up to 4% for the case study vehicles. The vehicle performance was also improved with the top speed increased by 12% and the 0-100 km/h time reduced by 2%. In addition, a novel four-speed dual-motor drivetrain was investigated through simulations and compared with optimised single-speed and two-speed variants. The novel four-speed transmission delivered up to a 9% and 5% improvement in energy consumption during standard driving cycles over the single-speed and two-speed transmissions, respectively. The four-speed transmission allowed up to a 25% improvement in top speed and a 10% improvement in 0-100 km/h time over the two-speed transmission.
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### Chapter 2: Review of the state of the art

- $T_e$: Engine torque
- $\dot{\omega}_e$: Engine acceleration
- $T_c$: Clutch torque
- $x_c$: Clutch displacement
- $J_c$: Moment of inertia of the clutch
- $i_d$: Differential gear ratio
- $\dot{\omega}_c$: Clutch acceleration
- $k_{tw}$: Driveshaft stiffness
- $\Delta \theta_{cw}$: Difference in angular displacement of the differential and wheel
- $\beta_{tw}$: Driveshaft damping ratio
- $\omega_c$: Clutch rotational velocity
- $\omega_w$: Wheel rotational velocity
- $J_{eq}$: Equivalent moment of inertia of the system
- $J_m$: Moment of inertia of the main gear
- $J_{s1}$: Moment of inertia of the primary shaft
- $J_{s2}$: Moment of inertia of the secondary shaft
- $J_t$: Moment of inertia of the differential
- $J_e$: Moment of inertia of the engine
- $\bar{\epsilon}^{-}$: Time before clutch engagement
- $\bar{\epsilon}^{+}$: Time after clutch engagement
- $\dot{\omega}_{sl}$: Clutch slip acceleration
- $C_{1-4}$: PI controllers
- $\Gamma_e$: Engine torque
- $F_N$: Normal force on the clutch friction discs
- $J_g$: Gearbox moment of inertia
- $\dot{\omega}_g$: Gearbox rotational acceleration
- $k_t$: Transmission stiffness
- $\theta_g$: Transmission angular torsion
- $\beta_t$: Transmission damping coefficient
- $\omega_g$: Gearbox rotational velocity
- $\omega_v$: Equivalent vehicle velocity
- $J_v$: Equivalent vehicle moment of inertia
- $\gamma$: Frictional parameter
- $\mu_d$: Coefficient of friction
- $R_d$: Clutch radius
- $\omega_e$: Engine rotational velocity
- $\Gamma_c$: Clutch torque
- $z_1$: Engine and gearbox velocity angular velocity difference
- $z_2$: Gearbox and vehicle angular velocity difference
- $p_i$: Clutch pressure
- $\beta_i$: Engine load reduction
- $T_c$: Clutch torque
- $F_n$: Normal clutch force
- $\mu$: Coefficient of friction of the clutch
\begin{align*}
R_a & \quad \text{Clutch radius} \\
\iota_{tot} & \quad \text{Total drivetrain gear ratio} \\
T_s & \quad \text{Wheel torque} \\
A_{1/2} & \quad \text{State space matrix} \\
B_{1/2} & \quad \text{State space matrix} \\
x & \quad \text{State vector} \\
u & \quad \text{Control vector} \\
a_n & \quad \text{Vehicle acceleration} \\
i_n & \quad \text{Gear ratio} \\
i_a & \quad \text{Final drive ratio} \\
R_{aer} & \quad \text{Aerodynamic resistance force} \\
R_f & \quad \text{Force due to rolling resistance} \\
R_i & \quad \text{Road inclination} \\
\delta_n & \quad \text{Equivalent mass of the vehicle} \\
m & \quad \text{Vehicle mass} \\
\end{align*}

Chapters 3 to 7.

\begin{align*}
F_{aer} & \quad \text{Aerodynamic force} \\
\rho & \quad \text{Air density} \\
S & \quad \text{Frontal area of the vehicle} \\
C_d & \quad \text{Aerodynamic drag coefficient} \\
V & \quad \text{Vehicle velocity} \\
F_{roll} & \quad \text{Force due to rolling resistance} \\
f_0 & \quad \text{Constant rolling resistance coefficient} \\
f_1 & \quad \text{Rolling resistance coefficient relating to vehicle velocity} \\
f_2 & \quad \text{Rolling resistance coefficient relating to vehicle velocity squared} \\
\ddot{x}_{sm} & \quad \text{Sprung mass longitudinal acceleration} \\
\ddot{z}_{sm} & \quad \text{Sprung mass vertical acceleration} \\
\ddot{\theta}_{sm} & \quad \text{Sprung mass rotational acceleration} \\
F_{fz} & \quad \text{Longitudinal force transmitted by the suspension arms through the suspension joints} \\
F_{Rgz,sm} & \quad \text{Resistive force for the sprung mass due to the inclination of the road} \\
m_{sm} & \quad \text{Mass of the sprung mass} \\
F_{fz} & \quad \text{Vertical force transmitted by the suspension arms through the suspension joints} \\
F_{fs} & \quad \text{Vertical force caused by the suspension spring and damper system} \\
\Delta F_{Rg,z,sm} & \quad \text{Force due to the weight variation of the sprung mass when there is an inclination of the road} \\
J_{sm} & \quad \text{Moment of inertia of the sprung mass} \\
H_{CG,sm} & \quad \text{Height of the centre of gravity} \\
a & \quad \text{Longitudinal distance from the front axle to the sprung mass centre of gravity} \\
b & \quad \text{Longitudinal distance from the rear axle to the spring mass centre of gravity} \\
c & \quad \text{Longitudinal distance from the front axle to the} \\
\end{align*}
equivalent front suspension mounting point
Longitudinal distance from the rear axle to the rear equivalent suspension mounting point
Height of the front equivalent suspension mounting point
Height of the rear equivalent suspension mounting point
Force due to rolling resistance
Wheel radius
Unsprung mass longitudinal displacement
Vertical displacement of the wheel
Angle of the trailing arm
Sprung mass vertical displacement
Tyre longitudinal force
Resistive force for the unsprung mass due to the inclination of the road
Mass of the unsprung mass
Road angle
Gravity
Unsprung mass vertical acceleration
Vertical tyre force
Force due to the weight variation of the unsprung mass when there is an inclination of the road
Half-shaft torque
Tyre stiffness coefficient
Tyre damping coefficient
Unsprung mass vertical velocity
Vertical displacement of the road
Vertical displacement rate of the road
Sprung mass vertical displacement
Sprung mass vertical velocity
Suspension stiffness coefficient
Suspension damping coefficient
Vertical displacement at the upper suspension mounting point
Sprung mass pitch
Slip ratio
Rotational velocity of the wheel
Equivalent rotational velocity of the vehicle
Tyre delay
Tyre relaxation length
Delayed slip ratio
Delayed motor torque
Force transmitted by the single-speed transmission input gear set
Radius of the single-speed transmission main gear set input gear
Moment of inertia of the electric motor
Single-speed transmission primary shaft
\( \dot{\theta}_{1s} \)  
Angular acceleration of the single-speed transmission primary shaft

\( F_{2s} \)  
Force transmitted to the single-speed transmission final drive set

\( R_{3s} \)  
Radius of the single-speed transmission final drive gear set input gear

\( R_{2s} \)  
Radius of the single-speed transmission main gear set output gear

\( \eta_{1s} \)  
Single-speed transmission main gear set efficiency

\( J_{2s} \)  
Moment of inertia

\( \dot{\theta}_{2s} \)  
Angular acceleration of the single-speed transmission secondary shaft

\( R_{4s} \)  
Radius of the single-speed transmission final drive gear set output gear

\( \eta_{2s} \)  
Single-speed transmission final drive efficiency

\( J_{3s} \)  
Moment of inertia of the single-speed transmission final drive shaft

\( J_{hs} \)  
Moment of inertia of the half-shaft

\( \ddot{\theta}_{3s} \)  
Angular acceleration of the single-speed transmission final drive shaft

\( \dot{\theta}_{diffs} \)  
Differential acceleration of the single-speed transmission

\( \tau_{1s} \)  
Single-speed transmission gear ratio

\( \tau_{diffs} \)  
Single-speed transmission final drive ratio

\( K_{hsL} \)  
Half-shaft stiffness

\( \beta_{hsL} \)  
Half-shaft damping coefficient

\( \dot{\theta}_{diff} \)  
Angular velocity of the differential

\( \theta_{diff} \)  
Torsional of the differential

\( \theta_{wr} \)  
Torsional angle of the wheel

\( F_1 \)  
Force transmitted by 2SED first gear set

\( R_1 \)  
Radius of the 2SED first gear set input gear

\( J_1 \)  
Moment of inertia of the 2SED primary shaft

\( \dot{\theta}_1 \)  
Angular acceleration of the 2SED primary shaft

\( F_1 \)  
Force transmitted by 2SED second gear set

\( R_2 \)  
Radius of the 2SED second gear input gear (which is on the friction clutch output)

\( \eta_2 \)  
2SED second gear efficiency

\( J_{1b} \)  
Moment of inertia of the 2SED friction clutch/gear assembly

\( \dot{\theta}_{1b} \)  
Rotational acceleration of the 2SED friction clutch output

\( R_4 \)  
Radius of the 2SED first gear set output gear

\( T_{C2} \)  
2SED sprag clutch output torque

\( J_{2b} \)  
Moment of inertia of the 2SED sprag clutch/gear assembly

\( \dot{\theta}_{2b} \)  
Rotational acceleration of the 2SED sprag clutch/gear assembly
\( R_3 \)
\( F_3 \)
\( R_5 \)
\( J_2 \)
\( \dot{\theta}_2 \)
\( R_6 \)
\( \eta_3 \)
\( \tau_1 \)
\( \tau_2 \)
\( \tau_{\text{diff}} \)
\( T_{\text{primary shaft}} \)
\( T_{\text{secondary shaft}} \)
\( \dot{\theta}_{\text{primary shaft}} \)
\( \dot{\theta}_{\text{secondary shaft}} \)
\( Temp_{\text{trans}} \)
\( P_{\text{loss,trans}} \)
\( \dot{Q}_{\text{trans,motor}} \)
\( \dot{Q}_{\text{trans,air}} \)
\( C_{p,\text{steel}} \)
\( m_{\text{steel}} \)
\( C_{p,\text{alu}} \)
\( m_{\text{alu}} \)
\( C_{p,\text{oil}} \)
\( m_{\text{oil}} \)
\( P_{\text{loss,gearbox}} \)
\( P_{\text{loss,diff}} \)
\( P_{\text{loss,t}} \)
\( P_{\text{out,t}} \)
\( \eta_i \)
\( A_{\text{trans,motor}} \)
\( Temp_{\text{motor}} \)
\( \alpha_{\text{trans,motor}} \)
\( \lambda \)
\( S \)
\( h_{\text{motor}} \)
\( h_{\text{trans}} \)
\( A_{\text{trans,air}} \)
\( Temp_{\text{air}} \)
\( \alpha_{\text{trans,air}} \)
\( P_{\text{loss,motor}} \)
\( Q_{\text{motor,air}} \)

Radius of the 2SED second gear set output gear
Force transmitted by 2SED final drive gear set
Radius of the 2SED final drive gear set input gear
Moment of inertia of the 2SED secondary shaft
Rotational acceleration of the secondary shaft
Radius of the 2SED final drive gear set output gear
2SED final drive efficiency
2SED first gear ratio
2SED second gear ratio
2SED final drive ratio
Primary shaft torque
Secondary shaft torque
Primary shaft angular velocity
Secondary shaft angular velocity
Transmission temperature
Power loss in the transmission
Heat exchange between the transmission and the motor
Heat exchange between the transmission and the environment
Specific heat capacity of steel
Steel mass in the transmission
Specific heat capacity of aluminium
Mass of aluminium in the transmission
Specific heat capacity of oil
Mass of oil in the transmission
Power loss in the gearbox
Power loss in the differential
Power loss (generic)
Power output (generic)
Efficiency (generic)
Heat transfer surface area between the transmission and the motor
Motor temperature
Heat transfer coefficient between the transmission and the electric motor
Separation surface thermal conductivity
Separation surface thickness
Heat transfer coefficient between the motor and the separation surface
Heat transfer coefficient between the transmission and the separation surface
Heat transfer surface area of the transmission
Air temperature
Heat transfer coefficient between the transmission and the electric motor
Power loss in the motor
Heat transfer between the motor and the
$\dot{Q}_{cooling}$
Heat transfer between the motor and the cooling water

$C_{p,motor}$
Specific heat capacity of the motor

$m_{motor}$
Mass of the electric motor

$A_{motor,air}$
Heat transfer surface area of the motor

$\alpha_{motor,air}$
Heat transfer coefficient between the motor and the environment

$c_{p,H_2O}$
Water specific heat capacity

$\rho_{H_2O}$
Water density

$q_{H_2O}$
Flow rate

$Temp_{H_2O}$
Water temperature

$T_{motor}$
Electric motor torque

$\omega_{motor}$
Electric motor speed

$P_{Inv.(Lim),traction}$
Power at the inverter in traction (limited)

$P_{Inv.(Lim),regeneration}$
Power at the inverter in regeneration (limited)

$SOC$ 
State of charge

$Q_{Battery}$
Battery capacity

$i_{battery}$
Battery current

$\beta$
Battery constant (a function of battery temperature)

$T_{Battery}$
Battery temperature

$V_{batt}$
Battery voltage

$i_{load}$
Previous simulation iteration battery current

$R_{1,batt}$
Total battery Ohm resistance

$P_{Inv.(Lim)}$
Power at the inverter (limited)

$\psi$
Equal to zero if term in Eqn. 87 is less than zero

$V_{module}$
Battery module voltage

$E_{module}$
Battery module equilibrium potential

$V_{R1,module}$
Potential loss across resistor bank one

$V_{R2/C,module}$
Potential loss across the capacitor and resistor bank 2

$E_{battery}$
Total battery equilibrium potential

$E_{NC}$
Initial battery equilibrium potential depending on state of charge

$\Delta E$
Battery equilibrium potential depending on battery temperature

$R_1$
First resistor resistance

$i_c$
Current through the capacitor

$i_{R2}$
Current through the second resistor

$R_2$
Second resistor resistance

$C$
Capacitance of the capacitor

$m_{battery}$
Mass of the battery

$c_{p,battery}$
Battery specific heat capacity

$h_{c,battery}$
Battery heat transfer coefficient

$A_{battery}$
Surface area of the battery

$P_{loss,battery}$
Power loss in the battery

$P_{loss,mod}$
Power loss in each battery module

$T_{fc}$
Friction clutch torque
\( t \)  
\( x \)  
\( u \)  
\( A \)  
\( B \)  
\( C \)  
\( D \)  
\( T_{m,\text{delay}} \)  
\( \tau_m \)  
\( T_{m,\text{theor.}} \)  
\( \tau_g \)  
\( \eta_g \)  
\( J_{\text{equiv}} \)  
\( F_{\text{rolling resistance,non linear}} \)  
\( F_{\text{rolling resistance,linear}} \)  
\( m_f \)  
\( \theta_{\text{wr,0}} \)  
\( F_{\text{xt,del}} \)  
\( B_{t,\text{equiv}} \)  
\( \tau_{\text{tyre const.}} \)  
\( F_{\text{aerodynamic force,non linear}} \)  
\( F_{\text{aerodynamic force,linear}} \)  
\( \theta_{\text{e,0}} \)  
\( \theta_{m,\text{ref}} \)  
\( DTD \)  
\( T_{fc,\text{saturation,IP,US}} \)  
\( \gamma \)  
\( t_{IP} \)  
\( K_p \)  
\( K_D \)  
\( K_I \)  
\( \tau_m \)  
\( T_{fc,\text{saturation,IP,US,C2}} \)  
\( T_{fc,\text{saturation,IP,US,C3}} \)  
\( T_{fc,\text{dis}} \)  
\( P_{\text{out}} \)  
\( P_{\text{in}} \)  
\( T_{\text{motor,70kW}} \)  
\( \omega_{\text{motor,70kW}} \)  
\( T_{\text{hub,left}} \)  

Time  
State vector  
Input vector  
State matrix  
Input matrix  
Output matrix  
Output matrix  
Delayed electric motor torque  
Motor time constant  
Theoretical electric motor torque  
Selected gear ratio  
Selected gear efficiency  
Equivalent moment of inertia  
Non-linearised rolling resistive force  
Linearised rolling resistive force  
Vehicle mass on the front axle  
Initial vehicle speed  
Delayed tyre longitudinal force  
Tyre damping coefficient  
Tyre time constant  
Non-linearised aerodynamic force  
Linearised aerodynamic force  
Initial equivalent vehicle velocity  
Electric motor angular speed reference during the inertia phase of the upshift  
Driver torque demand  
Saturation level of the reference friction clutch torque during the inertia phase of the upshift  
Adimensional parameter for the definition of the electric motor speed profile during upshift  
Time from the inertia phase start  
PID gain  
PID derivative gain  
PID integral gain  
Motor time constant  
Saturation level of the reference friction clutch torque during the inertia phase of the upshift, according to ‘Control 2’  
Saturation level of the reference friction clutch torque during the inertia phase of the upshift, according to ‘Control 3’  
Friction clutch torque value required for the disengagement of the sprag clutch during an upshift  
Power out of the transmission  
Power into the transmission  
Torque of the 70kW motor  
Speed of the 70kW motor  
Torque at the left hub
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{hub}, \text{right}}$</td>
<td>Torque at the right hub</td>
</tr>
<tr>
<td>$\omega_{\text{hub}, \text{left}}$</td>
<td>Angular velocity at the left hub</td>
</tr>
<tr>
<td>$\omega_{\text{hub}, \text{right}}$</td>
<td>Angular velocity at the right hub</td>
</tr>
<tr>
<td>$T_{\text{wheel, demand}}$</td>
<td>TTRP Wheel torque demand</td>
</tr>
<tr>
<td>$T_{\text{e, demand}}$</td>
<td>TTRP engine torque demand</td>
</tr>
<tr>
<td>$T_{\text{m, demand}}$</td>
<td>TTRP electric motor torque demand</td>
</tr>
<tr>
<td>$T_{\text{e, theor}}$</td>
<td>TTRP theoretical engine torque</td>
</tr>
<tr>
<td>$T_{\text{m, theor}}$</td>
<td>TTRP theoretical electric motor torque</td>
</tr>
<tr>
<td>$T_{\text{hs, est}}$</td>
<td>Estimated half-shaft torque</td>
</tr>
<tr>
<td>$G_{\text{PID, TTRP}}$</td>
<td>PID value for the TTRP</td>
</tr>
<tr>
<td>$\ddot{x}$</td>
<td>Longitudinal acceleration of the vehicle</td>
</tr>
<tr>
<td>$\tau_{gf}$</td>
<td>TTRP total front transmission ratio</td>
</tr>
<tr>
<td>$\tau_{gr}$</td>
<td>TTRP total rear transmission ratio</td>
</tr>
<tr>
<td>$P$</td>
<td>Proportional gain</td>
</tr>
<tr>
<td>$T_{\text{loss, odd/even}}$</td>
<td>Torque loss on the primary shaft of the ‘odd’/’even’ side of the drivetrain in conditions of both disengaged gears</td>
</tr>
<tr>
<td>$J_{\text{mot, odd/even}}$</td>
<td>Mass moment of inertia of the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$J_{1, \text{odd/even}}$</td>
<td>Mass moment of inertia of the ‘odd’/’even’ primary shaft of the transmission</td>
</tr>
<tr>
<td>$\ddot{\theta}<em>{\text{mot, odd/even}}, \dot{\theta}</em>{\text{mot, odd/even}}$</td>
<td>Angular acceleration and velocity of the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$T_{\text{mot, del, odd/even}}$</td>
<td>Delayed air gap torque of the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$T_{\text{windage, odd/even}}$</td>
<td>Windage torque of the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$T_{\text{mot, ref, odd/even}}$</td>
<td>Reference air gap torque of the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$TD_{\text{mot, odd/even}}$</td>
<td>Torque demand on the ‘odd’/’even’ motor</td>
</tr>
<tr>
<td>$\zeta_{\text{mot}}$</td>
<td>Damping ratio of the motor air gap torque characteristic</td>
</tr>
<tr>
<td>$\omega_{n, \text{mot}}$</td>
<td>Natural frequency of the motor air gap torque characteristic</td>
</tr>
<tr>
<td>$s$</td>
<td>Laplace variable</td>
</tr>
<tr>
<td>$x_{\text{gear, act}}$</td>
<td>Actual position of the gear actuator</td>
</tr>
<tr>
<td>$x_{\text{gear, ref}}$</td>
<td>Reference position of the gear actuator</td>
</tr>
<tr>
<td>$\tau_{\text{gear, act}}$</td>
<td>Time delay of the gear actuator</td>
</tr>
<tr>
<td>$T_{\text{gear, act}}$</td>
<td>Time constant of the gear actuator</td>
</tr>
<tr>
<td>flag$_{\text{sel, odd/even}}$</td>
<td>A Boolean variable (assuming 0 or 1 value) to select or deselect terms from the equation of motion of the system for the ‘odd’/’even’ side of the transmission</td>
</tr>
<tr>
<td>$\tau_{g, \text{sel, odd/even}}$</td>
<td>Gear ratio selected on the ‘odd’/’even’ primary shaft</td>
</tr>
<tr>
<td>$\eta_{g, \text{sel, odd/even}}$</td>
<td>Overall efficiency of the selected gear on the ‘odd’/’even’ primary shaft</td>
</tr>
<tr>
<td>$\tau_{\text{FRR}}$</td>
<td>Final reduction ratio (between the secondary shaft and the differential case)</td>
</tr>
<tr>
<td>$\eta_{\text{FRR}}$</td>
<td>Final reduction gear efficiency</td>
</tr>
</tbody>
</table>
\( T_{\text{mot, odd}} \) Torque of the ‘odd’ electric motor

\( J_{g, \text{unsel, odd/even}} \) Moment of inertia of the unselected gear on the ‘odd’/‘even’ primary shaft of the transmission

\( J_{g, \text{sel, odd/even}} \) Moment of inertia of the selected gear on the ‘odd’/‘even’ primary shaft of the transmission

\( T_{\text{HS, L/R}} \) Left/right half shaft torque

\( \dot{\theta}_{\text{diff}}, \dot{\theta}_{\text{diff}}^e, \theta_{\text{diff}} \) Angular acceleration, velocity and displacement of the differential

\( k_{\text{HS, L/R}}, c_{\text{HS, L/R}} \) Left/right half shaft torsion stiffness

\( \Delta \theta_{\text{diff, eq}}, \Delta \dot{\theta}_{\text{diff, eq}} \) Equivalent angular torsion angle and velocity between the differential and the wheel

\( \dot{\theta}_w, \theta_w \) Angular velocity and displacement of the wheel

\( \Delta \theta_{\text{play}} \) Equivalent backlash (play) in the transmission, at the output port

\( T_{\text{r, theor}}, F_{\text{x, theor}} \) Theoretical tyre torque and longitudinal force (without considering any dynamics)

\( C_s \) Tyre longitudinal slip stiffness

\( \dot{\theta}_v \) Equivalent angular velocity of the vehicle

\( \sigma \) Slip ratio

\( A_x, V \) Vehicle longitudinal acceleration, velocity

\( n_{\text{input}} \) Number of inputs to the state space system

\( A_{x,i} \) Vehicle longitudinal acceleration contribution due to the \( i^{th} \) input to the state space system

\( \phi_i \) Phase angle of the vehicle longitudinal acceleration contribution due to the \( i^{th} \) input to the state space system

\( u_i \) \( i^{th} \) input to the state space system

\( T_{\text{ref, odd/even, EMS}} \) Reference torque for the ‘odd’/‘even’ electric motor calculated by the EMS

\( T_{\text{w, ref}} \) Reference wheel torque

\( T_{\text{ref, motor constant gear, roll-off comp}} \) Reference torque to the motor on the transmission side in conditions of constant gear during the roll-off phase of the upshift

\( T_{\text{w, ref, motor gearshift, EMS}} \) Reference wheel torque contribution calculated by the EMS for the transmission side involved in the upshift

\( T_{\text{w, roll-off}} \) Estimated wheel torque for the transmission side involved in the upshift during the roll-off phase

\( T_{\text{w, ref, motor constant gear, EMS}} \) Reference wheel torque calculated by the EMS for the transmission side in constant gear during the upshift

\( \tau_{g, \text{mot constant gear}} \) Gear ratio of the transmission side in condition of constant gear during the upshift

\( T_{\text{motor, theor}} \) Theoretical motor torque

\( G_{\text{PID}} \) Transfer function of the Proportional Integral Derivative controller of the electric motor drive velocity
\( b_{\text{mot}} \)  
Equivalent damping coefficient of the electric motor drive and transmission primary shaft in condition of both disengaged gears on the primary shaft

\( T_{\text{resist}} \)  
Resistive wheel torque, i.e., rolling resistance and aerodynamic torques

\( T_{\text{min}} \)  
Minimum electric motor torque at the current velocity

\( T_{\text{max}} \)  
Maximum electric motor torque at the current velocity

\( T_{\text{even, output}} \)  
Even primary shaft output torque

\( T_{\text{FRR}} \)  
Differential output torque

\( k_{g, \text{set, even}} \)  
Equivalent efficiency parameter of the ‘even’ gear

\( P_{\text{input}} \)  
Total input power required by the electric drivetrain

\( \Delta E_{\text{Gearshift}} \)  
Energy required during the gearshift to achieve the electric synchronisation

\( k_{\text{equiv, odd/even}} \)  
‘odd’/‘even’ electric motor and optionally the energy storage system

\( \text{flag}_{0/1, \text{odd/even}} \)  
Boolean variable equal to 0 or 1 depending on the state before and after the gearshift

\( T_{T, \text{det}} \)  
Delayed tyre longitudinal torque

\( m \)  
Mass of the vehicle

\( L \)  
Wheel base

\( C_{0/1/2, \text{front/rear}} \)  
Rolling resistance coefficients (for the part independent from wheel velocity, the part linearly dependent on wheel velocity, and the part quadratically dependent on wheel velocity)

\( \tau_{\text{mot, odd/even}} \)  
Electric motor time constant of the ‘odd’/‘even’ motor

\( J_{\text{equiv.}} \)  
Equivalent moment of inertia

\( J_w \)  
Moment of inertia of the wheel

\( L_{\text{tyre}} \)  
Tyre relaxation length

\( a_{i,j} \)  
Element on the \( i^{th} \) row and \( j^{th} \) column of matrix \( A \)

\( b_{i,j} \)  
Element on the \( i^{th} \) row and \( j^{th} \) column of matrix \( B \)
1 INTRODUCTION

1.1 CONTEXT

During the past century there have been incredible advancements in technology driven by economics and warfare, however this has been achieved with little thought to any environmental consequences. The effect the complacency of previous generations had on the environment has resulted in concerns such as the Ozone layer depleting, sea levels rising and global warming. These concerns are currently driving industries to analyse their products and processes to increase efficiency and reduce emissions. The transport industry is a large contributor to global emission levels, 28% of total (EPA, 2012), split between aerospace, shipping, trains and, specifically, the automotive industry. The European Union have set CO₂ emission targets for car manufacturers to achieve where fines will be incurred if they are not met, which is a significant economic driver.

To reduce vehicle emissions vehicle manufacturers are required to look at all aspects of the vehicle design and analyse where improvements can be made. A key factor affecting fuel economy and emissions is the vehicle weight which can be reduced by utilising new materials and optimised vehicle layouts. The vehicle shape can be modified to reduce aerodynamic drag and low friction tyres adopted to reduce rolling resistance. In addition, the engine efficiency can be improved through engine downsizing by adopting turbochargers (Police et al., 2006) and different layouts, e.g. three cylinders (Ecker, Schwaderlapp and Gill, 2000). The drivetrain itself can be modified to include electric motors as well as internal combustion engines to create hybrid-electric vehicles and significantly reduce CO₂ emissions.

However, recently the development of fully electric cars where the electric motor completely replaces the internal combustion engine, resulting in zero CO₂ emissions, has made electric cars “the most promising solution to convert sustainable energy into drive energy, (Hofman and Dai, 2010)”. The technology relating to electric vehicles is rapidly changing due to large investment from governments and industry. Battery technology is constantly improving, reducing charging times and increasing power density (Electricvehicleresearch, 2015) along with electric motor efficiency improving (Petro, 2012). However, there are additional areas of electric vehicle design which require significant research to improve the vehicle performance and energy consumption.

Therefore, this project will investigate the use of advanced multiple-speed transmissions in electric vehicles as they are a key element in the vehicle drivetrain. Multiple-Speed transmissions increase the available wheel torque to reduce acceleration times and increase the achievable road grade, whilst prompting the power source to operate in a higher efficiency region during drive cycles. The research will look to see what type of transmission is required for a few case study electric axle drivetrains, be it a single-speed or multiple-speed transmission and quantify the differences. In addition, seamless gearshift
methods and control strategies for electric axles will be investigated to understand the impact on drivability.

1.2 SCOPE OF RESEARCH

The aim of this research is to ascertain the effect of adopting multiple-speed transmissions and multiple motor drivetrains for electric vehicles on energy consumption and vehicle performance. The development of gearshift controllers for seamless gearshifts to optimise drivability and optimal state selection/torque split algorithms.

Specifically, the technical objectives of the research are clearly defined below:

1) Compare the energy consumption over driving cycles and vehicle performance for different transmission layouts, specifically single-speed, two-speed and four-speed;

2) Design and implement an advanced state/torque split controller for the novel four-speed dual motor drivetrain;

3) Optimise drivability during gearshifts for multiple-speed transmissions for electric vehicles;

4) Develop an advanced anti-jerk controller for a Through-The-Road-Parallel (TTRP) Hybrid Electric Vehicle (HEV), including the two-speed transmission system discussed in 1);

5) Collaboratively build and commission the Hardware-in-the-Loop (HiL) electric drivetrain test rig at the University of Surrey.

The majority of work was conducted in simulation using the Matlab/Simulink environment as it affords the ability to model vehicle dynamics accurately. The development of a comprehensive vehicle model which could accurately simulate the characteristics and vehicle dynamics of an electric vehicle was the first major achievement of the research and allowed the completion of all objectives.

Referring to objective 1) single-speed and two-speed transmissions provided by the sponsoring companies were compared in simulation and physically on the HiL rig. A novel four-speed drivetrain was modelled and compared against optimised single-speed and two-speed transmissions. In both cases there was found to be a reduction in energy consumption over drive cycles and an increase in vehicle performance.

To allow a fair comparison during driving cycles for the novel dual motor-four-speed transmission an offline state and torque split optimisation procedure was developed. This allowed the transmission to select the optimum state and torque split for any achievable driving condition, thus realising objective 2).
The two-speed transmission utilised for the research was capable of achieving a seamless gearshifts due to the mechanical design. A comprehensive gearshift model was developed along with a controller to allow minimise the impact on drivability resulting in the completion of objective 3). Similarly a controller for the gearshifts of the four-speed dual-motor drivetrain was developed to minimise the torque gap for any state shift.

A TTRP HEV model was created using validated test data for each axle, from a test vehicle for the front internal combustion engine axle and from the HiL rig for the electric rear axle. A state space model of the system was built to permit the development of an anti-jerk controller to optimise the vehicles drivability during acceleration tests.

Furthermore, a major novel accomplishment of the research was the development of the Hardware-in-the-Loop test rig built at the University of Surrey. The test rig development initially required the management of the installation of the components on site and the commissioning, Human Machine Interface (HMI) and commissioning. Utilising the HiL performance tests and drivability tests were carried out along with drive cycles with the single-speed transmission. Gearshifts were successfully carried out with the two-speed which was a major accomplishment due to the difficulty that lay in controlling the change in inertias at the hubs. This resulted in all five objectives being completed.

Noise, Vibration and Harshness (NVH) was not considered part of the study.

1.3 OUTLINE OF THESIS

The thesis is laid out into six chapters, including the current chapter, where the current chapter contains a brief explanation into the motivation behind the research and the scope of work carried out.

The second chapter concerns a review of the state-of-the-art, and reviews the current literature pertaining to this field of research. Specifically the current technology relating to electric motor design, a review of transmissions designed for electric vehicles and gearshift methodologies and finally a review of gear and state selection techniques. Through reviewing relevant research up to the present time it was possible to confirm the gaps in knowledge and where further research can benefit the field of automotive engineering.

The third chapter explains the development of the vehicle models adopted for the research. An explanation of the eleven degree of freedom vehicle model is included, along with derivations of the governing equations. The models are based around the single-speed and two-speed transmissions adopted for the research and an explanation of the working principles is included. Preliminary results are presented, both for the performance of each vehicle and the energy consumption over standard driving cycles.

The fourth chapter explains the research carried out to model the gearshift of the two-Speed Electric Drive (2SED). The section describes the method utilised to model the gearshift dynamics, based on the three possible states. In addition, several gearshift control techniques are developed and the results presented.
Introduction: Outline of Thesis

The fifth chapter explains the development of the hardware-in-the-loop test rig at the University of Surrey. The layout of the test rig is described along with the components and simulation model adopted. The test rig results are included highlighting the test rig's ability to accurately simulate driving cycles and performance tests.

The final chapter includes research carried out on an additional multiple-speed transmission adopted for the research. An explanation of how the novel four-speed, dual-motor transmission was modelled is included along with the derivation of the transmission's governing equations. An investigation into the gearshift methodology with gearshift dynamics plots for the state changes is described as well as a description of the novel state selection optimisation procedure.

The report is finalised by a section with concluding remarks and an explanation of the suggested future work.
2 REVIEW OF THE STATE OF THE ART

2.1 INTRODUCTION

Since the introduction of the motor car, mechanical transmissions have been utilised to increase the torque produced by a power plant into a more usable wheel torque range. Manual transmissions using synchronisers and friction clutches with numerous gears are the standard solution in Europe to increase the operating range of the vehicle, the USA and Asia are still heavily reliant on automatic transmissions, Automated Manual Transmissions (AMT) and Continuously Variable Transmissions (CVT). Although various transmissions have been developed including automatic transmissions, CVTs and infinitely variable transmissions (dDrive, 2013). The majority of automotive transmissions have been developed for internal combustion engine vehicles, however as fully electric vehicles or parallel hybrid vehicles with a dedicated electric axle are now being utilised, transmissions specifically designed for electric motors are in demand.

Mechanical transmissions for fully electric vehicles are a relatively new area of research as the electric car industry has been primarily focused on the motor drive or battery technology. Generally, fully electric vehicles utilise single-speed transmissions (Tesla Motors, 2013, Sato et al., 2011, Greencarcongress, 2013), however, recent research suggests that the performance and efficiency of an electric drivetrain can be increased using a multiple-speed transmission (Sorniotti, 2010).

A transmission can be integrated into a fully electric vehicle architecture in various ways. Knödel et al. (2009) is particularly relevant to illustrate this and where the different vehicle layouts that can be adopted are shown in Figure 2-1. A traditional transaxle front-wheel-drive (FWD) layout with the drivetrain located at the front such as in (6) which is adopted in many ICE powered vehicles is the most conventional option, and this can be modified for all-wheel-drive (AWD) as in (5) through using a front to rear driveshaft and differential. Due to the reduced package size of an electric drivetrain compared to an ICE drivetrain two separate drivetrains can be installed on a vehicle as in (7), one located on the front axle and the second on the rear, giving rise to great advantages in driveability. Also illustrated in (1-4) are in-wheel motor (Cakir and Sabanovic, 2006) variants of a fully electric vehicle architecture, where a transmission can be installed between the motor and the hub.

Figure 2-1: Various fully electric vehicle layouts utilising transaxle or in-wheel motors. (Knödel et al., 2009)
2.2 ELECTRIC MOTORS FOR VEHICLE APPLICATIONS

Although there are many different variants of fully electric vehicle layouts, or even hybrid electric vehicles, the key component in each is the electric motor. Electric motor torque/power characteristics differ from an internal combustion engine, the main difference being that the generic electric motor can provide maximum torque from zero speed. The typical torque curve of an electric motor, as shown in Figure 2-2, is made up of a constant torque range from zero to base speed (Zeraolia, Benbouzid and Diallo, 2006), where the torque is electronically limited by the controller as the controller has a current limit. Theoretically the torque can be infinite at zero speed due to no back EMF (Electro-Magnetic Field) being present. The constant torque region is followed by a constant power region, in first approximation, where the available torque reduces (due to the increasing back EMF and by reducing the field flux of the motor) as the motor speed increases up to the maximum speed. Although in reality the torque reduces at a faster rate across the motors speed range reducing the power below a constant power.

![Figure 2-2: Typical torque/power characteristics on an electric motor. (Zeraolia, Benbouzid and Diallo, 2006)](image)

The main characteristics of an electric drive for a motor vehicle, as described in Chau, Chan and Chunhua (2008) should be a high torque density and power density, very wide speed range, high efficiency over wide torque and speed ranges, wide constant power operating capability, high reliability, robustness, low torque ripples and finally a low acoustic noise.

Knödel et al (2009) discusses the main options for the electric motor in an electric drivetrain which fall into two main categories, the brushed and the brushless. These two main groups of electric motors can be further categorised by subgroups as shown in Figure 2-3. The main designs of motor being considered for electric drivetrains are the direct current (DC) motor, the induction motor (IM), the permanent magnet (PM) synchronous motor (SM), and the switched reluctance motor (SRM).
2.3 MULTIPLE SPEED TRANSMISSIONS FOR ELECTRIC VEHICLES

A key area of this research is to understand if a multiple-speed transmission adopted for an electric drivetrain has energy consumption benefits over a single-speed equivalent (Bottiglione et al., 2014).

The torque characteristics of the electric motor lend well to a single-speed transmission due to a high torque region from rest providing enough torque for pull-off and inclines whilst the constant power region is extended for a large speed range, see Figure 2-4. However, a single-speed drivetrain performance is restricted by the application as a standard passenger car may require up to 5000 Nm of wheel torque and to provide this a large gear ratio will be required which would limit the top speed despite the motor having a wide speed range. This may be overcome by a high torque electric motor, but as the maximum torque is proportional to the size of the motor the motor would be voluminous, heavy, expensive and consequently has many disadvantages.

Knödel (2009) discusses the concept of a multiple-speed transmission for an electric drivetrain and points out the benefits over a single-speed transmission. Through utilising a
two-speed transmission the first gear ratio can be chosen to increase the low speed torque to improve acceleration and increase the achievable road grade, whereas the second gear ratio can be reduced to extend the operating vehicle speed range. The gear ratios need to be chosen to provide overlapping constant power regions to maintain acceptable drivability during gearshifts.

This is illustrated in Figure 2-4 where the available wheel torque is increased from \( \sim 400 \, \text{Nm} \) to \( \sim 700 \, \text{Nm} \) and whereas the maximum vehicle speed is 120 km/h in first gear the speed is increased above 140+ km/h in second gear greatly increasing the operating range of the vehicle. If a multiple speed transmission is adopted the maximum torque of the electric motor can be reduced as the wheel torque can be increased by increasing the first gear ratio. This results in further benefits such as reduced weight as the motor can be smaller and reduced cost.

The second major advantage from adopting a multiple speed transmission is that the drivetrain can theoretically operate in a higher efficiency region for a larger portion of a driving cycle. This is shown in Figure 2-5 where for a given vehicle speed and required wheel torque the motor would be operating in a low efficiency region in first gear whereas for the given second gear ratio the motor is operating in a higher efficiency region. The high efficiency region of an electric motor is generally in the low speed high torque region as opposed to the high speed low torque region, so if a low value of second gear is adopted the motor is forced into this region. Thus the gear ratios can be optimised to give the maximum drivetrain efficiency over standard driving cycles within a range that allows the effective field weakening region of the motor of each gear to overlap.

![Figure 2-5: Comparison of the operating point for a single and two-speed electric drivetrain (Gang: Gear; Guter Wirkungsgrad: Good efficiency; Schlechter Wirkungsgrad: Worse efficiency) (Knödel et al., 2009)](image_url)

Similar work was carried out by Rinderknecht and Meier (2010) where different electric drivetrain configurations were discussed along with the implications of installing a multiple speed transmission in a fully electric drivetrain. The author suggests that although an electric motor lends itself well to a vehicle application the achievable high efficiency region of the drivetrain can be increased. This is illustrated through Figure 2-6, where the arrow, (1), shows the operating point can be moved to a higher efficiency region if the next gear is of a smaller ratio and the same can take place in arrow, (2), if the ratio is bigger.
Review of the state of the art: Multiple speed transmissions for electric vehicles

The research carried out in Knödel et al. (2009) was progressed in Knödel et al. (2010), where single-speed and two-speed electric vehicles were compared in more detail. The two drivetrains had identical power, however the two vehicles differ as the single-speed drivetrain was powered by a high torque (450 Nm max torque) / low speed (4500 rpm max motor speed) and the two-speed drivetrain powered by a low torque (127 Nm max torque) / high speed (22,500 rpm max motor speed) motor. The space and weight saving achieved through adopting a smaller motor and a multiple speed transmission is also made evident through this paper as there is a weight saving of 51 kg between the two motors along with a 15 litre reduction in volume. The two motors are illustrated in Figure 2-7 below.

The energy efficiency benefit between the single and two-speed illustrated in Knödel (2009) is purely theoretical as it is only validated for a single operating point. In Knödel et al. (2010) a vehicle model was developed consisting of physical models of the components considering the moments of inertia, drag torques and efficiency maps which accounted for the variation on temperature (specifically the electric motor). Furthermore an optimisation of the gear ratios for both transmissions and shift points for the two-speed was undertaken, where a gear ratio of 3.3:1 was found for the single-speed and 19.5:1 for 1\textsuperscript{st} and 9:1 for 2\textsuperscript{nd} for the two-speed.
The model was used to run the New European Drive Cycle (NEDC) for each vehicle installed with the different drivetrains. The research found the two-speed drivetrain system to have an 18 % energy consumption improvement over the single-speed for the NEDC. The operating points during the NEDC are shown in Figure 2-8, where the effective high efficiency region is clearly larger for the two-speed system.

The two-speed vehicle evidently operates in the high efficiency region for a larger portion of the driving cycle giving rise to a reduction in energy consumption over the NEDC. Specifically, Knödel et al. found that on flat ground above 35 km/h the two-speed is saving energy being in second gear as opposed to first gear so spends the majority of the driving cycle in second gear. The authors also compared the two drivetrains with the same electric motor and found a 5-10 % energy consumption improvement in favour of the two-speed system.

Figure 2-8: Efficiency maps for the single-speed and two-speed drivetrains with operating points for the NEDC, the x-axis refers to the wheel speed (Knödel et al., 2010)

Furthermore, the performance of the two drivetrains was tested and the authors found a clear improvement over the single-speed for the two-speed system. The 0-100 km/h time for the single-speed vehicle was 8.4 s where as it was reduced to 7.4 s for the two-speed vehicle along with an improved top speed. This is due to the increased available wheel torque whilst in first gear and the extension of available torque across the vehicle speed range as shown in Figure 2-4.

The research presented by Knödel et al. analysed two different drivetrains, a single-speed drivetrain powered by a high torque low speed motor and a two-speed drivetrain with a low torque high speed motor. The benefits of the two-speed were clearly shown, however the paper lacked an explanation of the gear optimisation strategy and only considered one driving cycle so it is unknown if the benefit is true for all driving scenarios and moreover whilst maintaining practical performance requirements.
Antonov Plc have recently developed a three-speed transmission for an electric vehicle and published some of their research into the benefits of a multiple-speed transmission for an electric vehicle, Paul (2011). The presentation states that a multiple-speed transmission has benefits over a single-speed for an electric vehicle such as reduced powertrain weight as the motor can be downsized, improved drivetrain efficiency resulting in improved range, improved performance and reduced cost. As Antonov adopted a three-speed transmission the benefits of a three-speed over a two-speed were also discussed. Antonov suggest that a three-speed transmission in an FEV has increased vehicle launch capabilities with more available wheel torque and a further increased top speed, as shown in Figure 2-9.

![Figure 2-9: Comparison of achievable wheel torque for a 2-speed and 3-speed drivetrain (Paul, 2011)](image)

A larger number of gears also increase the possibility of the drivetrain operating in a high efficiency region reducing energy consumption. Paul, 2011 did an energy consumption comparison for single/two/three and four-speed transmissions using a drive cycle analysis tool. The software optimised the motor size and gear ratios for each case study vehicle and was capable of running drive cycles whilst considering regenerative braking. The results are presented in Figure 2-10 below, where the two-speed shows a considerable efficiency gain over the single-speed whilst only small further gains can be found for the three and four-speed drivetrains over the two-speed. The fact that Antonov chose a three-speed system was due to the ‘significant’ performance gains of the three-speed over the two-speed, although no quantitative evidence for this was published.

![Figure 2-10: Antonov drive cycle analysis tool results illustrating average efficiency over standard driving cycles for different transmissions in electric vehicle (Paul, 2011)](image)
Ren, Crolla and Morris (2009) carried out research in this field to analyse the effects of installing a gearbox in a fully electric drivetrain. They developed a simple backwards facing model in MATLAB to analyse the energy consumption over different drive cycles using a QSS toolkit.

Initially the NEDC was run for each of the five different transmissions considered, a single-speed transmission, a CVT and a two/three/four-speed transmission. The operating points of the single-speed transmission for the NEDC are shown in Figure 2-11 (a) below, where the efficiencies in positive torque are input power required/output power delivered and in negative torque it is power regenerated/input power.

The CVT drivetrain analysed used a simple optimisation system within the model which calculated the optimum ratio to be used for each required wheel torque and vehicle speed along the driving cycle to ascertain which ratio would allow the motor to operate in the highest efficiency region, however no transmission efficiency was considered. The CVT considered the physical constraints of the system and as such the gear ratios were limited between 0.6 and 4. The operating points for the CVT during the NEDC are shown in Figure 2-11 (b) and the gear ratios used during the cycle are illustrated in Figure 2-11 (c).

Ren, Crolla and Morris (2009) went on to analyse two-speed, three-speed and four-speed drivetrains with gear ratios selected based on the results of the CVT gear ratio optimisation. Consequently the two-speed had gear ratios of 2 and 0.8, the three-speed had gear ratios of 2, 1 and 0.8 and finally the four-speed transmission had ratios of 2.5, 1.5, 1 and 0.8. The model used a simple gear shift point selection system, where the shift point was purely defined by the vehicle speed.

The results of the single-speed case study, four-speed case study and the CVT for various driving cycles are shown in Table 2-1, including a column for percentage improvement over the single-speed for the four-speed and CVT drivetrains.
The results show a marked improvement over the single-speed drivetrain by using a multiple-speed transmission with energy consumption gains ranging from 4.5% to 11% over different driving cycles. It is evident that further gains can be made by adopting a CVT however the marginal gains over the four-speed transmission would in fact be lost in the additional transmission losses between the two systems.

Table 2-1: Comparison of improvements in energy consumption over 6 different driving cycles. (Ren, Crolla and Morris, 2009)

<table>
<thead>
<tr>
<th>Driving cycle</th>
<th>No gearbox</th>
<th>4 speed gearbox</th>
<th>Continuously variable gearbox</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Energy consumption per 100km (kWh/100km)</td>
<td>Energy consumption per 100km (kWh/100km)</td>
<td>Improvement %</td>
</tr>
<tr>
<td>Europe NEDC</td>
<td>8.33</td>
<td>7.96</td>
<td>4.5</td>
</tr>
<tr>
<td>Europe City</td>
<td>6.87</td>
<td>6.22</td>
<td>9.7</td>
</tr>
<tr>
<td>USA FTP-75</td>
<td>8.45</td>
<td>7.77</td>
<td>8.0</td>
</tr>
<tr>
<td>USA City I</td>
<td>9.06</td>
<td>8.43</td>
<td>7.0</td>
</tr>
<tr>
<td>Japan 11 mode</td>
<td>6.93</td>
<td>6.61</td>
<td>4.6</td>
</tr>
<tr>
<td>Japan 10 mode</td>
<td>7.20</td>
<td>6.41</td>
<td>11.0</td>
</tr>
</tbody>
</table>

The work by Ren, Crolla and Morris (2009) can be significantly improved as the model only took into account the efficiency of the motor, specifically a generic 40 kW motor (which used a non-experimentally attained efficiency map) and did not consider the efficiencies of the transmission/differential or the losses in the battery. The gear ratios of the multiple-speed transmissions were only chosen figuratively through the CVT results and no specific gear ratio optimisation procedure carried out. The shift points were only based on vehicle speed and not on driver demand and were not optimised either. Furthermore a time step of one second was used which significantly reduced the accuracy of the results. Ren, Crolla and Morris (2009) suggests that there may be an advantage to downsizing the electric motor when adopting a multiple-speed transmission which marries up with the work of Knödel (2010) and lends well to the idea of more research in this area being beneficial.

In contradiction to the above papers, Scharr et al. (2013) presented research on a transmission for an electric car at the 5th TM Symposium in China suggesting that a single-speed transmission is the optimum for a passenger car. The research focused on the development of a transmission for a fully electric small passenger car weighing 1250 kg, with a required axle torque of 1400 Nm, a top speed of 150 km/h and an acceleration time of less than 10 seconds from standing to 100 km/h. Two models were created to simulate driving cycles, one for both the single-speed and the two-speed with zero power loss during shift and a second for both drivetrains including power losses, extra mass and a shift strategy. The two-speed drivetrain was optimised, i.e. the gear ratio and shift points, although the method was not presented. The results of the simulations are shown below in Figure 2-12.

The results show there to be only a 1-3 % energy consumption benefit for the two-speed over the single-speed when no power losses are considered and this is reduced to less than 1-2 % for the more complex model. This does not tie up with the results of the aforementioned papers, and is most likely due to the authors not considering the efficiency.
of the electric motor as this is not mentioned in the paper. The efficiency of the motor tends to be significantly lower than the transmission for certain speeds and torques having a greater impact on the overall efficiency of the drivetrain and thus the energy consumption. The main highlight of the paper is that the efficiency of the transmissions were experimentally attained.

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The previously mentioned literature has suggested that the adoption of a multiple-speed transmission has potential benefits for a fully electric drivetrain. The literature has lacked in depth models considering all factors/efficiencies and optimising each parameter of the drivetrains. The papers all states that there is scope to further the research in this area, either through increasing the detail of the model or the development of more optimisation methods and control strategies.

2.4 TRANSMISSION LAYOUTS FOR FULLY ELECTRIC DRIVETRAINS

In this section, multiple-speed transmission technology up to the current state-of-the-art is reviewed including the transmissions being proposed for, or used in modern electric vehicles. Moreover, transmissions from petrol/diesel vehicles which could lend well to an electric drivetrain will also be considered.
Transmissions for automotive applications fall into four main categories; manual transmission, automatic transmission, CVT and Automated Manual Transmission (AMT). Each of these transmission typologies will be discussed in detail in the next section.

There have been many new developments in the transmission industry to improve the standard layshaft manual transmission (Müller, 1994), which utilises a clutch controlled by the driver to disengage the engine from the transmission during a gear shift. The gears are selected by moving a shift fork and consequently a synchroniser or dog clutch, however crash gearboxes which engage the gears with no synchronisation require the gear speeds to be matched during shifts. This standard manual transmission design has been in use for over a century but suffers from a torque gap during the time it takes for the driver to disengage/engage the clutch and also requires the driver to match the speed of the motor during the re-engagement of the clutch to maintain driveability.

The development of the automatic transmission (Heribert et al., 2008), revolutionised the transmission industry as it automated the gear shift process alleviating the need to manually change gear. The standard automatic transmission is comprised of a complex series of planetary gear sets, clutch packs and bands to change the gear ratio. In addition the automatic transmission has a torque converter to replace the friction clutch of the manual transmission and smooth the transfer of torque from the engine to the transmission during gear changes. The main drawback of the automatic transmission is the poor efficiency of the torque converter during gearshifts, although locking clutches are now adopted when the input and output speeds are equal.

A third genre of transmission that has been developed is the CVT (Ogata et al., 2012). The main benefit of the CVT is that it allows the input shaft to remain at a fixed velocity whilst the output shaft can rotate at any velocity (within range) allowing the power source (engine/electric motor) to remain at a high efficiency operating point. The drawbacks of the CVT are a low mechanical efficiency due to the belt or cone ‘slipping’ as the force is passed through the parts with friction and heat generation. An additional negative point relating to the CVT is that the subjective view of the driver during manoeuvres is distorted as the engine does not act directly in line with the drivers input reducing the drivers association with the vehicle. Although in theory the CVT is the perfect choice of gearbox as it allows the engine/motor to run at optimal efficiency points at all times, due the problems inherent in its design as stated above the industry is focused on developing the manual and automatic transmissions. A CVT has a few design variants, including the variable diameter pulley, toroidal CVT, hydrostatic CVT and the infinitely variable transmission (IVT). It should be noted that the recently designed infinitely variable transmission (IVT), (dDrive, 2013), does not use friction to transfer torque, instead adopting a complex series of gears to change the gear ratio, so the efficiency is much higher.

The latest transmission design is the automated manual transmission (AMT) [Taguchi et al, 2003], which adopts the layout of the modern synchronised manual transmission with shift forks and parallel shafts, whilst utilising the high efficiencies of the synchronisers and friction clutches. The transmission is automated through the use of electronic controllers.
and actuators to control the shift fork position and clutch force. The AMT is beneficial in that it is a cost effective way of having an automated transmissions as the cost of robotising a manual transmission is reduced compared to that of a genuine automatic transmission. The main drawback of the AMT is the torque gap which is still present and the difficulties in control, especially of the clutch force for each driving condition whilst taking into account clutch wear and temperature changes.

A modern evolution of the AMT is the DCT (Dual-Clutch Transmission), (Razzacki, 2009), which offers the seamless shift technology of an AT with only marginally lower overall efficiency than a MT due to the additional pump and churning losses. A DCT is essentially two manual transmissions joined together. The transmission is comprised of a quill shaft with the inner shaft being attached to one clutch and the outer shaft attached to a second clutch. Each primary shaft is attached to either the odd or even gears so whilst the current gears torque is being transferred through one primary shaft and the connected secondary shaft the next gear can be preselected on the other primary shaft. The second secondary shaft is spinning at a speed relative to the selected gears secondary shaft as they are both connected through the final gear, and consequently the preselected primary shaft and clutch is also spinning at a speed related to this. The gearshift can take place through lifting off the selected gear clutch and engaging the preselected gear clutch transferring torque from one input shaft to the other achieving a seamless gearshift. The DCT has many benefits such as a high efficiency, seamless gearshift however it is costly to manufacture. The 2SED transmission adopted in this research has the same performance as a DCT with significantly reduced mechanical, actuation and control complexity. A DCT schematic is shown in Figure 2-13.

Rhinderknecht and Meier (2011) included the results of a questionnaire taken at the 8th International CTI Symposium in Berlin in 2009 which asked the attendees about the future of transmissions for electric vehicles. The two questions posed were “which type of

![DCT Schematic](Qin et al., 2009)
transmission technology will we see in electrified drivetrains for the future?” and “Which transmission type will be the first to disappear from the market?” the results are shown in Figure 2-14 below.

The results show that the twin-stage (two-speed) transmission was thought to be the best transmission for a fully electric drivetrain, which is concurrent with the research from Paul (2011). Interestingly the CVT was considered the second best transmission for the electrified drivetrain, this is due to no clutch being necessary during launch and the authors felt the increased electric motor operating point efficiency would overcome the reduced transmission efficiency inherent with a CVT (as seen in Bottiglione et al., 2014). It may also be due to there being no engine sound from an electric motor so the downside of the driver having no link between the vehicle speed and engine speed is taken away. The results of the second question posed suggest that the standard AMT (a robotised MT) will be the first transmission to disappear from the market, due to cost and complexity whilst the torque gap still being present during a gearshift and the lack of user interaction. The second transmission thought most likely to disappear first is the CVT which is logical due to the major disadvantages such as low efficiency, NVH (noise, vibration and harshness) issues and high cost.

China is a prominent emerging automotive market and although there was not a similar questionnaire at the CTI Shanghai conference, the author of this thesis has experience in this subject as he currently works for a Chinese automotive manufacturer. To the purpose of this thesis, the author carried out an informal enquiry among the transmission managers of his company. Although AMTs are used extensively in China due to the low cost and automated shift, it is thought that this will be the first transmission type to disappear. The manual transmission is still the main transmission, and whilst the automatic transmission and DCT are key transmission types, the CVT is gaining credibility and will grow in market share. The CVT will also play a large role in the development of electric cars in the future along with the adoption of two and three-speed transmission. A large amount of electric vehicles will be built for city driving in China where single-speed transmissions will be utilized due to the low cost.

Getrag Innovations GmbH developed several single-speed transmissions for electric vehicles (Kohler and Knödel, 2008) showing the relevance of this transmission typology to
an electric drivetrain. The first is a single-speed transmission based on a coaxial transverse design using spur gears and a dog clutch to disengage the electric motor. It is useful to disengage the electric motor from the drivetrain to remove the inertia during coasting, Getrag Innovations GmbH also designed two different single-speed transmissions using a transaxle layout, the first using a layshaft design and the second a planetary gear set.

The transverse single-speed was later developed into a two-speed transmission through placing two bevel gears on the primary shaft with a dog clutch to select gears and disengage the electric machine. The transaxle using planetary gearsets has the potential to be developed into a two-speed design through modifying the planetary gear set. Both transmission concepts are shown in Figure 2-15 below.

![Figure 2-15: Schematics of the layshaft two-speed traverse transmission (left) and single-speed transaxle transmission using planetary gearsets (right) (Kohler and Knödel, 2008)](image)

The designs in (Kohler and Knödel, 2008) illustrate the simplicity of a two-speed transmission which previous authors have suggested would be sufficient for an electric drivetrain. The simple design reduces the cost of materials and manufacture and the developmental time for control strategies such as gearshift maps.

The standard form of AMT is one which uses a standard manual transmission and installs actuators on the shift forks and clutch. A good demonstration of this is presented in Taguchi et al. (2003). Here Taguchi et al. simply installed a DC ‘select motor’ to move the shift forks and a DC ‘shift motor’ to control the clutch, whilst an ECU was added to control the actuators movement. The ‘select motor’ incorporated a Proportional-Integral-Derivative (PID) to control the shift movements, achieving a 170 ms shift time once the clutch was disengaged with a 5 % tolerance. The ‘shift motor’ which controlled the clutch also utilised a PID but used a ‘torque assist’ mechanism which is essentially a spring attached to the bevel gear biased in the clutch release direction to improve responsiveness. In addition, the design included a ‘load control clutch cover’ which compensated for a worn clutch. It is the large shift time, which creates a torque gap and adversely affects the vehicle driveability which lends to this form of transmission being unpopular.

A possibility for an AMT developed purely for an electric drivetrain is to forego the use of a clutch completely, for launch and for the gearshift. This was realised in Risele and Bitsche (1995), where the two-speed transmission only incorporates two gears and one dog clutch as shown in Figure 2-16 below.
The gear shift takes place through rapidly reducing the motor torque to release the dog clutch and then the speed is matched with the next gear electronically. The shift is completed in less than 0.5 seconds but the transmission still suffers from a momentary torque gap which is completely overcome in the DCT design.

![Figure 2-16: Schematic of the two-speed gearbox and final drive (Risele and Bitsche, 1995)](image)

The DCT is much better suited to a vehicle drivetrain than an automatic transmission due to the high efficiencies of the helical gears and synchronisers whilst still eliminating the torque gap through the use of the two clutches. An early design of a DCT developed specifically for a fully electric vehicle was put forward by Koneda and Stockton (1985), it was labelled an AMT but due to it utilising two clutches it is more similar to a DCT. The paper details the design of a two-speed transmission built for an electric vehicle, planned to attain 30% gradeability, 0-50mph in less than 20 seconds and a top speed of 60mph by FORD Motor Company within a project for the United States of America (USA) Department of Energy. The design architecture of the drivetrain consists of the motor and transmission built concentrically around the drive axle.

The transmission has two planetary gear sets transmitting torque at all times, a one way clutch and a friction clutch as shown in Figure 2-17.

![Figure 2-17: Figure illustrating the design and layout of the DCT (Koneda and Stockton (1985))](image)
The two planetary gear sets are active all the time, where the output of the first gear set is the input to the second gear set. The input from the motor is through S2, see Figure 2-18, the overrunning one way clutch (OWC) prevents R1 from turning backwards and provides the reaction torque. The output sun gear carries, 6.37 times more torque than is inputted, then the ring gear R2 is also grounded and as the input pinions are stepped provides a further torque increase of 1.432 times. The regeneration takes place by locking CL1 and grounding R1 to prevent freewheeling.

The torque transfer in 2\textsuperscript{nd} gear is accomplished through applying the clutch CL2 and directly connecting the ring gear R1 to the output carrier C2, unloading the overrunning clutch. The simplicity of the gear shift, achieved only through the activation of one clutch is quite unique. Merely by applying the clutch CL1 the input ring reaction is removed by unloading the one way clutch, and the one way clutch releases as the torque is reversed. The shift only requires refined control of the one clutch to take torque away from the ring gear, so as not to be too quick for the change in torque to be perceived by the driver.

![Figure 2-18: Schematic of the DCT (Koneda and Stockton, 1985)](image-url)

The drivetrain has advantages such as being very compact through being built around the axle itself and utilising planetary gearsets. The design has several issues though, such as the weight distribution being quite poor due to the motor being on one axle side and the lack of travel for the axles due to the diameter of the motor and transmission making the drivetrain unsuitable for off-road vehicles.

The main problem with a standard AMT based on a layshaft design is that the torque gap during gear changes is still present from the standard manual transmission design. A second variation of a DCT for an electric vehicle is proposed in Kuroiwa et al. (2004), this design utilises a second clutch to increase shift quality. This transmission utilises a second clutch, as a ‘Torque Assist’, to apply torque to the wheels when the primary clutch is disengaged during a gearshift, as shown in Figure 2-19 below.
A shift takes place by transferring torque from the primary clutch to the torque assist clutch by reducing the pressure on the primary shaft and increasing the pressure on the assist clutch. When the torque being transferred by the primary clutch reaches zero and all torque is being transferred by the assist clutch through fifth gear the current gear can be disengaged. The next gear is then selected and once it is fully engaged the pressure on the primary clutch is increased until all the torque is passed through this clutch and the assist clutch is ‘open’.

Galvagno et al. (2011) published research on a similar powershift automated manual transmission which comprised a wet clutch which replaced the fifth gear synchronizer, named an assist clutch (ACL). The ACL reduced the torque gap inherent of a standard AMT by utilizing the wet clutch as a torque path during gearshifts. The authors proved the advantages in terms of gearshift quality and comfort.

The transmission presented in Kuroiwa et al. (2004) was developed to achieve the overall efficiency of a manual transmission but overcome the torque gap during shifts intrinsic in the manual transmission design whilst having the shift quality of an automatic transmission. The transmission was compared with an automatic transmission for six performance factors as shown in Figure 2-20 below.
The authors did not explain how each factor was measured, it may be purely subjective so not conclusive proof, but the results show that there is an improvement in efficiency of the AMT over the AT which would be expected. The design is simplistic in that a standard manual transmission can be modified to accept the assist-clutch system and the system appears to be successful in reducing the torque gap during gear shifts.

An early form of the modern DCT (Govindswamy, 2013; Remmlinger, Fischer and Patzner, 2008) was presented in a paper by Webster (1981). The transmission is based on a standard layshaft four-speed transverse transmission which achieved several goals laid out by the authors; have a similar efficiency and weight of a manual transmission and costs no more than a normal automatic transmission.

The intelligent design requires several steps to alter a standard manual transmission into the new dual clutch system. The evolution of the gearbox is shown in Figure 2-21 below.

Initially the second and third gears are swapped, so that one synchroniser can select first, third and reverse while the other synchroniser selects second and fourth, Fig. 1B in Figure 2-21. Then a second clutch is added at the rear of the gearbox with the input shaft of the transmission split into a quill shaft. The quill shaft is split so that first, third and reverse are driven by the primary clutch whilst second and fourth are driven by the rear clutch, Fig. 1C in Figure 2-21. Finally actuators are fitted to control the gear shift forks with an oil pump to supply the actuators and the wet clutch. With this layout a hot shift is simply performed by having the next gear preselected and by applying one clutch as releasing the other clutch so the torque to the wheels is not interrupted.
The paper goes on to suggest a two layshaft design as shown in Figure 2-22 which is applied to many modern manual transmissions (Jackson and Stanton, 2010). This utilises two layshafts with each synchroniser placed on a separate shaft and each shaft driving the final drive wheel. This design reduces the length of the transmission, allows more gears to be added per unit of space and is the layout adopted by modern DCTs. A higher number of gears is particularly beneficial as Webster (1981) suggests that with a four-speed design a gear ratio spread of 4:1 can be achieved however tests show that with ratio spreads of 8:1 or 10:1 gains of 25% efficiency can be made however no quantitative evidence is given.

Webster (1981) goes on to present four and six-speed gearboxes in conjunction with a fluid coupling and torque converter. However, for a fully electric vehicle application a torque converter is not necessary due to the availability of torque from zero motor speed.

Antonov (2011) developed a three-speed DCT purely for an electric vehicle, adopting a three gear design due to suggested increased performance and efficiency gains over a single or two-speed system. The primary shaft design consists of two clutches at the same end of the transmission attached to a quill shaft. First and third gears are attached to the inner shaft whilst the second gear is attached to the outer shaft as shown in Figure 2-23.
The design consists of several other features to increase the efficiency as illustrated in Figure 2-24, such as an offset output shaft to reduce the gear pairs. A synchroniser is added to second gear to reduce the drag whilst a low pressure mechanical oil pump is used to reduce parasitic losses and a high pressure electrical oil pump is used to feed further oil into the system. The dual clutch design allows the gear shifts to take place seamlessly through eliminating the torque gap and improving the driveability.

Antonov states that the transmission can remain in third gear during breaking, alleviating the need to downshift which increases driver comfort. Although downshifting can take place to select the first gear during specific braking scenarios to maximise the amount of energy recuperated through regenerative braking. Antonov developed an intelligent energy management system to maximise efficiency whilst the vehicle is being driven, which must essentially be a shift map. Furthermore they state that if the vehicle is linked to telematics to ascertain the road gradient the transmission can respond to stay in the most efficient gear when approaching a gradient (or hill).

The transmission itself is essentially a very efficient DCT which allows seamless gear changes and promotes excellent transmission efficiency and performance. However it is unknown why Antonov developed a three-speed instead of a two-speed. Antonov only states that it is for “significantly improved performance” over a two-speed as the energy consumption benefits are negligible, however no performance comparison data is given. The added complexity of a three-speed over a two-speed increases the cost and production time so it is questionable if it is the optimum solution. For instance the new BMW i3 just uses a single-speed transmission (BMW, 2015).

A modern version of the DCT was developed by LUK GmbH & co. (Berger, Meinhard and Bünnder, 2002) which was originally developed for racing in the 1980’s (Flegl, H et al. (1987)) and is the basis for most modern transverse DCTs. The DCT consists of a dual clutch pack at the input of the transmission with a quill shaft consisting of an inner and outer shaft on the same axis. The even gears attached are attached to the outer shaft and the odd gears attached to the inner shaft. A single secondary shaft connects to each primary shaft and the
final drive allowing for the transmission to be slim but lengthy and ideal for transaxle vehicles. The schematic of the transmission is shown in Figure 2-25 below.

![Schematic of the transmission](image)

Figure 2-25: Schematic of the PSG (Flegl, H et al., 1987)

In Flegl, H et al. (1987) the authors analyse the benefits of adopting dry clutches in a DCT. The paper compares the power losses of the PSG (which uses two dry clutches) against a manual transmission, an automatic transmission and a DKG dual clutch transmission which uses wet clutches. The results are shown in Figure 2-26, where the losses of both dual clutch transmissions are far lower than the automatic transmission, primarily due to the lack of a torque converter slip losses even though the clutch losses are slightly higher. The auxiliary losses are also reduced which is primarily due to the reduction in pump losses as no oil needs to be pumped through the clutch pack. The reduction is drag torque with the PSG over the wet DSG is due to the clutches spinning in air instead of oil reducing churning losses. The PSG with dry clutches is the optimal DCT as losses of 3-5% are saved compared to the wet clutch design making the overall losses similar to a standard manual transmission.

![Power losses in various transmissions](image)

Figure 2-26: Power losses in various transmissions, operating point is unspecified. (Flegl, H et al., 1987)
However, while the work of Flegl, H et al. (1987) illustrated in Figure 2-26 describes the contributing factors to losses for each transmission the research does not represent modern transmission systems. More recently, Vogelaar (2012) presented fuel efficiency comparisons for each transmission typology as illustrated in Figure 2-27 from the 2012 CTI Shanghai conference. The recent development in AT and DCT technology shows them to be more competitive against a manual transmission. The AT on average is still 10-20% worse than a manual, however a DCT is as good or even better than a MT at best. The comparison may be over the NEDC where the manual has a fixed gear schedule giving the MT some handicap.

An advanced seven-speed DCT developed by BMW (Munk and Klingermann, 2008) for internal combustion engine (ICE) driven vehicles is the state of the art for this transmission typology. The transmission is designed for ICE vehicles however the transmission would lend well to an electric drivetrain if transmissions with a higher number of gears and a higher ratio spread with lower ratio steps are required. The transmission is a seven-speed system which is capable of handling 600 Nm of input torque and 9000 rpm.

The DCT energy consumption and performance are compared against a manual transmission and an automatic transmission for two different vehicles, the BMW 335i passenger car and the high performance BMW M3. There is a significant reduction in the consumption of the DCT over the AT and MT for both vehicles as shown in Figure 2-28. The benefits can be said to also be relevant to an electric motor power source as the efficiency map of an electric motor varies in the same fashion as the BSFC (Brake Specific Fuel Consumption) map of an ICE. In addition there are performance gains to be seen through the adoption of the DCT over both the MT and AT. This is primarily due to the lack of torque gap during shifting which is present in the MT and the reduction in take-off losses which are due to the torque converter in the AT.
Munk and Klingermann (2008) compared the transmission against a base six-speed automatic transmission and found that the CO$_2$ reduction surpassed a target of 3% less than the 6-speed AT by a further 2%. Of course an electric drivetrain would not suffer from CO$_2$ emissions but this is a good reflection of the overall drivetrain efficiency. This was achieved with a sporty low gear ratio spread of 4.8, if the ratio spread is increased to 6.7 the reduction is increased to 6% which is in-line with the gains forecasted for the new generation of 8-speed automatic transmissions. The gains can be attributed to increased mechanical efficiency over the AT and the introduction of an intelligent operating system.

Figure 2-29 shows that further reductions can be found through electrification of the demand-orientated actuator system. The electrification of the actuators reduces the pump losses and if the pump used for lubrication and cooling was also electrified further losses would be recovered.
An original transmission concept for an electric drivetrain was proposed by Rinderkneckt, Meier and Fietzek, (2011) (and patented in Bologna, Everitt and Fracchia, 2011) which is a four-speed design utilising two motors. The schematic of which is shown in Figure 2-30. The transmission is characterised by an ‘odd’ electric machine, which is connected to the ‘odd’ primary shaft and, through a dog clutch, to either gear 1 or gear 3, and an ‘even’ electric machine, which is connected to the ‘even’ primary shaft and, through a dog clutch, to either gear 2 or gear 4. The gearshifts can take place through the control of the electric motor drive torques and the position of the electro-mechanical dog clutch actuators which drive barrel cams to select the gears. The high controllability inherent of electric motor drives permits the actuation of the gearshifts without the need for synchronisers, as the synchronisation is carried out electrically. This transmission can be coupled to a torque vectoring differential, and thus providing the energy efficiency benefit of a multiple-speed transmission and the vehicle dynamic performance of individual wheel powertrains. The dual-motor layout of this novel drivetrain concept provokes a high load factor (in the high torque region where the efficiency is higher) of the electric machines when they are operated singularly. This brings about a further potential increase of the overall drivetrain efficiency depending on the motor characteristics.

This transmission is used in the research of this thesis and will be covered in depth in later chapters.

Figure 2-30: Layout of a two-drive transmission based on AMT technology (Rinderkneckt, Meier and Fietzek, 2011)

The transmission which is used for the majority of the research in this thesis is a two-speed dual clutch transmission developed by Oerlikon Graziano and Vocis Drivelines (Cavallino, 2009). The novel two-speed transmission system combines the mechanical simplicity of a layshaft type transmission, with the high quality of a clutch-to-clutch gearshift. Its primary components are a one-way sprag clutch located on the secondary shaft and a friction clutch on the primary shaft together with an open differential, as displayed in Figure 2-31. The
input torque is transmitted by the sprag clutch while in first gear, and by the friction clutch while in second gear. The system can work either as a fully automated transmission or as an automated manual transmission through a seamless shift system. The friction clutch is applied to transfer torque from the sprag clutch during an upshift and is released to allow the sprag clutch to engage to accomplish a downshift.

Figure 2-31: Two-speed transmission illustration showing 2SED internals and schematic

A prototype of this transmission was supplied for the development of a hardware in the loop test rig and consequently was the focus of both simulation and physical testing. The transmission is covered in later chapters where performance and energy efficiency results are presented along with hardware-in-the-loop testing/validation results.

The CVT is another form of transmission which should be considered for an electric drivetrain. Through having the ability to smoothly transition between any number of gear ratios between its higher and lower limit it can in theory keep the drivetrain operating in the high efficiency area better than any multiple-speed transmission. Torque transfer is accomplished through various different methods such as a chain between two moveable gears, two helical gears which rotate to change the gear ratio and various other methods. However this transmission typology suffers from low efficiency due to the way the torque is transferred in the transmission, i.e. by pressure which generates slip and friction.

A CVT was adopted in the parallel HEV drivetrain of Debal et al. (2009) which consists of an ICE and an electric motor connected to the CVT which used a high volute chain to maintain a high efficiency, provide durability and produce low noise. The authors analysed whether the electric motor should input to the CVT input shaft or input to the CVT output shaft. If the electric motor was attached to the input it could be of smaller size but the authors found that if the electric motor was attached to the output of the CVT it resulted in higher efficiency gains, even though a larger more expensive motor is required. The total efficiency map of the system is shown in Figure 2-32 below, which is low in comparison to ICE drivetrains with a manual transmission.
A prototype electric two-wheel vehicle was fitted with a CVT in Carter, McDaniel and Vasilotis, (2007). The authors compared their old transmission which was a standard one-speed gearbox with the new prototype comprising of a CVT and controller (both drivetrains were fitted with a 1000 W motor). The version of CVT used is a continuously variable planetary transmission (CVP) which uses a bank of balls (or planets) placed between two discs. The torque is input through one disc with a helical design similar to the output disc which is connected to the output (wheel). Through changing the position of the planets between the two discs the effective radius is altered which changes the gear ratio. The transmission design is illustrated in Figure 2-33.
addition the range of the vehicle was improved by 20% for a city drive cycle and 7% for a drive cycle incorporating a large incline.

The light electric vehicle (LEV) tested in Carter, McDaniel and Vasilitis, (2007) shows the performance and efficiency gains possible through adopting a CVT over a single-speed transmission. However, as this is for a LEV and low torque levels were required the CVT did not suffer from the losses typical of a CVT in a high torque drivetrain due to slipping (in this case between the planets and the discs).

In Hofman and Dai (2010) several different transmissions were analysed for an electric vehicle to find which is most suitable. The author considered an identical passenger car fitted with the same 115 Nm and 10,000 rpm max electric motor in each case study vehicle. The transmissions analysed were a single-speed, shifting DC manual, an optimal manual (which did not account for shifting losses) and a CVT. A vehicle model of each case study was created to simulate drive cycles which included the different drivetrain torque balance equations and accounted for efficiencies, different part inertias and clutch losses for the MT.

Each case study vehicle was analysed for the NEDC and FTP75 (Federal Test Procedure 75). The transmission efficiencies were fixed at 0.95 for the final drive, 0.95 for the single-speed and MT whilst the CVT had an efficiency of 0.85. The results found that there was an energy saving of up to 6% for the MT over the single-speed whereas the CVT had an energy consumption increase of 8% for the NEDC and 10% for the FTP. The author went onto see if the energy consumption was affected by the transmission efficiency by using the same fixed efficiency value for each transmission and found that indeed it was.

The research in Hofman and Dai (2010) shows that the CVT is generally a poor choice if in real world applications it suffers from a low efficiency and as such a MT or even a single-speed is more beneficial in terms of energy consumption. However if all the transmissions have the same fixed efficiency an energy consumption reduction of 7% for the CVT can be found over a MT due to the motor operating in a higher efficiency region for a larger portion of a driving cycle. Nevertheless, the research is flawed and as it is only a mild representation, as through having a fixed efficiency value for the transmission efficiency an important aspect of the drivetrain is greatly simplified.

As a number of transmissions have been discussed in this section, it is necessary to briefly summarise each transmission to understand which are best suited for electric drivetrains and should be the focus of this research. Each transmission is marked from 1 to 5 for various attributes including “Relevance to EV” (1 is not relevant and 5 is most relevant), “Efficiency” (1 is a low efficiency transmission and 5 is a high efficiency transmission, “Performance” (1 is detrimental to the performance of the vehicle and 5 optimises the performance of the vehicle), “Cost” (1 is low cost and 5 is high cost), “Complexity” (1 is a simple design and 5 is a complex design) and “state-of-the-art” (1 is an old design and 5 is a modern state-of-the-art design).
The review of each transmission is purely qualitative as the performance and energy efficiency results would be required for an identical case study vehicle with an electric motor drivetrains. However the results give a good indication of which transmission type is best suited to a fully electric drivetrain. As can be seen in Figure 2-34 the highest scoring transmissions are predominantly the two-speed variants (Cavallino, 2009) based on a layshaft architecture as they have high efficiency due to using helical gears and friction/sprag clutches. Furthermore two-speeds provide adequate performance with a marked improvement over a single-speed whilst more gears provide little benefit for the additional cost and complexity. The four-speed two-motor design of Rinderknecht, Meier
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and Fietzek, (2011) scores highly due to the novel design suggesting significant energy consumption improvements whilst maintaining good performance.

During the course of this research several transmissions were adopted to analyse the benefits of each transmission typology to ascertain in more detail which is most beneficial to a fully electric vehicle. Therefore several of the transmissions mentioned in this section will be modelled to understand their performance and effects on energy consumption.

![Figure 2-34: Final results of the analysis of each transmission](image)

**2.5 MULTIPLE-SPEED TRANSMISSION GEARSHIFT METHODOLOGIES**

A multiple speed transmission in constant gear can be considered a simple one degree-of-freedom system (when not considering shaft torsion) and whilst the torque path through the transmission is dictated by the transmission design the wheel torque is solely affected by the input torque. The complexity comes when considering the gear shift, as depending on the transmission design a series of mechanical steps take place which can be difficult to model and control. The purpose of the gearshift is to switch between gear ratios to vary the ratio of input torque/speed to output torque/speed to keep the wheel torque within a usable range. This may be to increase the torque at the wheels to overcome a greater road gradient or to reduce the speed of the wheels relative to the engine speed increasing the attainable top speed.

A standard manual transmission drivetrain layout is illustrated in Figure 2-35 comprising an engine (or motor for an electric drivetrain), clutch, shafts, gears and synchronisers. In simple terms, a gearshift takes place through opening a clutch to remove the engine inertia from the transmission system and disengaging the current gears synchroniser and engaging the new gears synchroniser before closing the clutch again.
A synchroniser cannot engage nor disengage whilst a torque is acting upon it so the clutch must be opened to disconnect the input torque from the transmission and allow the synchronisers to move. Consequently the wheel torque is reduced to zero during the time the clutch is open, creating a torque gap. It is this torque gap which is undesirable from a drivability point of view as there is a large variation in wheel torque when the clutch is disengaged and reengaged. Therefore the aim of any gearshift methodology is to reduce the torque gap during the gearshift process, either through the transmission design or through gearshift control methodologies. In addition, for standard manual transmissions poor control of the clutch during the opening and closing phases can provoke driveline oscillations and lurches as the drivetrain torque can suffer from large variations. When the clutch is opened it creates oscillations in the transmission if the clutch is opened too quickly as the shafts tension under a driving torque is quickly released. The clutch closing phase is also particularly detrimental to the vehicles driveability if not properly controlled as poor delivery of the input torque can result in unwanted vehicle jerk (rate of vehicle acceleration). A slow clutch closing phase would result in a very smooth gearshift but would increase the shift time, alternatively a fast clutch closing phase would reduce the gearshift time but cause the wheels to spin or the engine to stall or lurch the vehicle forward.

Actual clutch control in real operating conditions must be based on the detailed experimental analysis of the actual behaviour of the system (Mattiazzo et al., 2002). The principles of clutch engagement control are summarised in Dolcini et al. (2010). Galvagno et al. (2011) discusses the clutch control phase for a dual clutch transmission system. Bemporad et al. (2001a) proposes advanced piecewise linear controllers, within an explicit model-predictive control framework, for clutch engagement control.

If we analyse the gearshift methodology of a manual transmission in more detail we can suppose it to be made up of five stages; engaged, slipping open, synchronisation, go-to-slip and slipping close as described in Glielmo et al. (2006). The ‘engaged’ phase takes place before the gearshift request takes place, the clutch is locked and the transmission is in constant gear. The ‘slipping open’ stage is while the clutch is opened by moving the throw out bearing and the transmission input torque is reduced relating to the clutch position
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until the clutch is fully open and the input torque is zero but the engine still rotates at a similar speed to the transmission due to the inertia of the engine/motor. The ‘synchronisation’ phase covers the period where one synchroniser is disengaged from the current gear by moving a shift fork, the new gear is engaged by engaging the relevant synchroniser and the primary shaft/clutch speed changes. The ‘go-to-slip’ stage refers to the point in the gearshift where the synchroniser has fully engaged and the clutch is re-engaging, however this stage only covers up to the point where the clutch is touching the flywheel or the ‘biting point’. The final phase is the ‘slipping close’ stage where the clutch is gradually engaged and the input torque to the transmission increases relative to the throw out bearing position due to the increasing axial clutch force. The ‘slipping close’ phase is completed when the ‘engaged’ phase criteria are met. Each stage is illustrated in Figure 2-36.

![Figure 2-36: Engine speed (solid line) and clutch speed (dashed line) signals during a gearshift; the five operating phases are highlighted. (Glielmo et al., 2006)](image)

For a manual transmission the input torque, gear shift position and clutch position are controlled by the driver and therefore the vehicle response, shift time, torque gap and drivability are purely a consequence of the driver input. The development of automated manual transmissions and DCTS’s which automate the gearshift process require the careful control of the input torque/speed, clutch position and synchronisers during the gearshift to minimise the shift time and torque gap to maximise drivability. In particular, clutch control is important during the clutch re-engagement phase where poor control of the clutch
position can result in excessive clutch wear and unwanted vehicle jerk as explained in Lei, Niu and Ge (2000).

The majority of automated gearshift/clutch control research is based on standard manual transmissions, however various transmissions have been developed using novel mechanical gearshift techniques to remove the torque gap. The automated manual transmission presented in Kuroiwa et al. (2004) uses a second clutch to provide wheel torque while the main clutch is disengaged. This requires several steps, where initially the assist clutch is engaged proportionally to how much the primary clutch is disengaged until all the torque (equal to the oncoming gears torque) is being transferred by the assist clutch. As no torque is being passed through the primary shafts now, the synchronisers are free to move and the next gear can be selected. Finally the assist clutch is disengaged as the primary clutch is engaged at a rate at which the total wheel torque remains constant. The gearshift methodology is graphically illustrated in Figure 2-37.

![Gearshift methodology](image)

The transmission developed by Kuroiwa et al. (2004) is essentially a simplified version of the modern DCT which has a similar gearshift methodology. However, a DCT has an on-going and off-going clutch so only half the steps required by the Kuroiwa et al. (2004) transmission.

A DCT gearshift is fully automated as the two clutches are not directly controlled by the driver, so the clutches and engine control needs to be very robust and have the shift time and driveability tuned to suit the application. A DCT gearshift is illustrated in Figure 2-38.
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Figure 2-38: DCT upshift from first to second gear (Goetz, Levesley and Crolla, 2005)

The DCT gearshift is comprised of two phases, the torque phase and inertia phase. For an upshift the torque phase takes place first where the offgoing clutch is initially released until is close to slipping (step 1). The oncoming clutch is then applied to prefill the torque demand (step 2). The oncoming clutch is then applied at a steady rate (step 4) and the offgoing clutch is carefully controlled until it is fully open (step 3). The inertia phase is then initiated where the engine/motor speed is reduced to the new gears speed whilst the oncoming clutch is still slipping (step 5/6). When the engine/motor speed has matched the required speed the oncoming clutch is fully closed and the gearshift is complete (step 7).

As can be seen from Figure 2-38 the DCT design and gearshift methodology allows a seamless gearshift to take place. The top graph shows the wheel torque, which remains fairly constant during the shift, albeit for some brief oscillations most likely due to torsion in the drive shafts. The gearshift does have several steps which need to be carefully controlled and with shift times of 0.2 seconds easily achievable the reaction times of the actuators need to be very fast whilst the control needs to be robust.

Each automated transmission, whether an AMT or DCT, contains a clutch which needs to be controlled to maintain a desirable vehicle performance and acceptable vehicle drivability during a gearshift. Therefore, it is necessary to understand the state of the art clutch control methods to accurately assess the performance of each automated transmission. Therefore various state-of-the-art automated clutch control methodologies will be briefly reviewed in the next section.

Several control methods have been proposed for friction clutch engagement control within AMTs during launch or during the gearshift. Stage orientated PI controllers and estimators were utilised in Glielmo (2006), an observer-based optimal control method was proposed in Dolcini, Béchart and Canudas de Wit (2005), quantitative feedback theory in Sliker and Loh (1996), model predictive control in Lu et al. (2011), fuzzy control in Tanaka and Wada
Review of the state of the art: Multiple-speed transmission gearshift methodologies


In Glielmo et al. (2006) a dry clutch control methodology of gearshifts for an AMT is explained which is primarily based on a series of decoupled speed and torque control loops. Glielmo et al. split the gearshift up into five phases (explained previously) and developed control strategies for each phase. The authors developed a model of the driveline comprising the characteristics of the physical components including the actuators which were validated against experimental data.

The simple automated transmission design adopted for the development of the control system is illustrated in Figure 2-39.

These equations are the basis of the driveline model (although the secondary shaft and driveshaft equations are omitted here) and are adopted for each of the control phases. The first equation is the torque balance equation of the motor and clutch where the clutch is in a slipping condition.

\[ J_e \dot{\omega}_e = T_e - T_c(x_c) \]  

\( J_e \) is the engine inertia, \( T_e \) is the engine torque, \( \dot{\omega}_e \) is the engine acceleration, \( T_c \) is the clutch torque and \( x_c \) is the clutch displacement. The primary shaft equation is then given in (2).

\[ [J_c + J_{eq}(i_g i_d)] \ddot{\omega}_c = T_c(x_c) - \frac{1}{i_g i_d} k_{tw} \Delta \theta_{cw} + \beta_{tw} \left( \omega_c \frac{i_g i_d}{i_t} - \omega_w \right) \]  

\( J_c \) is the moment of inertia of the clutch, \( i_g \) is the selected gear ratio, \( i_d \) is the differential gear ratio, \( \dot{\omega}_c \) is the clutch acceleration, \( k_{tw} \) is the driveshaft stiffness, \( \Delta \theta_{cw} \) is the difference in angular displacement of the differential and wheel, \( \beta_{tw} \) is the driveshaft damping ratio, \( \omega_c \) is the clutch rotational velocity and \( \omega_w \) is the wheel rotational velocity. \( J_{eq} \) is the equivalent moment of inertia of the system, given by.
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\[ J_{eq}(i_g i_d) = J_m + \left( \frac{1}{i_g^2} \right) \left( J_{s1} + J_{s2} + \frac{J_d}{i_g^2} \right) \]  \hspace{1cm} (3)

where, \( J_m \) is the moment of inertia of the main gear, \( J_{s1} \) is the moment of inertia of the primary shaft, \( J_{s2} \) is the moment of inertia of the secondary shaft and \( J_d \) is the moment of inertia of the differential.

The clutch engagement phase is the critical part of the gearshift as poor control results in mechanical oscillations and vehicle lurch. Glielmo et al. showed that the engagement smoothness is related to the clutch slip acceleration at lock up and the derivation is shown below.

Glielmo et al. (2006) assume that two systems exist, one where the clutch and engine speeds match and one where they are different before and after time \( \bar{t} \), where \( \bar{t}^- \) and \( \bar{t}^+ \) are the moments instantly before and after clutch engagement. Then through modelling the discontinuity \( \left( \dot{\omega}_c(\bar{t}^+) - \dot{\omega}_c(\bar{t}^-) \right) \) whilst assuming that the motor torque is constant and the engine and clutch speeds match after lock up the following derivation can be made. \( J_e \) is the moment of inertia of the engine.

From (1) and (2),

\[ \dot{\omega}_c(\bar{t}^+) = \frac{1}{J_e + J_c} \left[ T_e(\bar{t}^+) - \Omega(\bar{t}^+) \right] \]  \hspace{1cm} (4)

Where,

\[ \Omega = \frac{1}{i_g l_d} \left[ k_{tw} \Delta \theta_{cw} + \beta_{tw} \left( \frac{\omega_c}{i_g l_d} - \omega_w \right) \right] \]  \hspace{1cm} (5)

and by assuming that the state and engine torque is continuous at lock up,

\[ \dot{\omega}_c(\bar{t}^+) - \dot{\omega}_c(\bar{t}^-) = \frac{J_e}{J_e + J_c} \dot{\omega}_c(\bar{t}^-) \]  \hspace{1cm} (6)

Thus the discontinuity is shown to be affected by the clutch slip acceleration, \( \dot{\omega}_{si} \), and this must be properly controlled during the clutch closing phase. However the equations are not fully derived in the paper with the authors omitting steps in the derivation and inertial terms. During the slipping-closing phase a controller which uses clutch and engine reference speeds outputs the reference clutch throw-out bearing position and engine torque, the controller is illustrated in Figure 2-40. The authors do not explain the origin of the engine and clutch speeds which one could assume would come from the driving cycle reference speed but working on a real world vehicle would come from the ECU and wheel speed sensors.

The four controllers \( C_{1-4} \) are PI controllers, where \( C_1 \) is a feedforward controller and \( C_2 \) and \( C_3 \) are simple PI single-input single-output linear controllers. The engine torque reference is
calculated by comparing the output of \( C_1 \) and \( C_2 \) with the clutch torque estimate output calculated from the engine speed and estimated engine torque.

The slipping-opening controller is an open loop system where the clutch opens at a fixed rate. The synchronisation phase takes place when the engine torque is zero so the reference clutch speed is purely dictated by the previous and oncoming gear ratios. The go-to slipping phase calculates the reference throwout bearing position before torque is transmitted through the clutch torque estimator and averaging past bearing positions where torque started being transmitted.

![Block diagram of the gearshift controller during the slipping-closing phase](Glielmo et al., 2006)

The authors, Glielmo et al. (2006), present graphs of the engine torque/speed and clutch position for several upshifts and down shifts showing how the control system reacts for each manoeuvre. However, no graphs showing the resulting wheel torque, vehicle acceleration or jerk and presented so the quality of the gearshifts from a driveability point of view cannot be evaluated.

An observer-based optimal control method was proposed in Dolcini, Béchart and Canudas de Wit (2005). The optimal control method was adopted with prescribed final states and the clutch torque as the controlled input and the engine torque as the known controlled input. Here the authors focused on the dry clutch control from a standing start, however the clutch control theory can be applied to a gearshift as well as this situation would not arise with an electric drivetrain.

A simple drivetrain was initially assumed as a four state linear system whose dynamic equations are given below.

\[
J_e \dot{\omega}_e = \Gamma_e - \gamma F_N \tag{7} \\
J_g \dot{\omega}_g = \gamma F_N - k_t \theta - \beta_t (\omega_g - \omega_v) \tag{8}
\]
\[
j_\nu \dot{\omega}_v = k_t \theta + \beta_t (\omega_g - \omega_v) \quad (9)
\]
\[
\dot{\theta}_g = \omega_g - \omega_v \quad (10)
\]

where \( \Gamma_e \) is the engine torque, \( F_N \) is the normal force on the clutch friction discs, \( J_g \) is the gearbox moment of inertia, \( \dot{\omega}_g \) is the gearbox rotational acceleration, \( k_t \) is the transmission stiffness, \( \dot{\theta}_g \) is the transmission torsion, \( \beta_t \) is the transmission damping coefficient, \( \omega_g \) is the gearbox rotational velocity, \( \omega_v \) is the equivalent vehicle speed, \( J_v \) is the equivalent vehicle moment of inertia, \( \dot{\omega}_v \) is the vehicle acceleration and \( \gamma = \mu_d R_d \) where \( \mu_d \) is the coefficient of friction and \( R_d \) is the clutch radius.

The aim of the controller as described by the authors is for the engine speed to match the gearbox speed at the point of engagement, \( \omega_e = \omega_g = \omega_v \), whilst reducing any wasted energy. The authors state that to reduce unwanted driveline oscillations and satisfy a no lurch condition the clutch torque must match the engine torque minus the engine inertia reaction torque.

Through defining \( z_1 = \omega_e - \omega_g \) and \( z_2 = \omega_g - \omega_v \) the final state of the system should be that \( z_1 \) and \( z_2 \) both equal zero. Thus equations for \( z_1 \) and \( z_2 \) are defined below where \( \Gamma_c \) is the clutch torque:

\[
\dot{z}_1 = \frac{\beta_t}{J_g} z_2 + \frac{k_t}{J_g} \dot{\theta}_g + \frac{1}{J_v} \Gamma_e - \frac{1}{J_{\ell_1}} \Gamma_c \quad (11)
\]
\[
\dot{z}_2 = -\frac{\beta_t}{J_{\ell_2}} z_2 - \frac{k_t}{J_{\ell_2}} \dot{\theta}_g + \frac{1}{J_g} \Gamma_c \quad (12)
\]
\[
\dot{\theta}_g = z_2 \quad (13)
\]

When adopting optimal control the aim is to find \( u(t) \) on \( T=[t_0, t_f] \) which minimises the following quadratic value function.

\[
j = \frac{1}{2} \int_{t_0}^{t_f} [z^T Q z + u^T R u] dt \quad (14)
\]

Under the constraint,

\[
\dot{z} = A_z z + B_{z1} \Gamma_e + B_{z2} u \quad (15)
\]

The matrices \( A_z, B_{z1}, B_{z2}, Q \) and \( R \) are formed from equations (11), (12) and (13), which leads to a Two Point Boundary Value Problem (TPBVP) to solve for \( u \).

The final controller is illustrated in Figure 2-41 below. The “Unperturbed linear model” is the state space model of the drivetrain and the ‘Optimal control’ generates a target optimal trajectory \( y^* \) in the \( z \) state space and the optimal clutch torque. As the optimal control system is sensitive to perturbations a ‘Trajectory tracking’ system was adopted to ensure clutch engagement despite engine torque oscillations.
Finally a multiple-input multiple-output (MIMO) linear time invariant (LTV) observer was developed to estimate the clutch friction co-efficient. This is due to the control system only knowing the clutch position, but to estimate the clutch torque the friction co-efficient must be known.

The effect of the MIMO-LTV observer to estimate the friction co-efficient can be seen in Figure 2-42 where the engine and clutch speeds are given for a take-off manoeuvre. The controller with a 100% error in the coefficient of friction results in twice the required torque being delivered to the drivetrain giving rise to large drivetrain oscillations. However, when there is no error or even a small 20% error there are no oscillations and the system behaves as expected.
It should be noted that the authors claim the vehicle model to be extremely complex and based on a real world vehicle however one would expect more high frequency oscillations in the drivetrain which would appear in the engine speeds presented in Figure 2-42.

In Haj-Fraj and Pfeiffer (2002) a non-linear offline method is adopted to optimise the gearshift considering two sets of cost functions. The first focusing on performance parameters such as vehicle acceleration and jerk whilst the second attempts to prolong the life of the transmission.

The authors developed a drivetrain model utilising an engine based on a rotating rigid body neglecting high frequency oscillation where the torque output is calculated from preset look-up tables. The simplified transmission is a two-speed system consisting of a wet clutch and a one-way clutch where an upshift takes place by applying pressure to the friction clutch to take off torque from the one-way clutch. A torque converter was added where the output torque is found from a lookup table in a similar way to the engine torque. Furthermore the model considered the elasticities of the tyres and shafts as torsional elements. The resistive forces affecting the vehicle such as the aerodynamic, rolling resistance and road inclination forces were also considered. The model was validated against experimental results.

The authors defined two parameters to be controlled by the engine control unit (ECU), the clutch pressure, $p_i$, and the engine load reduction, $\beta_i$. The two parameters are then optimised for any turbine torque and speed and are represented as look-up tables for the ECU to use. Therefore the optimisation parameter vector can be expressed as:

$$\bar{p} = (p_1, \ldots, p_n, \beta_1, \ldots, \beta_n)^T$$

(16)

The parameter $\bar{p}$ is then optimised for the different cost functions including weighting factors for each cost function. The first cost function is based on minimising the vehicle jerk during the gearshift, $G_1$. The second is based on a reference vehicle acceleration profile for the gearshift where the difference between the desired and actual acceleration defines the cost function, $G_2$. The third is a more complex cost function which aims to reduce the peak-
Review of the state of the art: Multiple-speed transmission gearshift methodologies

to-peak values of acceleration, G₃. The fourth and fifth are focused on maintaining the shift elements where specifically the fourth calculates the frictional losses, G₄, and the fifth simply calculates the shift time, G₅.

The authors presented results when considering only two elements in the cost function to reduce the computational requirement. Figure 2-44 shows that when aiming to reduce the shift time and increase driveability the shift time was reduced successfully and the peak vehicle jerk is reduced.

![Figure 2-44: Comparison of optimization results using the criteria (G₁,G₅), 100% load (Haj-Fraj and Pfeiffer, 2002)](image)

When considering the alternative cost functions, G₄ and G₅, the gearshift duration and frictional losses were also improved. However, the authors found the optimal weighting factors to be conflicting between reducing the vehicle jerk and reducing the frictional losses.

Haj-Fraj and Pfeiffer (2002) successfully developed a method for optimising the gearshift to reduce vehicle jerk and frictional losses. However, the method was not fully explained as the rate of engine load reduction is not defined and the results are only proved for a singular manoeuvre.

Van Der Heijden et al. (2007) consider two optimal control strategies to automate dry clutch control during engagement, model predictive control (MPC) and a piecewise linear quadratic controller. The model predictive controller builds on the work of Bemporad et al. (2001a,b) which consisted of an offline method but had limitations as it did not consider driveline dynamics.

The work by Van Der Heijden et al. (2007) used a simple drivetrain model consisting of an engine, clutch, gearbox, differential and driveshafts. The model is based on two sets of equations, one for when the clutch is slipping and a second for when the clutch is sticking. The clutch was modelled as a Coulomb friction model where the clutch torque, Tᵣ, is calculated as:

\[ Tᵣ = F_n \mu R_a \text{sign}(\omega_e - \omega_c) \]  \hspace{1cm} (17)

where \( F_n \) is the normal clutch force, \( \mu \) is the coefficient of friction and \( R_a \) is the clutch radius.
The model predictive control theory adopted is based on the standard strategy with a linear performance index as adopted in Kvasnica et al. (2004). The aim of model predictive control is to predict the future of a system using the model of the plant and to optimise specific performance parameters under pre-set constraints. The control problem is formulated as:

$$
\min_p \left\| P_N (x(N) - x_{ref}) \right\|_\infty + \sum_{k=0}^{N-1} \left\| Q(x(k) - x_{ref}) \right\|_\infty + \left\| R \Delta u(k) \right\|_\infty
$$

subject to:

$$
x(k+1) = A_{d,i} x(k) + B_{d,i}(k)
$$

with constraints:

$$
u_{min} \leq u(k) \leq u_{max}
$$

$$
\Delta u_{min} \leq u(k) - u(k-1) \leq \Delta u_{max}
$$

$$
x_{min} \leq u(k) \leq x_{max}
$$

The variable $N$ defines the number of steps ahead the model predicts the outcome of the system and is a variable to be minimised to reduce computation time. The matrices $Q$ and $R$ need to be tuned to control the states and inputs respectively. The authors found that due to the complexity of the system there was not enough time to do an online optimisation to solve the above function. Instead, they adopted the method developed in Bemporad (2000) which allows for the optimal control to be tuned offline. The optimal control problem is then solved parametrically which creates a large number of piecewise affine (PWA) control laws that can be solved as simple linear functions online with a vastly reduced computational time.

The state and input variables selected from the drivetrain model are given below:

$$
x = \begin{bmatrix} \omega_e \\ (\omega_e - \omega_c) \\ (\omega_c - \omega_w) \\ i_{tot} - \omega_w \\ T_s \end{bmatrix}, \quad u = \begin{bmatrix} T_c \\ T_e \end{bmatrix}
$$

$i_{tot}$ is the total drivetrain gear ratio and $T_s$ is the wheel torque.

The state space model takes the generic form of:

$$
\dot{x} = \begin{cases} 
A_1 x + B_1 u & \text{if } x_1 > \eta \\
A_2 x + B_2 u & \text{if } x_2 < \eta
\end{cases}
$$

$A_{1/2}$ and $B_{1/2}$ are matrixes, $x$ is the state vector and $u$ is the control vector. The subscript 1 relates to the clutch slipping mode and the subscript 2 relates to the clutch sticking mode where $\eta$ is a variable that defines which mode the system is in.
The results of the optimisation gave the matrices $Q$ and $R$ which are shown below. These were the result of the MPC tuning for a horizon length of 2 as the authors found that this gave the same results for a horizon length of 50. The low horizon length is beneficial as whilst a high horizon length theoretically improves the performance the number of constraints and control laws increases increasing the computation time.

$$N = 2, \quad Q = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 300 & 0 \\ 0 & 0 & 0 & 0.1 \end{bmatrix}, \quad R = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}$$  \tag{25}

The results for a take-off simulation shown in Figure 2-45 illustrate the response of the system. The graphs show that the ‘active controller’ is controlling the system until the clutch lock-up takes place at 0.8 seconds and the engine torque demand rises above the clutch torque. The lack of oscillations present in the wheel torque is evidence that the MPC controller successfully manages the clutch engagement (although it suggests a simple vehicle model). However the large computational demands make it incompatible for this application at the moment and the computation could not be done in real time on a vehicle.

![Figure 2-45: Launch action with the MPC controller: (a) engine and clutch speeds; (b) drive shaft torque; (c) clutch and engine torque; (d) active controller regions. (Van Der Heijden et al., 2007)](image)

In Van Der Heijden et al. (2007) the authors go on to develop a piecewise linear quadratic controller (PWLQ) which is similar to the work of Dolcini, Béchart and Canudas de Wit (2005). The authors found that the computational time was reduced compared to the MPC method with satisfactory results. However, the performance of a PWLQ controller is heavily dependent on reference trajectories which must be pre-tuned requiring a lot of offline work. In addition, the PWLQ uses inputs that are not available on a production car so a comprehensive and accurate state estimation model must be developed which is not presented by Van Der Heijden et al. (2007).

Other methods have been adopted to automate the clutch and engine control during vehicle take-off and gearshifts. Sadati and Talasaz (2004) propose using sliding mode
control for a theoretical flexible transmission system. Sliding mode control is a control method designed to keep a system on an abstract sliding surface, such as a position or state. A sliding mode controller is a robust control algorithm for non-linear systems however it is susceptible to “chattering”, where the system can rapidly switch from each “side” of the plane creating oscillations in the system. Although several papers including Slotine and Li (1991) and Kaynak, Erbatur and Ertugrul (2001) propose methods to reduce the chattering effect. Sadati and Talasaz (2004) show that sliding mode is an effective tool for controlling the position of a pulley when applied to disturbances including low frequency oscillations which is similar to a clutch application.

Sliding mode control was adopted for dry clutch engagement successfully in Prabel et al. (2011) to control the feedback of the clutch position. Similarly in Song and Sun (2012) sliding mode control is used to control the clutch pressure following a reference clutch pressure.

Another dry clutch control method is proposed in Tanaka and Wada (2007) using fuzzy control. The controller adjusts the clutch displacement parameter, \( D_Y_E \), according to the output of the fuzzy control and the pre-determined engagement start point (biting point), \( Y_s \) (memorised from when the clutch starts to rotate). The inputs to the controller are the accelerator pedal position, \( U_E \), plus the rate of depression, \( DU \), and the engine/clutch rotational velocity, \( N_E/N_C \) plus their respective changes, \( DN_E/DN_C \).

The fuzzy control is based on the four rules given below, and are represented graphically in Figure 2-46. The controller analyses the value of \( U_E \) and \( DU \) to ascertain which rule is accepted to determine which membership function should be used to calculate \( GDY_E \) and find the clutch displacement parameter. However, the defuzzification method is not stated.

The rules are expressed as:

\[
\begin{align*}
R11: & \quad \text{If } DU = DU_0 \text{ or } U_E = U_{E0} \text{ Then } D_Y_E = D_Y_{E0} \\
R12: & \quad \text{If } DU = DU_S \text{ or } U_E = U_{ES} \text{ Then } D_Y_E = D_Y_{ES} \\
R13: & \quad \text{If } DU = DU_M \text{ or } U_E = U_{EM} \text{ Then } D_Y_E = D_Y_{EM} \\
R14: & \quad \text{If } DU = DU_L \text{ or } U_E = U_{EL} \text{ Then } D_Y_E = D_Y_{EL}
\end{align*}
\]
The final function is then given as:

\[ DY_R = \frac{[A_0 + A_S + A_M + A_L]}{[GDU + GD_E]} \]  
(27)

The \( A_k \) terms are:

\[ A_0 = (GDU_0 + GU_{EO})DY_{EO} \]
\[ A_S = (GDU_S + GU_{ES})DY_{ES} \]  
(28)
\[ A_M = (GDU_M + GU_{EM})DY_{EM} \]
\[ A_L = (GDU_L + GU_{EL})DY_{EL} \]

The variables \( DY_{E0/E5/M/L} \) determine the proportion of the full stroke as 0, 30%, 60% and 100% respectively.

The authors built a hydrostatic test rig comprised of a pump to deliver a torque curve depending on throttle position (i.e. to simulate the engine), a transmission, sensors for engine and clutch speed and a resistive pump to simulate vehicle load. The results of the simulation show the system to react well even for an uphill start, compensating for alterations in load. The system is robust and can be easily implemented to reduce computational load for non-linear systems and lends well to clutch control. However, fuzzy logic requires a lot of time to tune the membership functions offline from experimentally attained test data and requires further testing to ensure the functions work effectively.

2.6 GEAR/STATE SHIFT CONTROL

An important aspect of multiple-speed transmission control concerns the management of gear-shift maps or schedules. When developing the control of gearshift points for a multiple-speed transmission two factors must be considered, namely the vehicles performance and economy. Consequently, the “optimisation” of the gearshift maps for different vehicles may have different connotations as a high performance vehicle would sacrifice fuel/energy consumption in favour of low acceleration times and a high top-speed. Whereas a family car would look to maximise the fuel economy to reduce the running costs and only have reasonable performance goals. However, for any case study vehicle the driveability must be considered as shifting gears at certain points can result in unacceptable vehicle behaviour.

It is necessary for the research undertaken in this thesis to understand how to optimise a gearshift map for any multiple-speed transmission. As several transmissions will be compared in different case study vehicles, each transmission/vehicle package will need to be optimised so that proper comparisons can be made.

To maximise the performance of a vehicle the gearshift points need to be selected at 100% throttle to reduce the acceleration times whilst still attaining the maximum top speed. The optimal shift point to reduce the acceleration times is set to stay in the highest available torque at each respective vehicle speed. As can be seen from Figure 2-47 the available torque in first gear is much higher than second gear resulting in a higher vehicle
acceleration however first gear has a limited available maximum speed of 30 km/h. Therefore, to minimise the acceleration time across the whole vehicle speed range the vehicle would change gear at 30 km/h.

To optimise the energy consumption of a vehicle, the gearshift map must be optimised to keep the vehicle in the highest efficiency operating point at any vehicle driving condition (wheel torque and vehicle speed). The energy consumption of the vehicle is predominantly concerned with the efficiency of the drivetrain which consists of the power plant, transmission and differential. The resistive forces due to the tyre dynamics, air resistance and road inclination must also be considered.

Internal combustion (IC) engines efficiency is dictated by its brake specific fuel consumption (BSFC) at each operating point. The BSFC determines the ability to turn fuel into usable power (rate of fuel consumption divided by the power produced). An electric drivetrain consists of a motor which has an efficiency that varies across the speed and torque operating range, similarly to the BSFC of an engine (an electric drivetrain may also include an inverter which has its own efficiency map). However, as can be seen from Figure 2-48, when considering an electric drivetrain the efficiency of the system varies significantly between each gear. For example, at a required wheel torque of 500 Nm and wheel speed of 100 r/min the efficiency in first gear is ~75% whereas in second gear it is ~80%. Therefore a gearshift map needs to be designed which puts the transmission in the gear with the highest drivetrain efficiency for each requested wheel torque and vehicle speed.

When considering the shift logic for an electric vehicle the downshifting strategy in braking is far more important than for an IC engine vehicle. In an IC vehicle the engine is only used to provide some braking force through the drivetrains inertia and internal friction, however in an electric vehicle braking energy can be converted into electricity through regenerative braking. During braking the motor can, in effect, be turned into a generator converting mechanical energy into electrical energy recharging the battery. The drivetrain has different efficiencies during recharging for different gears at each operating point so it is as important to select the correct gear during braking as during traction.
Review of the state of the art: Gear/state shift control

Generally shift control falls into three categories. Single-parameter-controlled shift logic which is solely dependent on vehicle speed has the main shortcoming of the driver not being able to manipulate the performance of the vehicle in any way. Further disadvantages are explained in Ge et al. (2001) and Yamaguchi, Narita and Takahaski (1993). Dual-parameter-controlled shift schedules consider the throttle position along with the vehicle speed to provide a more comprehensive gearshift control system that takes into account the drivers aim in real time and is the gearshift control method adopted in most production vehicles. However, this system is still limited as it cannot predict sharp changes in acceleration or fast changing road conditions which can result in frequent and undesired gear shifting. The latest tri-parameter-controlled gearshift schedules consider not only the vehicle speed and throttle position but the vehicle acceleration (or in some cases the throttle pedal acceleration) to deliver a more accurate prediction of the drivers intention. However this design system is also limited as the same inputs may relate to different driving conditions for a laden vehicle on a flat road or an unladen vehicle on a slope.

Control theory can be employed to design and optimise a more complex and comprehensive gearshift schedule for a vehicle. There are very few papers published which focus on gearshift map design for electric vehicles due to the technology being so modern. However there are several papers on the subject of gearshift schedule optimisation for IC engine vehicles with automatic transmissions or DCTs and the theory can be applied to electric vehicles. For example, Yamaguchi et al. (1993) proposed a system based on fuzzy logic to control a production vehicles shift logic which includes the ability to prevent shift hunting. A knowledge based gear system for automatic transmissions with IC engines was developed by Qin, Ge and Lin (2004). A neural-network based on vehicle speed, throttle position and vehicle acceleration for automatic transmissions is proposed in Yin et al. (2005).
A selection of the papers published on this topic will be explored in more depth in the next section to understand the state-of-the-art control methods being adopted in this field.

In Jun-Quí, Meng and Ya-Qi (2012) the dynamic performance shift logic, which is designed to minimise the acceleration of the vehicle, is extrapolated from test data as shown in Figure 2-47. The point at which the maximum acceleration available at each throttle position in each gear intersects is the optimum shift point. If the lines in first and second gear do not intersect the shift to second gear is the maximum speed available in first gear.

The authors use a similar technique to generate the shift logic to optimise the efficiency of the drivetrain over any drive cycle. A graph, as shown in Figure 2-49, is created which illustrates the efficiency of the drivetrain in each gear at each vehicle speed and is repeated for each throttle position. The speed at which the lines in first gear and second gear intersect gives the vehicle speed where the vehicle should change gear. The final shift map uses the throttle position and vehicle speeds as inputs, so becomes a dual-parameter shifting strategy.

The authors only place one constraint on the shifting strategy designed to optimise the economy which is that the vehicle must accelerate to 50 km/h in 25 seconds. The shifting strategy is then limited to account for this constraint and is shown in Figure 2-50. The downshifting strategy is simply created by subtracting 5 km/h from the upshift strategy to avoid any unwanted oscillations from the shifting control unit.

The authors tested the shifting strategy on the NurembergR36 driving cycle and compared the strategy previously explained with a single-parameter shifting strategy with a fixed upshift point of 15 km/h and a down shift speed of 10 km/h. The authors found the single-parameter shifting strategy to consume 5.15 kWh whilst the optimal shifting strategy used 3.73 kWh giving an energy consumption improvement of 27.5%.

Although the authors have shown their strategy to give a significant improvement over a single-parameter shifting strategy the method is quite crude as it does not account for the drivers intention. In addition, it is lacking any attempt to optimise the regeneration of energy in braking by simply defining the downshift map as 5 km/h less than the upshift
Review of the state of the art: Gear/state shift control

map. Finally the authors do not explain how the acceleration graph shown in Figure 2-47 was created, through a derivation, modelling or real world testing.

Liu et al. (2009) use a similar technique to Jun-qui, Meng and Ya-Qi (2012) but for a six speed DCT. The dynamic performance shift strategy was extrapolated from the acceleration graph shown in Figure 2-51 where the vehicle acceleration was simply calculated from a vehicle dynamic equation, given below:

\[
a_n = \frac{T_e i_n i_a \eta_T - R_{aer} - R_f - R_I}{\delta_n m}
\]  

(29)

where \(a_n\) is the acceleration of the vehicle, \(i_n i_a\) are the gear and final drive ratios, \(R_{aer}\) is the aerodynamic resistance force, \(R_f\) is the force due to rolling resistance, \(R_I\) is the road inclination force, \(\delta_n\) is the equivalent mass of the vehicle and \(m\) is the vehicle mass.

The shift schedule designed to optimise the fuel economy is taken from the fuel rate graph shown in Figure 2-52, similarly to the efficiency map given in Jun-qui, Meng and Ya-Qi (2012).

The final dynamic performance shift schedule is given in Figure 2-53 which is modified to increase the buffer zone between the upshift and downshift lines according to the throttle angle. Furthermore the schedule forces a down shift if the throttle demand is above 70%, which infers that the driver requires more performance or wheel torque due to an incline or heavy load.

The optimised shift schedule for fuel economy is given in Figure 2-54. The shift schedule is used by the transmission controller however further shift logic is introduced to account for the driving condition. If the acceleration of the vehicle is not positive after an upshift demand it is not allowed and if the throttle is not wide open whilst the acceleration is positive a downshift during acceleration is not allowed.
The gearshift map generation technique presented in Liu et al. (2009) has the same shortcomings as explained in Jun-Qui, Meng and Ya-Qi (2012) however through adding an explanation of the generation of vehicle acceleration the research is better presented. Furthermore through adding some shift logic to account for the driving condition a more holistic shifting strategy is developed which is more beneficial to the driver.

In Jun-qiang, Guang-ming and Yan (2008) the authors presented their research on the development of a pure electric bus which was to be introduced for the Beijing Olympics. For this application the authors used a dual-parameter shift map using vehicle speed and accelerator pedal position as the inputs to the shift table. An inactive shifting strategy was introduced to reduce the unwanted shift demands due to sharp changes in the throttle angle which can be a detrimental consequence from adopting a dual parameter shifting strategy.

The shift map is shown in Figure 2-55 and is designed through optimising the battery and electric motor/transmission efficiency point whilst considering the vehicle dynamics. In addition, the authors suggest that the shift map is designed to help maintain the battery charge level through optimising the gearshift points to maximise regenerative breaking. However, the technique used to derive the gearshift map is not quantitatively explained. The low speed shift points promote good efficiency as moving into the higher gears pushes the electric motor into higher torque demands where the motor efficiency is higher.

The result of optimising the shift schedule is illustrated in Figure 2-56 where the energy consumption of the vehicle is compared for the optimised shift schedule, a fixed second gear and fixed third gear run. The figure is not properly explained in the paper but it can be assumed that the right bar is for the motor, the middle bar includes the transmission and the left bar is the total energy consumption. From the graph it is evident that motor is the main drivetrain component affecting energy consumption. The graph shows that a 9 % reduction in the energy consumption is achieved through optimising the shift schedule. In addition, the acceleration time is reduced from 28.3 s for a second gear 0-50 km/h manoeuvre to 23.2 for the optimised shift schedule giving an improvement of 18 %.
The work by Jun-qiang, Guang-ming and Yan (2008) was expanded in Xiong et al. (2010) where the authors researched alternative techniques to optimise the gearshift map. The authors initially adopt a vehicle-speed method which is similar to the procedure outlined in Jun-qiang, Guang-ming and Yan (2008). Firstly an acceleration graph is drawn dictating the acceleration produced by the vehicle in each gear at interval of throttle position as shown in Figure 2-57. Constant speed lines are drawn and where they intersect the acceleration lines for each gear the efficiency is taken which results in Figure 2-58 and the intersection point is where the gearshift point at 10 km/h (in this case) is.

However, the authors found that the shift point throttle positions dictated using this method were less than 50 % so could not be applied in reality as it would significantly affect the vehicles performance. Therefore they analysed two different methods called the throttle-method and traction-force-method. The procedures used to design the shift maps for these methods are similar to the vehicle-speed method but instead of constant speed lines, constant throttle or traction force lines are drawn on Figure 2-57.

The results of the two methods were compared using a vehicle simulator carrying out a drive cycle based on the actual route the electric bus will take in real life. The gearshift map designed using the traction-force-method was found to be more economical than the
throttle-method with a ~0.5 kWh (164.92 vs 164.469 kWh) advantage at half-load. The authors then employed a graphical method to optimise the shift points through plotting the efficiency of the drivetrain at each point in the drive cycle for each gear as shown in Figure 2-59.

![Figure 2-59: Efficiency curve comparison (Xiong et al., 2010)](image)

The points highlighted (A, B and C) are where the gearshift was too early or late and thus the drivetrain was not moved to the optimal gear at the optimal time. Through forcing the vehicle to shift at the correct point a further 0.5 kWh reduction was achieved. The final results comparing the new optimised economic shift schedule based on the traction-force-method with the method from Jun-qiang, Guang-ming and Yan (2008) (existing shift schedule) and fixed third gear driving for varying vehicle loads, are shown in Table 2-3

<table>
<thead>
<tr>
<th>Vehicle load</th>
<th>Optimised economic shift schedule [kWh/100km]</th>
<th>Existing shift schedule [kWh/100km]</th>
<th>3-gear driving [kWh/100km]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Idle</td>
<td>147.21</td>
<td>149.13</td>
<td>156.4</td>
</tr>
<tr>
<td>Half-load</td>
<td>163.9</td>
<td>166.46</td>
<td>174.63</td>
</tr>
<tr>
<td>Full-load</td>
<td>181.21</td>
<td>183.93</td>
<td>192.84</td>
</tr>
</tbody>
</table>

The results in Table 2-3 show a significant reduction in energy consumption over single gear driving and a definite reduction over the existing shift schedule. However, the method used to design the shift schedule is limited to inputs from the throttle position and vehicle speed and is not accounting for the changing load demands due to road inclination.

In Hayashi et al. (1993) a more complex method for developing a gearshift map is proposed based on a combination of fuzzy control (Tanaka and Wada (2007)) and neural network control, namely neuro-fuzzy control. Fuzzy control has been explained previously but essentially uses values attained from the physical system as inputs for a set of control rules used to generate output values. Neural network control differs in that it is based on a large amount of data generated through physical testing to predict the output. The overview of the control system is illustrated in Figure 2-60.
The authors initially explain a gearshift map generated for the case study vehicle to be used as a base to compare the new control system against. This gearshift map differs from the previous papers mentioned as it is not only based on vehicle speed and accelerator pedal position but also on the displacement rate (velocity) of the accelerator pedal position, Figure 2-61. This provides increased gear shift control as the velocity of the accelerator pedal gives some indication of the driver’s intention.

For this application fuzzy control is used to estimate both the vehicle load and the driver’s intention. The estimated load is found through a series of rules and membership functions based on the vehicles speed and acceleration, similar to (Tanaka and Wada (2007)) mentioned earlier. In the same way, the driver’s intention is found from rules and membership functions using the accelerator position displacement and accelerator position rate of movement. It is not explained how the final fuzzy output function is calculated, through centroid of area, maxima/minima, etc.

The outputs of the two fuzzy control membership functions are then inputted into the neural network controller as in Figure 2-62.

Along with the results of the two fuzzy controllers, the neural controller also uses the vehicle speed and accelerator position, with several hidden layers and produces the gear position as the output, as in Figure 2-63.

The neural network control (Hagan and Demuth, 1999) tunes a set of rules which can predict the outcome of the system based on the state of the inputs. Each node in the
hidden layer consists of a log-sigmoid function (or similar) where the raw inputs are individually weighted and summed for each node. A neural network may consist of several hidden layers with a number or functions to allow the system to effectively predict the correct response. The output layer has a node for each system output and consists of a linear function using the sum of all the outputs of the hidden layer. The weights for each node are calculated through a back-propagation method to tune the system against real world data so any variation of the inputs gives the optimal output.

Specifically, the authors state that a 3 layer back-propagation method was adopted to tune the membership functions however little explanation of the methodology or the specific functions used in the neural network were presented.

The two gearshift maps were then tested on an experimental vehicle for a short manoeuvre consisting of a 10% and 8% slope where one can assume the objective for the control system is to keep a constant speed of 40 km/h. The results for the conventional method using the three-parameter shift maps mentioned earlier and the new neuro-fuzzy control can be seen in Figure 2-64.

The vehicle tested with the conventional shift map can be seen to shift gears frequently requiring the driver to alter the accelerator position regularly causing more shifting and creating oscillations in the vehicle speed. The unwanted shifting is provoked by the changes in road inclination which give rise to different vehicle load conditions and which the three parameter shift map cannot account for. The vehicle with the shift map created by neuro-fuzzy control only shifts gear once, reducing the need to alter the accelerator pedal position so the vehicle maintains a steady speed. A steady speed is conducive to good fuel economy.
(reduced energy consumption for an electric vehicle) as the drivetrain is likely to remain in a high efficiency point. Gearshifts increase carbon emissions in petrol or diesel vehicles due to the wasted energy during the torque gap and are equally deficient for electric vehicles in that they increase the amount of wasted energy due to the disengagement of the clutch and/or clutch slip.

A fuzzy-neural network was also adopted in Li and Hu (2010) to generate a shift schedule although in a different way to Hayashi et al. (1993). Whereas Hayashi et al. (1993) combined fuzzy control and neural network control in the vehicle controller, Li and Hu (2010) used neural network control to tune the membership functions required for fuzzy control which is the primary vehicle controller.

In Li and Hu (2010), the fuzzy control uses the vehicle speed and engine throttle as inputs which represent the first layer of nodes in the neural network and simply pass their values to the second layer. The second layer represents the membership functions which need to be tuned. The third layer consists of the predetermined fuzzy rules where the outputs are normalised against the sum of the third layer outputs in the fourth layer. The fifth layer nodes simply represent the gears which can be selected.

The membership functions (as previously seen in Figure 2-46) need to be tuned through the neural network. The second layer consists of twelve nodes, seven for the vehicle speed and five for the throttle position. There are seven Gaussian membership functions for the vehicle speed representing negative middle, positive large, etc. The throttle position membership function is similarly composed of five Gaussian membership functions. Therefore the values to be tuned through the neural network are the width of each Gaussian membership function and the centre of each value along the x-axis of the membership function.

The learning algorithm to ascertain the values of the centre and width of each Gaussian function represented by each node of the second layer is found through a back propagation method using the error of the output and the target output. The target outputs used to tune the neural network control were found from real world test data where a skilled driver logged a large amount of shift points for various speeds and throttle positions.

The authors simulated two driving cycles comparing the gearshift schedule found through the neuro-fuzzy control with a simple dual-parameter gearshift. The results of the two driving cycles are shown in Figure 2-65 and Figure 2-66.

The results of the driving cycle show that the number of gearshifts were reduced for the neuro-fuzzy gearshift map compared to the dual-parameter. This is due to the variations of throttle during the drive cycle provoking unnecessary gearshifts with the dual-parameter gearshift map whereas the neural network has ‘learned’ to incorporate the drivers intentions through the membership functions.
Review of the state of the art: Gear/state shift control

The results of the driving cycle show that the number of gearshifts were reduced for the neuro-fuzzy gearshift map compared to the dual-parameter. This is due to the variations of throttle during the drive cycle provoking unnecessary gearshifts with the dual-parameter gearshift map whereas the neural network has ‘learned’ to incorporate the drivers intentions through the membership functions.

The research on neuro-fuzzy control presented in Li and Hu (2010) is novel due to the intelligent use of adopting neural network control theory to tune fuzzy logic membership functions. However, it is only a method to tune the membership functions faster than tuning them on a test rig (or other manual method) through trial and error. The control system still needs to use test data which can be imperfect as the driver may not be driving most efficiency or in the most desirable fashion relative to the requirements of the passenger. Essentially it is only suited to generating a gearshift map which can accurately simulate that of a real world driver and potentially have the advantage of increasing the enjoyment of the driver.

In Casavola, Prodi and Rocca (2010) the authors compare a gearshift map generated through fuzzy logic with a new method based on an online optimisation which computes the most efficient gear to be in at each instant. The focus of this paper is to maximise the economy of the case study vehicle over the NEDC and not consider the performance of the vehicle. Consequently, whilst the results are not applicable to a real world vehicle it does give an indication of the benefits of each strategy.

Initially the gear ratios for the case study vehicle are optimised for the NEDC using the gearshift points imposed for a manual transmission through adopting a MATLAB toolbox called fminsearch. The authors found that through the optimisation of the gear ratios alone, whilst maintaining the same gearshift points, a 5.9% fuel saving was found.

The first of the two gearshift map optimisation strategies (called EGA – Efficient Gear Actuator) uses the efficiency maps of each gear to calculate the fuel required to provide the power needed to satisfy the current driving condition in each gear simultaneously. Through comparing each result, the gear which uses the least fuel to satisfy the current driving condition can be selected. The layout of the control system is illustrated in Figure 2-67 where the inputs to the control system dictated by the driving conditions are first fed into a...
filtering block to ascertain how much power will be required from the engine for the current conditions in each gear. The second block selects which gears are available according to certain constraints such as keeping the engine speed above idle, etc. It should be noted that for an electric vehicle the idle speed does not exist. The third block calculates the efficiency of the drivetrain in each available gear using experimentally attained efficiency maps, as shown in Figure 2-68, to select which gear would result in the minimum fuel consumed by the engine to satisfy the driving conditions. The final block applies some limits to the gearshift points to produce more realistic results such as a buffer between gearshifts (as gearshifts do not happen instantaneously).

The optimised shiftmap through the EGA method gave a 9.14% improvement over the vehicle with non-optimised gear ratio and the standard NEDC gearshift map. This is primarily due to the EGA method requesting a high gear at each instant as shown in Figure 2-69 below which reduces the fuel consumption.

The fuzzy logic method is similar to the method in Hayashi et al. (1993) where two fuzzy controllers are used to determine initial parameters to be fed into a secondary controller as shown in Figure 2-70. For both papers the drivers intention is predicted, however in Casavola, Prodi and Rocca (2010) the engine state is predicted instead of the vehicle load. The output of these two fuzzy logic controllers are an actual predicted gear value and are fed into another fuzzy controller which outputs a ‘shift up’, ‘shift down’ or ‘maintain gear’ request. Each fuzzy controller consists of the membership functions and rules as previously explained.

The membership functions were all tuned against data collected from a test vehicle and as such is not optimised to minimise the vehicle energy consumption or the performance. Therefore the authors used a genetic optimisation to modify the vertexes of the membership functions to encourage the system to upshift early, which essentially means widening the membership functions to promote an upshift and therefore better fuel consumption. The result of this control system gave an improvement over the EGA method which was almost as good as the benchmark value, being 10.97% over the non-optimised case study.
In conclusion, Casavola, Prodi and Rocca (2010) first introduced a method based on the efficiency of the drivetrain which resulted in reduced fuel consumption but is dependent on accurate efficiency maps attained through test rigs offline. The main issue with this method is that the efficiency maps will change over time as the vehicle ages, reducing the accuracy of the gear choice predictions. The second method using fuzzy logic and genetic optimisation gave excellent results and showed how although the system is tuned on real world test data, the membership functions can be modified to promote better fuel economy.

In Ngo et al. (2010) the authors propose a method of adopting dynamic programming to select the optimal gear at each point over a driving cycle to find the absolute minimum fuel consumption. Dynamic programming, as first proposed by Zadeh (1965), is a control method whereby a large problem can be broken up into small sections which can be optimised individually. When each small section is optimised it includes the overall optimal solution from the previous sections so the global solution can be found with reduced computational demand as each section only needs to be computed and optimised once.

Dynamic programming works retrospectively so in this case the gear ratios are optimised over a driving cycle starting from the end working towards the start. For this application the gear shifts need to be optimised over a driving cycle so instead of running a simulation calculating the fuel consumption for every possible combination of gearshift changes over the drive cycle (i.e. for an 800 second drive cycle for a transmission with 6 gears and a time step of 1 second would require $6^{800}$ iterations) only the fuel consumption for each gear at each time step needs to be calculated once. The global solution is found by saving the combination of gears which give the minimum fuel consumption to end up in each gear leading up to that step.

An innovative transmission is used for the research called a PS-AMT (Power Shift AMT) which consists of a bypass controller to pass torque during shifting so there is no torque interruption at the wheels during gearshifts. However, for the purpose of dynamic programming the gearshift dynamics and therefore the clutch losses are not considered. The gearshifts are assumed to take at least one second to complete so a time step of one second is used, meaning the control system calculates the optimum gear ratio to be used over each second.
Initially the authors defined a cost function that minimised the fuel consumption over the driving cycle with length, $t_f$. The problem is formulated as to solve an optimal control law $u'(t)$ that optimises the control variable. The optimal control law is shown in equation (30) below, where $\dot{m}_f(x(t), u(t))$ defines the fuel rate used by the vehicle.

$$J_1 = \int_0^{t_f} L(x(t), u(t))dt = \int_0^{t_f} \dot{m}_f(x(t), u(t))dt$$

(30)

The dynamics of the powertrain are considered through the function $\dot{x} = f(x(t), u(t))$ which gives the longitudinal dynamics of the vehicle. The state variables, $x(t)$, are the engine torque, $T_e(t)$, and the gear ratio, $r_g(t)$, whereas the control variable, $u(t)$, is the rate of gear ratio change, $\dot{r}_g(t)$. Each of the state and control variables are subject to physical constraints such as the engine torque being constrained between the maximum and minimum available engine torque at each engine speed.

The control law is then modified to form a discrete optimal control law over a driving cycle, $N$, as given in equation (31), using time step $k$.

$$J_2 = \sum_{k=0}^{N-1} L(x(t), u(t))\Delta t$$

(31)

The constraints are similar to the non-discretised optimal control law; however, gear ratio limits are defined as $1 \leq n_g \leq 6$ as it is a 6-speed transmission. The control variable, $u(t)$, is the rate of gear ratio change and is limited to a single gear ratio change over each time step, $k$, e.g. from second gear the controller only allows a shift to first or third or to remain in second. It is this limit which makes the use of dynamic programming a viable option to optimise this problem. If any gear was allowed in each time step it would be a simple case of finding the most efficient gear to produce the optimal gearshift schedule.

The dynamic programming algorithm is then given below in equation (32).

Step $k$, $(0 \leq k \leq N - 1)$

$$J^*_2(k(x_i(k)) = \min_{u(k)=1\in u_g(k)} \left[L(x_i(t), u(t)) + J^*_2(k(x_j(k + 1)) \right]$$

(32)

The dynamic programming algorithm is calculated at each time step for each possible control law movement, i.e., each possible gearshift. The result, $J^*_2(k(x_i(k))$, is then calculated three times for each gear ratio, so 18 times, except it is impossible to upshift from sixth gear or downshift from first so in fact it is calculated 16 times each time step, $k$. The equation $L(x_i(t), u(t))$ is the fuel consumption over the current time step to move from the current state to the next depending on the control variable where $J^*_2(k(x_j(k + 1))$ is the optimal fuel consumption to be in that state from previous time steps. The entire function is minimised to find the optimal option from the control variable for each state, i.e. whether an upshift, downshift or no-shift would result in the minimum fuel being used.
The results of the dynamic programming were found for the ECE drive cycle and compared against baseline results which used prescribed gearshift points that can be seen in Figure 2-71 and are referred to as “normal shifting”. The results of the dynamic programming gave a 15.4% fuel consumption improvement over the “normal shifting” using a backward facing model for comparison. The engine torque and speed are shown in Figure 2-71 along with the gear ratios and the operating points in the engine BSFC map are shown in Figure 2-72 where the improvement can clearly be seen as the DP shifting points are in regions with a lower overall BSFC.

The authors then went on to compare the two gearshift schedules in a forward facing model and confirmed an improvement of 15.7%. An experimental validation was carried out with a test vehicle on a rolling road and a skilled driver which attempted to follow the driving cycle speed and shift points for each of the two gearshift map methods and found an 11.2% improvement.

Dynamic programming is an excellent tool to find the optimal solution for a finite horizon control problem however it is not practical for real world applications. This is due to dynamic programming working backwards and requiring knowledge of the velocity profile prior to the optimisation so no ‘online’ optimisation can take place. Essentially it is a good method to find the optimal solution using little computational power due to the reduced number of iterations required to find the solution compared to other brute force methods. In addition, dynamic programming can be used to find the optimal solution by which other control methods can be compared. The question is whether we simply need a local optimum or a global solution to the specific problem of electric vehicle energy management.

2.7 CONCLUSION
The review of the literature concerning the state-of-the-art technology relating to electric vehicle drivetrains has been comprehensively completed.

Initially, the review has found that there are several electric motor types being adopted for electric vehicles such as PMSM, SRM, IM, and there is not a clear favourite. However, this project will not focus on battery or electric motor technology due to the technology being more relevant to an electronic or chemical research topic.

The research up to date relating to multiple-speed transmission for electric vehicles has been reviewed and found to be quite limited. The benefits of employing a multiple-speed transmission have been clearly defined by Knödel (2009) for example by illustrating that the electric motor can operate in a higher efficiency operating point for a number of vehicle operating conditions. Therefore there are some potential energy consumption benefits of adopting multiple-speed transmissions for electric vehicles.

The subject has been broached by various authors such as Knödel, Ren, Crolla and Morris, however the techniques employed were limited in their complexity and do not result in a comprehensive conclusion. For example, the vehicle models used in Knödel et al. (2010) and Ren, Crolla and Morris (2009) do not consider the transmission efficiency, gearshift dynamics or include any optimisation of the drivetrains considered, i.e. gear ratios. The research suggests there is a significant benefit to be seen, both in energy consumption and in vehicle performance through adopting a multiple-speed transmission over a single-speed transmission for electric vehicles and points to more research in this area being beneficial.

A number of different multiple-speed transmission designs for electric vehicles were reviewed to understand the difference in the design architecture of traditional manual or automatic transmissions. The main factor driving any difference in the design architecture is that an electric motor is not limited by the idle speed of an internal combustion engine so a clutch is not required during launch or in some cases during a gearshift. For instance, Risele and Bitsche (1995) present a transmission which does not utilise a clutch, merely a dog clutch and control of the electric motor for synchronisation. The research in this project will not focus on the design or development of a brand new transmission due to the significant costs involved, the transmissions will be working prototypes developed by the industrial partners involved.

The lack of an idle speed and the improved reaction time of an electric motor lends itself to different gearshift methodologies as presented in the literature review. It was important to review the gearshift methods to understand which methods could be employed in simulation when modelling the prototype transmissions.

The final section of the review of the state of the art concerns research into gearshift maps and gear/state shift schedules. The use of multiple-speed transmissions, whether with internal combustion engines or electric motors, requires a robust method to ascertain the correct gear to be in for each driving condition considering fuel economy and vehicle performance. The review has seen that many different methods have been adopted, from the relatively simple approach by Liu et al. (2009) to the complex neuro-fuzzy approach by
Hayashi et al. (1993). These methods can be employed offline and used to generate shift maps or membership functions, however certain methods such as dynamic programming (Ngo et al. (2010)) can only be used as an analysis tool and not in practice. Each method appears to be effective, however no comparison between the methods can be made unless they are applied to the same case study vehicle. The simulation analysis carried out in this project will require the development of a method to ascertain which gear to be in at each instant and there is certainly scope to add to this field of research.

In conclusion, the review has shown that there are considerable grounds to add to the field of electric vehicle research and particularly relating to the adoption of multiple-speed transmissions.
3 SIMULATION MODEL DEVELOPMENT

3.1 INTRODUCTION

The aim of the research is to investigate multiple-speed transmissions for electric vehicles. Simulations were the main tool adopted for this research as it only requires an understanding of a system and the derivation of the governing equations to develop a model that, with accurate data, can produce accurate results.

The models developed by the author were generated in Matlab (Matlab, 2015) as it is a high level technical tool allowing rapid processing of complex systems while giving the user the ability to easily visualise data. In addition, Matlab is a comprehensive programming tool allowing the user the ability to quickly program algorithms for post processing whilst a model can be developed in Simulink which allows excellent visualisation of a model/system.

The other motivation for adopting Matlab/Simulink as the simulation environment is that a dSPACE unit was used to control the test rig which uses c-code generated through simulink so any models developed can be ported directly into dSPACE.

The majority of the research is based on two prototype transmissions developed by Oerlikon Graziano and Vocis Drivelines which are to be installed on a test rig at the University of Surrey. Therefore, the first models are based on these two transmissions (a single-speed and two-speed) which will allow for initial predictions of performance and energy consumption and later allow validation of the test rig results. The models will also be used as the basis for the test rig models uploaded into dSPACE.

The aim of the models is to analyse firstly the energy consumption of case study vehicles over standard driving cycles and to analyse the performance of the vehicle. When developing models there is always a trade-off between accuracy, simulation time, size and complexity. When considering a model for simulating driving cycles which may last up to an hour it would be beneficial to reduce the complexity of the model to reduce the simulation time. In the same way, a model which is to be used to undertake acceleration tests which may only last one hundred seconds can be more complex.

When considering non-linear models of fixed parameter systems there are two different types of models which can be considered, and are referred to as “Backward-Facing” or “Forward-Facing”. As the non-linear system being adopted in this research is the vehicle the concept of backward-facing (Sorniotti, 2010) and forward-facing (Sorniotti, 2011) can be easily explained. Essentially a backward facing simulation is quasi-stationary and can be expressed as one equation, where the governing equations of the system can be derived to create a single input/single output equation.

When considering a vehicle the input to a backward-facing model is the required wheel torque which can be ascertained by the forces on the vehicle, i.e. the aerodynamic force, rolling resistance, etc. The equations which simulate the mechanics of the wheels, drivetrain and transmission whilst considering the efficiency and the moment of inertia of
Simulation model development: The vehicle model

each component are included. This allows the computation of the required torque of the electric motor and therefore the power required by the electric motor which gives the energy consumption. The speed of the electric motor is known at all times as it is kinematically linked to the wheel as the wheel speed is dictated by the driving cycle. Essentially it allows the calculation of the steady state values of the vehicles parameters at the cost of significant complication.

A backward-facing model does however, have its down falls as it does not generally allow the computation of dynamics within the vehicle systems, i.e. tyre slip, half-shaft torsion or gearshift dynamics. To model these phenomena and dynamics a forward-facing model can be adopted which uses the throttle (or drive demand) as an input to the model and the behaviour of the system is a consequence of this input. The same governing equations, efficiencies and inertias that are used in a backward-facing model are used here also but the dynamics can be modelled. The simulation time of a forward-facing model is increased over a backward-facing model as the computation time for each step is larger due to the models complexity.

Each model type was adopted for various studies that took place in this research and will be explained in more depth in this chapter. The forward-facing models of the single-speed and two-speed transmissions for simulating driving cycles will be explained in the next section as they include all of the vehicle dynamics and were adopted for the performance simulations too and leads onto the simulation of the gearshift dynamics of the two-speed transmission. Backward facing models were adopted for multi-parameter optimisations to reduce the computation time and will be explained later on in the thesis.

The models mentioned in this chapter are non-linear, however linear models were developed to analyse low frequency system response, i.e. vehicle drivability, for various manoeuvres.

3.2 THE VEHICLE MODEL

The vehicle model is identical for any powertrain and whilst the parameters such as mass, wheel base, wheel radius, etc, can be modified to simulate different case study vehicles the governing equations are the same. The vehicle model consists of the vehicle body which is the ‘sprung mass’, due to it being connected to the ‘unsprung masses’, which are the wheels, through the suspension. The longitudinal and vertical acceleration of the vehicle can be found through considering the forces acting on the vehicle body which are essentially the wheel torque and the resistive forces.

The resistive forces are the aerodynamic drag, which is a fluid drag force that acts on a body moving through another substance, in this case a vehicle moving through air. The equation for calculating the aerodynamic force, $F_{aer}$, is given below in equation (33). $\rho$ is the air density, $S$ is the frontal area of the vehicle, $C_d$ is the aerodynamic drag coefficient and $V$ is the vehicle speed.
Simulation model development: The vehicle model

\[ F_{\text{aer}} = \frac{1}{2} \rho S_v C_d V^2 \]  

(33)

The other resistive force is the rolling resistance acting between the tyre and the road. The rolling resistance is mainly due to hysteresis which is caused by the tyre constantly deforming under compression and recovering as the tyre rotates which gives off energy as heat. The rolling resistance can be ascertained through coastdown data or through a polynomial where the coefficients are tuned to match the vehicle’s behavior. The equation for the rolling resistance, \( F_{\text{roll}} \), is given below, where, \( f_0 \) is the constant rolling resistance coefficient, \( f_1 \) is the rolling resistance coefficient relating to vehicle speed and \( f_2 \) is the rolling resistance coefficient relating to vehicle speed squared.

\[ F_{\text{roll}} = f_0 + f_1 V + f_2 V^2 \]  

(34)

The constant, \( f_0 \), is defined by the road conditions and varies according to the road surface material, i.e. tarmac (0.013). The constants dependent on the vehicle speed are tuned to match the vehicle/tyre characteristics. The vehicle also has to overcome any resistance due to the inclination of the road, along with any inertial changes due to the vehicle accelerating.

An equivalent trailing arm configuration has been adopted for the suspension model, as any suspension design can be modelled using this layout, through a graphical equivalency procedure (Reimpell et al., 2000) starting either from the three-dimensional suspension geometry, or from the experimental characterisation of the suspension elasto-kinematics (i.e., wheel longitudinal displacement as a function of bump). The variation of the equivalent pivot point (represented by the dimensions \( c,d,e,f \) in Figure 3-1) of the trailing arm onto the chassis can be easily incorporated in the form of a look-up-table within the model. Therefore, this simplified but representative model includes the non-linear anti-dive, anti-lift and anti-squat characteristics of the suspension systems, which affect the sprung mass pitch dynamics.

The vehicle is illustrated through the free-body diagram of the sprung mass in Figure 3-1 showing the torques at the wheels, the resistive forces, the vehicle inertia and the vehicle dimensions. The variables in the diagram are explained in the following section.

The symbols used in Figure 3-1 and in the equations here after utilize subscripts where \( f/r \) represented front/rear, \( L/R \) represents left/right and \( x/z \) represents longitudinal/vertical force directions respectively.

The equations for the sprung mass longitudinal acceleration \( \ddot{x}_{sm} \), vertical acceleration, \( \ddot{z}_{sm} \), and rotational acceleration, \( \dddot{\theta}_{sm} \), can be derived from this diagram, Figure 3-1. The longitudinal acceleration of the vehicle is found from the sprung mass longitudinal force balance equation, equation (35). The tractive forces from the tyres are transmitted to the vehicle longitudinally through the suspension joints, \( F_{jz} \); the external forces that affect the sprung mass are also included, namely, the aerodynamic drag force and the resistive force for the sprung mass due to the inclination of the road, \( F_{R \times x_{sm}} \); \( m_{sm} \) is the sprung mass mass.
Simulation model development: The vehicle model

Figure 3-1: Free body diagram of the sprung mass.

\[ m_{sm} \ddot{x}_{sm} = \sum_{k=L,R} F_{j_{sf,k}} + \sum_{k=L,R} F_{j_{sr,k}} - F_{aer} - F_{RG,x_{sm}} \]  
(35)

The sprung mass vertical force balance is shown in equation (36), giving the sprung mass vertical acceleration. This includes the vertical forces transmitted by the suspension arms through the suspension joints, \( F_{j_{sf}} \), the vertical forces caused by the suspension spring and damper system, \( F_{j_{sr}} \), along with the force due to the weight variation of the sprung mass when there is an inclination of the road, \( \Delta F_{RG,x_{sm}} \). The vertical displacement of the sprung mass affects the drivability of the vehicle along with the vehicle pitch.

\[ m_{sm} \ddot{x}_{sm} = \sum_{k=L,R} F_{j_{sf,k}} + \sum_{k=L,R} F_{j_{sr,k}} + \sum_{k=L,R} F_{j_{sf,k}} + \sum_{k=L,R} F_{j_{sr,k}} + \Delta F_{RG,x_{sm}} \]  
(36)

The pitch dynamics of the vehicle affects the weight transfer and consequently the available tractive force as the wheel slip is affected by the vertical force on the tyre. The sprung mass angular acceleration (and by integration, the vehicle pitch), is found through considering the moment balance equation about the sprung mass center of gravity and is shown given equation (37). \( I_{sm} \) is the moment of inertia of the sprung mass, \( H_{CG,sm} \) is the height of the centre of gravity, \( a \) is the longitudinal distance from the front axle to the sprung mass centre of gravity, \( b \) is the longitudinal distance from the rear axle to the sprung mass centre of gravity, \( c \) is the longitudinal distance from the front axle to the equivalent front suspension mounting point, \( d \) is the longitudinal distance from the rear axle to the equivalent suspension mounting point, \( e \) is the height of the front equivalent suspension mounting point and \( f \) is the height of the rear equivalent suspension mounting point. \( F_{roll} \) is the total rolling resistance force (\( F_{react} \) in the diagram).
Simulation model development: The vehicle model

$$J_{sm}\ddot{\vartheta}_{sm} = -\left(F_{fsf,R} + F_{fsf,L}\right)a - \left(F_{fsf,R} + F_{fsf,L}\right)(a - c) - \left(F_{szr,R} + F_{szr,L}\right)b$$

$$- \left(F_{ztr,R} + F_{ztr,L}\right)(b - d) - \left(F_{fsf,R} + F_{fsf,L}\right)(H_{CG,sm} - e)$$

$$+ \left(F_{ztr,R} + F_{ztr,L}\right)(H_{CG,sm} - f) - F_{roll,f} - F_{roll,r}$$

Currently, the model presented neglects the dynamic effect of the mounting system of the powertrain, i.e. the effect of the engine/transmission “twisting” on the engine mounts which could be considered a torque input to the sprung mass. This subject has been studied in Eller and Hetet (2010) and Sorniotti (2008), and requires further analysis, as the powertrain mounting system applies longitudinal and vertical forces to the chassis during torque transients, with a direct impact on vehicle drivability.

The unsprung masses, the “wheels”, can be modeled in the same way as the sprung mass. A free body diagram of the unsprung mass is shown in Figure 3-2 (a), along with the free body diagram of the trailing arm, (b). The longitudinal, vertical and angular accelerations can be found through considering the moment balance equations about the centre of the wheel at the contact patch of the tyre with the road. The three accelerations represent a single degree of freedom as they are linked through the wheel hub. The longitudinal and vertical forces at the pivot point of the suspension arm are also found through the moment balance equations of the unsprung mass.

Note. In this section the equations only represent a single form wheel and not the four individual wheels.

![Figure 3-2: Free body diagram of the unsprung mass, (a) and the trailing arm, (b).](image)

The unsprung mass longitudinal displacement, $x_{usf}$, varies according to the vertical displacement of the wheel, $z_{usf}$, suspension dynamics and vehicle pitch, which combine to alter the angle of the trailing arm, $\gamma_f$. The total unsprung mass displacement is a sum of the sprung mass vertical displacement, $x_{sm}$, (as the vehicle travels) and the additional displacement due to the trailing arm movement, which is also a function of the wheel radius, $R_w$, as shown in equation (38).
Simulation model development: The vehicle model

\[ x_{usf} = x_{sm} + z_{usf} \tan \gamma_f = x_{sm} + z_{usf} \left( \frac{e - R_w}{c} \right) \]  

(38)

The longitudinal force balance equation of the unsprung mass gives the longitudinal acceleration of the unsprung mass through considering the tyre longitudinal force, \( F_{txf} \), resistive force for the unsprung mass due to the inclination of the road, \( F_{RGusf} \), and the unsprung mass, \( m_{usf} \).

\[ m_{usf} \left[ x_{sm} + z_{usf} \left( \frac{e - R_w}{c} \right) \right] = F_{txf} - F_{jzf} - F_{RGusf} \]  

(39)

The force due to the weight variation of the unsprung mass when there is an inclination of the road at angle, \( \alpha \), is found through equation (40), \( g \) is gravity.

\[ F_{RGusf} = m_{usf} g \sin \alpha \]  

(40)

The unsprung mass vertical acceleration, \( \ddot{z}_{usf} \), which is used to calculate the suspension force and vehicle pitch is found by considering a force balance equation through the vertical axis of the unsprung mass. The equation considers the vertical tyre force, \( F_{tzf} \), the vertical suspension force, \( F_{jzf} \), and the vertical force at the suspension pivot point, \( F_{jzf} \), along with the weight variation force of the unsprung mass due to a changing road angle, \( \Delta F_{wusf} \).

\[ m_{usf} \ddot{z}_{usf} = F_{tzf} - F_{jzf} - F_{jzf} + \Delta F_{wusf} \]  

(41)

The weight variation force due to the inclination of the road is calculated using Equation

\[ \Delta F_{wusf} = m_{usf} g (1 - \cos \alpha) \]  

(42)

The rotational acceleration of the wheel, which is used to calculate tyre slip and thus the longitudinal force between the tyre and the ground is found through considering a moment balance equation about the contact point between the wheel and the ground directly below the wheel centre. This equation includes the half-shaft torque, \( T_{hsf} \).

\[ m_{usf} R_w \left[ \ddot{x}_{sm} + \ddot{z}_{usf} \left( \frac{e - R_w}{c} \right) \right] = -F_{jzf} e - F_{RGusf} \frac{R_w}{c} \right) \]  

(43)

The moment balance equation to calculate the angular acceleration is different for a non-driven wheel as there is no half-shaft torque.

\[ m_{usf} R_w \left[ \ddot{x}_{sm} + \ddot{z}_{usf} \left( \frac{e - R_w}{c} \right) \right] = -F_{jzf} e - F_{incusf} \frac{R_w}{c} \right) \]  

(44)

The vertical tyre force, \( F_{tzf} \), is still an unknown and is modeled dynamically as a spring and damper, characterised by the tyre wall deflections. The force due to the tyre stiffness (which is typically very high) is a function of the tyre/road vertical displacement whilst the force due to the tyre damping is a function of the tyre/road vertical velocity. The stiffness
coefficient, $P_t$, and damping tyre coefficient, $C_p$, are considered constants. The inputs to the spring and damper system are the wheel vertical displacement, $Z_{USP}$, and speed, $\dot{Z}_{USP}$, along with the road vertical displacement, $Z_{INP}$, speed, $\dot{Z}_{INP}$ (which may be due to a road bump or obstacle).

$$F_{tzf} = P_t(Z_{INP} - Z_{USP}) + C_p(\dot{Z}_{INP} - \dot{Z}_{USP})$$ (45)

The suspension is modeled in a similar way, as a spring and damper system. The vertical suspension force, considers the vertical displacement of the wheel and vertical velocity of the wheel, along with the vertical displacement of the sprung mass, $Z_{sm}$, and vertical velocity of the sprung mass, $\dot{Z}_{sm}$. The suspension stiffness coefficient, $K_f$, and the suspension damping coefficient, $C_f$. Whereas the stiffness and damping variables are constants for the tyre, due to the increased travel of the suspensions when considering the suspension dynamics look-up tables are used to represent the varying suspension characteristics. Typically the stiffness increases for a reducing displacement, as the spring compresses, up to a point where a “bump stop” is simulated by using a stiffness well above any realistic value.

$$F_{sf} = K_f(Z_{USP} - Z_{sm}) + C_f(\dot{Z}_{USP} - \dot{Z}_{sm})$$ (46)

The vertical displacement (and by differentiation the speed) of the sprung mass at the upper suspension point, $Z_{pf}$, is affected by the pitch of the vehicle, $\theta_{sm}$, and is therefore calculated through equation (47 and 48).

$$Z_{pf} = Z_{sm} - \theta_{sm}a$$ (47)

$$Z_{pf} = Z_{sm} + \theta_{sm}b$$ (48)

Due to the nature of the simulations being carried out in this research no lateral forces or steering/cornering properties are considered. Only the pitch dynamics are modelled due to the impact on vertical tyre loads and therefore traction limits along with the effect on drivetrain oscillations. Therefore only a 2-D suspension system is utilised based around a simplified equivalent trailing arm model as previously mentioned. Adopting this model is beneficial as it can represent the MacPherson strut system installed on a vehicle demonstrator involved in the project which is used for model validation.

The important factor for designing the suspension system when considering the pitch dynamics is the pitch angle, which is the angle between the pivot point and the centre of the wheel on the road surface. If the pitch angle is infinite then the longitudinal forces are concentrated in the wheel centre and the higher the pitch angle is the better the pitch equalization.

The front suspension pitch angle must be calculated to give a pitch angle which provides reaction forces in the vertical direction and thus through the shock absorber.
The rear pitch angle is required to be as high as possible, as with the front to retain pitch equalisation but must also be as close to the wheel centre longitudinally as possible to increase the force which pulls the end down during braking. Although too short a control arm will result in large rotation angles and will not allow the desired spring travel.

The wheel speed calculated in (44) only represents the rotational velocity of the wheel on the road which does not represent the vehicle speed. The propulsive force which propels the vehicle forward is the tyre longitudinal force which is the tractive force between the tyre and the ground. This results in the longitudinal force at the suspension arms that are used to calculate the longitudinal velocity of the sprung mass (or vehicle).

The slip ratio, $\sigma$, is defined as the rotational velocity of the wheel, $\dot{\theta}_{w,f}$, in relation to the equivalent rotational velocity of the vehicle relative to the wheel, $V_{act}$. A greater value denotes greater traction, until wheel spin occurs, and the opposite is true in braking.

The equations to calculate the slip ratio are given below:

- during acceleration:

$$\sigma = \frac{\dot{\theta}_{w,f}R_w - V_{act}}{\dot{\theta}_{w,f}R_w} = 1 - \frac{V_{act}}{\dot{\theta}_{w,f}R_w}$$  \hspace{1cm} (49)

- during deceleration:

$$\sigma = \frac{\dot{\theta}_{w,f}R_w - V_{act}}{V_{act}} = 1 - \frac{\dot{\theta}_{w,f}R_w}{V_{act}}$$ \hspace{1cm} (50)

Therefore the longitudinal slip ratio assumes a value between -1 and +1.

The tyre model adopted the Pacejka ‘89 Magic Formula. The equation, given in (51), generates a curve which represents the tyre longitudinal force as a function of the slip ratio where changing the coefficients in the equation changes the shape of the curve and the tyre characteristics.

$$F_{t,x} = D_F \sin \left(C \arctan \left[B_F(1 - E_F)(\sigma_F + S_{h,F}) + E_F \arctan \left(B_F(\sigma_F + S_{h,F}) \right) \right] \right) + S_v$$ \hspace{1cm} (51)

A typical curve is illustrated in Figure 3-3 below. Essentially a slip ratio of -1 means full braking lock, a ratio of 0 means the tyre is spinning at the same rate as the road is being covered and a slip ratio of 1 means the wheels are spinning with a zero longitudinal velocity of the vehicle.
Figure 3-3: An example characteristic of tyre longitudinal force as a function of slip ratio

The steady-state longitudinal force of the tyre is not directly applied to the road as the tyre is not a rigid body. Therefore the tyre can be deformed and as such a torque on the inner rim inputted from the wheel hub is not applied to the road instantly and is subject to a delay. The tyre delay, $\tau_{RL}$, is modeled as a first order differential equation with the tyre relaxation length, $S_{RL}$, and is inversely proportional to the vehicle speed.

The tyre relaxation length is the distance the tyre needs to cover to generate 63% of the steady state longitudinal force.

The delayed slip ratio, $\sigma_d$, is calculated through,

$$\sigma = \tau_{RL} \dot{\sigma}_d + \sigma_d$$  \hspace{1cm} (52)

The above equation is then integrated to give the final longitudinal delayed slip ratio,

$$\sigma_d = \int \frac{V_{act}}{S_{RL}} (\sigma + \sigma_d) \, dt$$  \hspace{1cm} (53)

The previous section has detailed the equations used to model the vehicle, from the differential to the tyre contact patch including the suspension forces and pitch dynamics. The equations governing the powertrain will be explained in the next section.

3.3 DRIVETRAIN MODEL

The main transmission adopted for this research was a two-speed prototype transmission for electric drivetrains along with its single-speed equivalent.

3.3.1 SINGLE-SPEED TRANSMISSION

The single-speed transmission was developed by Oerlikon Graziano and Vocis Drivelines for electric drivetrains. It adopts a dual-stage spur gear reduction and consists of a helical gear for the main gear then a second helical gear for the final drive which is connected to the half-shafts through an open differential. The transmission is illustrated in Figure 3-4.
The schematic of the single-speed transmission is shown in Figure 3-5, illustrating the two shaft design, the forces at the gears, the efficiencies and the moments of inertias. The equations are derived from this diagram and are explained below. The electric motor air gap torque is modelled as a first order transfer function to account for the motor response and delay. The slew rate and the windage torque are also accounted for through this transfer function.

The torque balance equation about the primary shaft, is given in equation (54). The subscript, s, denotes single-speed, the two-speed transmission variables in the next section do not have a subscript as they are used extensively in the report. $T_{m,det}$ is the delayed motor torque, $F_{1s}$ is the force transmitted by the input gear set, $R_{1s}$ is the radius of the main gear set input gear, $J_m$ is the moment of inertia of the electric motor, $J_{1s}$ is the primary shaft moment of inertia and $\ddot{\theta}_{1s}$ is the angular acceleration of the primary shaft.

$$T_{m,det} - F_{1s}R_{1s} = (J_m + J_{1s})\ddot{\theta}_{1s}$$ (54)

The torque balance equation about the secondary shaft, is given in equation (55). $F_{2s}$ is the force transmitted to the final drive set, $R_{3s}$ is the radius of the final drive gear set input gear, $R_{2s'}$ is the radius of the main gear set output gear, $\eta_{1s}$ is the main gear efficiency, $J_{2s'}$ is the secondary shaft moment of inertia and $\ddot{\theta}_{2s'}$ is the angular acceleration of the secondary shaft.

$$F_{2s}R_{3s} - F_{1s}R_{2s}\eta_{1s} = J_{2s'}\ddot{\theta}_{2s'}$$ (55)

The torque balance equation about the secondary shaft, is given in equation (56). $R_{4s}$ is the radius of the final drive gear set output gear, $\eta_{2s}$ is the final drive efficiency, $J_{3s}$ is the
moment of inertia of the final drive shaft, \( J_{hS_L} \), is the moment of inertia of the half-shaft and \( \ddot{\theta}_{3S} \) is the angular acceleration of the final drive shaft.

\[
F_z R_4 \dot{\eta}_2 S - T_{hS_L} - T_{hS_R} = \left( J_{3S} + \frac{1}{2} (J_{hS_L} + J_{hS_R}) \right) \ddot{\theta}_{3S} \tag{56}
\]

The resultant differential acceleration, \( \ddot{\theta}_{diff_S} \), is given by equation (57). The terms cancel out using the ratio of gear radius' to give the gear ratios where, \( \tau_{1S} \) is the main gear ratio and \( \tau_{diff_S} \) is the final drive ratio.

\[
\ddot{\theta}_{diff_S} = \frac{T_{m,del} \tau_{1S} \tau_{diff_S} \eta_{1S} \eta_{2S} - T_{hS_L} - T_{hS_R}}{\left[ (J_{motor} + J_{1S}) \tau_{1S}^2 \tau_{diff_S} \eta_{1S} \eta_{3S} + J_{2S} \tau_{diff_S}^2 \eta_{3S} + J_{3S} + \frac{1}{2} (J_{hS_L} + J_{hS_R}) \right]} \tag{57}
\]

Due to the single-speed transmission not containing any clutches and if the torsion of the shafts and the dynamics of the differential are not considered, the electric motor and transmission system can be considered as a single degree of freedom. The half-shafts are modelled as torsional dampers which represent an additional degree of freedom and kinematically separate the drivetrain from the wheels as shown in equations (58) and (59). \( K_{hS_L} \) is the half-shaft stiffness and \( \beta_{hS_L} \) is the damping coefficient of the half-shaft.

\[
T_{hS_L} = \beta_{hS_L} \left( \dot{\theta}_{diff} - \dot{\theta}_{w_1} \right) + K_{hS_L} (\theta_{diff} - \theta_{w_1}) \tag{58}
\]

\[
T_{hS_R} = \beta_{hS_R} \left( \dot{\theta}_{diff} - \dot{\theta}_{w_r} \right) + K_{hS_R} (\theta_{diff} - \theta_{w_r}) \tag{59}
\]

### 3.3.2 TWO-SPEED TRANSMISSION

The main focus of the research is based on a novel two-speed transmission (2SED) using the same twin shaft layout of the single-speed variant described above.

The two-speed transmission system is unique in its design through combining the simplistic and efficient dual layshaft layout with a high quality clutch-to-clutch gearshift. The primary components are multi-plate dry friction clutch on the end of the primary shaft, the one-way sprag clutch on the secondary shaft and the open differential. Essentially, in first gear the torque is passed through a positively engaged sprag clutch with the friction clutch whilst in second gear torque is passed through the friction clutch and the sprag clutch is overrun. The friction clutch is applied allowing the sprag clutch to over-run to achieve an upshift whilst the friction clutch is disengaged during a downshift to engage the sprag clutch. Upshifts and downshifts consist of torque phases and inertia phases similar to a DCT and require careful control of the friction clutch to ensure good drivability. The gearshifts are fully automated so the transmission can operate as an Automated Manual Transmission (AMT) or a fully Automated Transmission (AT) using gearshift maps to provoke gear changes. A full description of the gearshift methodology is presented later in the thesis, including a description of the modeling method employed.
The dry multi-disc friction clutch utilizes a sintered metal friction material and is electro-hydraulically controlled by a remote brushless motor-driven actuator, pressurizing a master cylinder mechanically connected to the Belleville spring of the friction clutch. The actuator displacement is used to provide a feedback loop to control the friction clutch actuator. Due to the sprag clutch overrunning for a negative torque input by design, a locking ring is engaged when the transmission is in first gear. This allows the vehicle to travel in reverse in first gear, but more importantly regenerate energy during braking.

The novel two-speed transmission has the park-lock function inherent in its design through engaging the locking ring and closing the friction clutch when the vehicle is stationary. The mechanical layout of the two-speed transmission results in a compact design, with the...
overall distance between the primary and differential shaft being about 200mm for a premium passenger car. Furthermore, for this application the distance from the primary shaft to the secondary shaft is less than 110mm and from the secondary shaft to the differential is approximately 125mm. The transmission with this layout results in a mass of about 38 kg. The dimensions and weight are comparable with those of the single-speed unit (mass, 25 kg) from which this novel transmission was derived.

Figure 3-8: Novel two-speed transmission schematic illustrating variables adopting for the governing equations.

The electric motor is connected directly to the primary shaft with no clutch damper or torque converter, the system can be considered a single degree-of-freedom system in fixed gear condition. Therefore there is only one moment balance equation to be derived for each gear, however to model the gearshift dynamics properly further equations will need to be derived. In this section only the fixed gear equations are considered and explained below.

The equations to describe the torque transfer in first gear are listed first.

First gear

The torque balance equation about the primary shaft, is given in equation (60). $F_1$ is the force transmitted by first gear set, $R_1$ is the radius of the first gear set input gear, $J_1$ is the moment of inertia of the primary shaft and $\ddot{\theta}_1$ is the angular acceleration of the primary shaft.

$$T_{m, det} - F_1 R_1 = (J_m + J_1) \ddot{\theta}_1 \tag{60}$$

The torque balance equation about the dry clutch, is given in equation (61), $F_1$ is the force transmitted by second gear set, $R_2$, is the radius of the second gear input gear (which is on the friction clutch output), $\eta_2$ is the second gear efficiency, $J_{1b}$ is the moment of inertia of
the friction clutch/gear assembly and $\dot{\theta}_{1b}$ is the rotational acceleration of the friction clutch output gear.

$$F_2 R_2 \eta_2 = J_{1b} \dot{\theta}_{1b}$$  (61)

The torque balance equation about the sprag clutch shaft, is given in equation (62). $R_4$ is the radius of the first gear set output gear, $T_{C2}$ is the sprag clutch output torque, $J_{2b}$ is the moment of inertia of the sprag clutch/gear assembly and $\dot{\theta}_{2b}$ is the rotational acceleration of the sprag clutch/gear assembly.

$$F_1 R_4 \eta_2 - T_{C2} = J_{2b} \dot{\theta}_{2b}$$  (62)

The torque balance equation about the secondary shaft, is given in equation (63)(62). $R_3$ is the radius of the second gear set output gear, $F_3$ is the force transmitted by final drive gear set, $R_5$ is the radius of the final drive gear set input gear, $J_2$ is the moment of inertia of the secondary shaft and $\dot{\theta}_2$ is the rotational acceleration of the secondary shaft.

$$T_{C2} - F_2 R_3 \eta_2 - F_3 R_5 = J_2 \dot{\theta}_2$$  (63)

The torque balance equation about the differential shaft, is given in equation (64). $R_6$ is the radius of the final drive gear set output gear, $\eta_3$ is the final drive efficiency.

$$F_3 R_6 \eta_3 - T_{h_{SL}} - T_{h_{SR}} = J_3 \dot{\theta}_3$$  (64)

The resultant differential acceleration in first gear is then given in equation (65). Again, the terms cancel out using the ratio of gear radius' to give the gear ratios where, $\tau_1$ is the first gear ratio, $\tau_2$ is the second gear ratio and $\tau_{diff}$ is the final drive ratio.

$$\ddot{\theta}_{diff} = \frac{T_{m,del} \tau_1 \tau_{diff} \eta_1 \eta_3 - T_{h_{SL}} - T_{h_{SR}}}{\left[(J_{mot} + J_1) \tau_1^2 \tau_{diff}^2 \eta_1 \eta_3 + (J_2 + J_{2b}) \right]} \left[\tau_2^2 \eta_3 + J_3 + \frac{1}{2} (J_{h_{SL}} + J_{h_{SR}}) \right]$$  (65)

Second gear

The torque balance equation about the primary shaft, is given in equation (66).

$$T_{m,del} - F_1 R_1 - F_2 R_2 = (J_m + J_1 + J_{1b}). \ddot{\theta}_1$$  (66)

The dry clutch shaft moment balance equation is not required as it is locked to $J_1$ and taken into account in the above calculation.

The torque balance equation about the secondary shaft, is given in equation (67).

$$F_2 R_5 \eta_2 - F_3 R_5 = J_2 \dot{\theta}_2$$  (67)

The torque balance equation about the sprag clutch which spins freely on the secondary shaft due to the over run, is given in equation (68).
Simulation model development: Drivetrain model

\[ F_1 \cdot R_4 \cdot n_1 = J_{2b} \cdot \ddot{\theta}_{2b} \]  \hspace{1cm} (68)

The torque balance equation about the differential shaft, is given in equation (69).

\[ F_3 \cdot R_6 \cdot \eta_3 - T_{h_{SL}} - T_{h_{SR}} = J_3 \cdot \ddot{\theta}_3 \]  \hspace{1cm} (69)

The resultant differential acceleration in second gear is given by equation (70).

\[ \ddot{\theta}_{\text{diff}} = \frac{T_e \cdot \tau_2 \cdot \tau_{\text{diff}} \cdot \eta_2 \cdot \eta_3 - T_{h_{SL}} - T_{h_{SR}}}{(J_{\text{mot}} + J_1 + J_{1b} + J_{2b}) \cdot \tau_2 \cdot \tau_{\text{diff}} \cdot \eta_2 \cdot \eta_3 + (J_2 \cdot \tau_{\text{diff}} \cdot \eta_3 + J_3 + \frac{1}{2}(T_{h_{SL}} + T_{h_{SR}}))} \]  \hspace{1cm} (70)

The differential accelerations in both gears can now be found, which are used in the spring/damper equations to calculate the half-shaft torque, which is the input to the unsprung mass equations.

The only unknowns in the differential acceleration equations are the in-gear and differential efficiencies. The efficiency of the driven gear is a function of the input torque (either the primary shaft torque, \( T_{\text{primary shaft}} \), or secondary shaft torque, \( T_{\text{secondary shaft}} \)), input speed (either the primary shaft speed, \( \dot{\theta}_{\text{primary shaft}} \), or secondary shaft speed, \( \dot{\theta}_{\text{secondary shaft}} \)) and transmission temperature, \( T_{\text{Temp trans}} \).

\[ \eta_{\text{diff}} = \eta_3 = \eta_3(T_{\text{secondary shaft}}, \dot{\theta}_{\text{secondary shaft}}, T_{\text{Temp trans}}) \]  \hspace{1cm} (71)

\[ \eta_{\text{gear}} = \eta_{1/2} = \eta_{1/2}(T_{\text{primary shaft}}, \dot{\theta}_{\text{primary shaft}}, T_{\text{Temp trans}}) \]  \hspace{1cm} (72)

![2 SED - 1st Gear Gearbox efficiency map at 20 degrees (Viscosity 90 cSt)](image)

Figure 3-9: 2SED gearbox efficiencies at 20 degrees Celsius
Efficiency maps for the transmission and differential in each gear were provided as 2-D efficiency maps by Oerlikon Graziano, with torque and speed as the inputs, taken at different transmission temperatures. Therefore 3-D efficiency maps were generated with each of the 2-D efficiency maps representing a layer for each temperature. A temperature model of the transmission/differential was created in Simulink to provide an input, as well as torque and speed, to the 3-D efficiency maps. The transmission efficiency is higher in second gear than first gear due to the relative size of the physical gears, as first gear ratio is larger than second gear ratio, which reduces the meshing efficiency.

Essentially the transmission temperature is affected by three different factors. Firstly, the power losses in the transmission due to heat, friction, churning and hydrodynamic losses which are modelled as a function of the efficiency map. Secondly, the heat exchange between the transmission and the environment due to air flowing over the transmission providing a cooling effect. Lastly, the heat exchange between the transmission and the electric motor where thermal energy can flow in both directions depending on which component is hottest.

The basis for the temperature model is thermodynamics first law, “In a thermodynamic process involving a closed system, the increment in the internal energy is equal to the difference between the heat accumulated by the system and the work done by it”. Through considering the power loss in the transmission and the heat exchange as explained previously the transmission temperature can be found through integrating equation (73). $P_{loss,trans}$ is the power loss in the transmission, $\dot{Q}_{trans,motor}$ is the heat exchange between the transmission and the motor, $\dot{Q}_{trans,air}$ is the heat exchange between the transmission and the environment, $C_{P,steel}$ is the specific heat capacity of steel, $m_{steel}$ is the steel mass in the transmission, $C_{P,alu}$ is the specific heat capacity of aluminium, $m_{alu}$ is the mass of aluminium in the transmission, $C_{P,oil}$ is the specific heat capacity of oil and $m_{oil}$ is the mass of oil in the transmission.
Due to the simulation considering the gearbox and differential efficiencies separately through independent look up tables, the total transmission power loss is considered as a summation of power loss in the gearbox, $P_{\text{loss, gearbox}}$, and the power loss in the differential, $P_{\text{loss, diff}}$. The losses are combined for the temperature model of the transmission too as the gearbox and differential share the same oil and sit in the same case.

$$P_{\text{loss,trans}} = P_{\text{loss, gearbox}} + P_{\text{loss, diff}}$$ (74)

The power loss, $P_{\text{loss,i}}$, is calculated through considering the power output, $P_{\text{out,i}}$, of the gearbox or differential multiplied by one minus the efficiency, $\eta_i$, as shown in equation (75).

$$P_{\text{loss,i}} = P_{\text{out,i}}(1 - \eta_i)$$ (75)

The specific heats and masses of the materials are fixed values however the heat exchanges from the transmission to motor (or vice versa) and from the transmission to the environment (or vice versa) need to be defined.

The heat transfer between the transmission and the electric motor is calculated through the heat transfer surface area between the transmission and the motor, $A_{\text{trans,motor}}$, the difference between the transmission temperature and the electric motor temperature, $\Delta T_{\text{motor}}$, and the heat transfer coefficient between the transmission and the electric motor, $\alpha_{\text{trans,motor}}$.

$$Q_{\text{trans,motor}} = -\alpha_{\text{trans,motor}} A_{\text{trans,motor}} (\Delta T_{\text{trans}} - \Delta T_{\text{motor}})$$ (76)

The heat transfer coefficient is considered as the reciprocal of the total thermal resistance between the transmission and the motor. There are three heat transfer processes between the transmission and the motor, two of them being convective and one being conductive. The conductive process takes place in the wall of the transmission, a function of the separation surface thickness and thermal conductivity, whereas the convective processes are between the motor or transmission and the separation surface, each with their own heat transfer coefficients.

The total thermal resistance can be found through summing the thermal resistance of the three heat transfer processes, as shown in equation (77). $\lambda$ is the separation surface thermal conductivity, $S$ is the separation surface thickness, $h_{\text{motor}}$ is the heat transfer coefficient between the motor and the separation surface and $h_{\text{trans}}$ is the heat transfer coefficient between the transmission and the separation surface.

$$\alpha_{\text{trans,motor}} = \frac{1}{\frac{1}{h_{\text{motor}}} + \frac{S}{\lambda} + \frac{1}{h_{\text{trans}}}}$$ (77)
Simulation model development: Drivetrain model

The heat transfer coefficient between the motor and the separation surface is considered a constant value whereas the heat transfer coefficient between the transmission and the separation surface is a function of the primary shaft speed. This is illustrated in Figure 3-11 below.

The heat exchange between the transmission and the environment is calculated in the same way as the heat transfer between the transmission and the motor as shown in equation (78). Considering the heat transfer surface area of the transmission, $A_{trans,air}$, the difference between the transmission temperature and the air temperature, $Temp_{air}$, and the heat transfer coefficient between the transmission and the electric motor, $\alpha_{trans,air}$.

$$Q_{trans,air} = -\alpha_{trans,air} \cdot A_{trans,air} (Temp_{trans} - Temp_{air})$$

(78)

The heat transfer coefficient between the transmission and the environment is found in the same way as before as the reciprocal of the total thermal resistance. The heat transfer coefficient between the transmission and the separation surface is considered a constant. The heat transfer coefficient between the environment and the separation surface however is a function of the vehicle speed, with increased speed providing a higher cooling effect due to more air passing over the transmission, as shown in Figure 3-12.
To accurately model the transmission temperature it is necessary to know the electric motor temperature, specifically to calculate the heat transfer between the transmission and the electric motor. The electric motor thermal model is modelled in a similar way to the transmission, considering the power loss in the motor, $P_{\text{loss,motor}}$, however the electric motor is water cooled. Therefore three heat transfer parameters are needed, the heat transfer between the motor and the transmission, the heat transfer between the motor and the environment, $\dot{Q}_{\text{motor,air}}$, and the heat transfer between the motor and the cooling water, $\dot{Q}_{\text{cooling}}$, as seen in equation (79). The term $C_{p,motor}$ is the specific heat capacity of the motor and $m_{motor}$ is the mass of the electric motor.

$$\frac{dT_{\text{motor}}}{dt} = \frac{P_{\text{loss,motor}} - Q_{\text{trans,motor}} + \dot{Q}_{\text{motor,air}} + \dot{Q}_{\text{cooling}}}{C_{p,motor} \cdot m_{motor}} \quad (79)$$

The power loss in the motor is a function of the electric motor efficiency map, however due to a PMSM motor with an inverter being used in this investigation the efficiency map includes the losses for both the inverter and motor. Therefore a constant needs to be included to separate the losses of the motor from the total motor plus inverter losses.

The heat transfer between the motor and the transmission is already known from equation (76). The heat transfer from the motor to the environment is calculated in the same way as the transmission and is shown in equation (80). Considering the heat transfer surface area of the motor, $A_{\text{motor,air}}$, the difference between the transmission temperature and the air temperature, and the heat transfer coefficient between the motor and the environment, $a_{\text{motor,air}}$.

$$\dot{Q}_{\text{motor,air}} = -a_{\text{motor,air}} A_{\text{motor,air}} (T_{\text{motor}} - T_{\text{air}}) \quad (80)$$

The cooling water to motor heat transfer is modeled in a similar way but considers the water specific heat capacity, $C_{p,H_2O}$, water density, $\rho_{H_2O}$, and flow rate, $\dot{q}_{H_2O}$, plus the difference between the motor temperature and the water temperature, $T_{\text{temp,H_2O}}$, as shown in equation (81).

$$\dot{Q}_{\text{cooling}} = -C_{p,H_2O} \rho_{H_2O} \dot{q}_{H_2O} (T_{\text{motor}} - T_{\text{temp,H_2O}}) \quad (81)$$

The required electric motor torque, $T_{\text{motor}}$, and speed, $\omega_{\text{motor}}$, give the electric motor output power and with the motor efficiency map the power at the inverter, $P_{\text{Inv.(Lim).TRACTION/REGENERATION}}$, is found (as shown below), where the motor efficiency, $\eta_{\text{motor}}$, is reversed for either traction or regeneration.

$$P_{\text{Inv.(Lim).TRACTION}} = \frac{1}{\eta_{\text{motor}}} (T_{\text{motor}} \omega_{\text{motor}}) \quad (82)$$

$$P_{\text{Inv.(Lim).REGENERATION}} = \eta_{\text{motor}} (T_{\text{motor}} \omega_{\text{motor}}) \quad (83)$$

However, to analyse the energy consumption of an electric vehicle it is necessary to know the state-of-charge of the battery and therefore to model the losses in the battery. During traction the battery losses will be considered to give the actual power drain on the battery.
and during regenerative braking the losses will be subtracted to give the actual battery charge. The losses in the battery are due to the resistances in the battery which generate heat.

The state of charge of a battery module, $SOC$, is given by equation (84) below, where $SOC = 1$ when full. $Q_{Battery}$ is the battery capacity.

$$SOC = 1 - \frac{1}{Q_{Battery}} \int_0^t \beta(T_{Battery}) \cdot i_{batt}(t) \, dt$$  \hspace{1cm} (84)

The state of charge is a function of the battery current, $i_{batt}$, and a constant, $\beta$, which is a function of the battery temperature, $T_{Battery}$. The constant, $\beta$, is a multiplicative factor which is altered depending on the battery type and is modelled as a one dimensional look-up table. The characteristics are illustrated in Figure 3-13.

![Figure 3-13: Temperature factor of the battery](image)

The unknowns are therefore the battery temperature and the battery current. The battery power is found through the multiplication of the current and the voltage. The battery voltage, $V_{batt}$, is found through equation (85), using ohm’s law.

$$V_{batt} = (V_{batt} - i_{load} \cdot R_{1,batt}) + i_{batt} \cdot R_{1,batt}$$  \hspace{1cm} (85)

The previous time steps current, $i_{load}$, is found from the previous simulation loops battery voltage and the total battery ohm resistance, $R_{1,batt}$, as the actual battery current and voltage cannot be computed in the time step.

The limited battery power, $P_{inv,(lim)}$, can then be found from multiplying equation (85) by the battery current to equal the inverter power as shown in equation (86).

$$P_{inv,(lim)} = V_{batt} \cdot i_{batt} = (V_{batt} - i_{load} \cdot R_{1,batt}) \cdot i_{batt} + i_{batt}^2 \cdot R_{1,batt}$$  \hspace{1cm} (86)

Equation (86) forms a quadratic equation that can be solved for the battery current as shown in equation (87).
Simulation model development: Drivetrain model

\[ i_{\text{batt}} = -\left( V_{\text{batt}} - i_{\text{load}} R_{1,\text{batt}} \right) + \sqrt{\Psi \left( \left( V_{\text{batt}} - i_{\text{load}} R_{1,\text{batt}} \right)^2 + 4 R_{1,\text{batt}} P_{\text{inv.(Lim)}} \right)} \]
\[ \frac{2 R_{1,\text{batt}}} \] (87)

The variable, \( \Psi \), is given a value of 0 if the term within the square root is less than zero, i.e. \( (V_{\text{batt}} - i_{\text{load}} R_{1,\text{batt}})^2 + 4 R_{1,\text{batt}} P_{\text{inv.(Lim)}} \).

The unknown in equation (87) is now the battery voltage, which can be considered as an individual battery module voltage multiplied by the number of modules. Each module is modeled as having two resistor banks in series and a capacitor, with the capacitor in parallel with the second resistor bank.

The module battery voltage, \( V_{\text{module}} \), can be found through the module equilibrium potential, \( E_{\text{module}} \), the potential loss across resistor bank one, \( V_{R1,\text{module}} \), and the potential loss across the capacitor and resistor bank 2, \( V_{R2/C,\text{module}} \), as shown in equation (88).

\[ V_{\text{module}} = E_{\text{module}} - V_{R1,\text{module}} - V_{R2/C,\text{module}} \] (88)

Each module equilibrium potential is simply the total battery equilibrium potential divided by the number of modules. The total battery equilibrium potential, \( E_{\text{battery}} \), is found through equation (89),

\[ E_{\text{battery}} = E_{NC} + \Delta E \] (89)

The battery equilibrium potential, \( E_{NC} \), is initially found by a look-up table depending on the battery state of charge. And corrected by a second term, \( \Delta E \), which depends on the battery temperature.

![Figure 3-14: Battery equilibrium potential (initial)](image1)

![Figure 3-15: Equilibrium potential correction term](image2)
Simulation model development: Drivetrain model

The two potential losses are found through Ohm’s law, considering the battery current and internal resistances as shown in equation (90) for the potential loss across the first resistor bank. $R_1$ is the resistance in series with both the capacitor and the resistance.

\[ V_{R1,module} = i_{batt} \cdot R_1 \] (90)

The potential loss across the capacitor and resistor bank 2 is more complicated considering the sum of the currents through both components as shown in equation (91), which can simply be integrated to find the voltage. $i_c$ is the current through the capacitor, $i_{R2}$ is the current through the second resistor, $R_2$ is the resistance of the second resistor, $C$ is the capacitance of the capacitor.

\[ V_{R2/C,module} = i_c + i_{R2} = \frac{V_{R2/C,module}}{R_2} - C \frac{dV_{R2/C,module}}{dt} \] (91)

The thermal model of the battery is similar to the transmission and motor thermal model, using the battery mass, $m_{battery}$, specific heat capacity, $C_p,battery$, heat transfer coefficient, $h_{c,battery}$, surface area of the battery, $A_{battery}$, temperature differential between the battery temperature and air temperature and battery power loss, $P_{loss,battery}$. The power loss of each module, $P_{loss,mod}$, is simply the sum of the squared potential losses in the battery divided by the corresponding resistance, as shown in

\[ P_{loss,mod} = \frac{V_{R1,module}^2}{R_1} + \frac{V_{R2/C,module}^2}{R_2} \] (92)

The total battery power loss is each individual battery module loss multiplied by the number of modules. The battery temperature can then be found through the thermal balance equation, equation (93), below.

\[ m_{battery}C_p,battery \frac{dT_{battery}}{dt} = P_{loss,battery} - h_{c,battery}A_{battery}(T_{battery} - Temp_{air}) \] (93)

The input to the model electric motor torque which is found through a ‘driver model’ which outputs a throttle demand, or more specifically a percentage of the maximum available torque at the current motor speed. The driver model is a controller developed to simulate the driver response to a required manoeuvre, be it an acceleration test, tip in test or driving cycle. The driver model is broken into two parts, the driver demand controller for a positive torque simulated as a throttle position and the braking controller when a braking torque is required.

To simulate performance tests the driver model simply inputs a fixed percentage of throttle at a set time with a set ramp rate. However, to follow driving cycles the difference between the vehicles actual and a prescribed reference speed is the input to a series of controllers. There are two main controllers, one for the throttle and one for the brakes to slow the vehicle.
Initially a simple feedback controller was adopted for both the throttle and braking controllers, but after some experimenting it was found that this was insufficient for following driving cycles, specifically the vehicle would not smoothly follow the reference speed. If high gains were used in the PID then the vehicle followed the reference speed closely but would result in high frequency oscillations. If too low gains were used then the vehicle struggled to follow the reference speed. A feed-forward controller was added which used the reference speed to calculate a required driven wheels force, an estimate of the gearbox and differential efficiency to approximate the required engine torque and therefore throttle position. Any error was then corrected through summing the feed-forward and feedback controller outputs with specific gains. The driver demand controller also includes a dead zone, so if the vehicle speed error is very small, typically -0.5 to 0.5 km/h the feedback controller is not used. A weighted moving average and transfer function are also included to smooth the required throttle position.

The braking controller calculates the required braking torque to reduce the speed of the vehicle to follow the reference speed. The vehicle is powered by an electric motor so it has a certain amount of regenerative torque. Therefore if the required braking torque is within the regenerative limit of the motor, the model purely uses regenerative torque to brake the vehicle. If a greater torque is required the additional braking torque is fed into the moment balance equations for each wheel, at a value related to the front and rear braking bias. A weighted moving average is also included to smooth the required braking torque.

An addition to the driver model is a traction controller which limits the throttle position if the slip ratio becomes too great.

3.4 RESULTS

The two models using the same vehicle model but different transmissions, the single-speed and two-speed, were used to simulate different manoeuvres to compare the two systems. The vehicle model needs to be parameterised to test a case study vehicle. The vehicle data used is based around a test vehicle being manufactured by Vocis Drivelines and Oerlikon Graziano based on a Mercedes Vito. The electric motor being used to drive the vehicle is a 70 kW PMSM developed by Zytek Ltd for this specific application. The torque characteristics and efficiency map of the electric motor were provided by Zytek and are show below in Figure 3-16.

An additional variable required by the model is the gearshift map for the two-speed transmission. The single-speed transmission has a single fixed gear whereas the 2SED has two gears and therefore requires a method to ascertain which gear to be in at a given time. To do this gearshift maps are utilised which are two dimensional look-up tables that depend on the vehicles speed and drivers throttle demand as inputs to determine when to change gear.
Simulation model development: Results

A “standard map” was supplied by the industrial partners of this research, however due to large impact the shift map has on both vehicle performance and energy consumption two additional gearshift map variants were tested. Each of the gearshift maps are presented below in Figure 3-17.

The additional vehicle data such as mass, aerodynamic drag coefficient, frontal area, wheel span, etc, are given in Appendix B.

The results of performance tests will be presented first, followed by an analysis into energy consumption over standard driving cycles and finally a brief investigation into vehicle drivability. The results presented in this section will give some indication of the benefits of each transmission and allow some conclusions to be made.
3.4.1 VEHICLE PERFORMANCE

The vehicle model was used to simulate full throttle acceleration tests to analyse the vehicles performance with each transmission, the single-speed and two-speed (2SED). The vehicle speed traces for the full throttle acceleration test are shown in Figure 3-18.

The first observation is that the vehicle achieves a higher top speed when using the two-speed transmission rather than the single-speed transmission. The second observation is that the speed increases at a faster rate with the two-speed transmission than with the single-speed transmission. Both of these observations are due to the differences in the gear ratios between the two transmissions.

The vehicle accelerates faster due to the first gear in the 2SED being higher than the single-speed transmission so there is a comparative increase in wheel torque between the two case studies. However, the increased torque with the two-speed in first gear compared to the single-speed results in larger oscillations which affect the level of traction and will marginally increase the acceleration times. The standard gearshift map is utilised which defines an upshift speed of 45 km/h at 100% throttle and at this point it is visible to see the vehicle speed gradient change as the wheel torque reduces significantly with the shift to second gear. Second gear of the 2SED is lower than first gear so the available wheel torque is lower, although this is only the case if the shift happens during the field weakening section of the electric motor torque curve. However, due to the characteristics of the electric motor and with the shift taking place in the constant power section of the electric motor torque curve the vehicle acceleration does not reduce, as shown in Figure 3-19. The lower value of the 2SED second gear than the single-speed also promotes a typically higher vehicle top speed as the maximum motor speed translates to a higher vehicle speed. However, the resistive torque due to aerodynamic drag and rolling resistance need to be overcome and become and are more significant at higher speeds so the top speed of the 2SED case study vehicle is limited by this fact.

The full throttle acceleration test was repeated with the two gearshift map variations, gearshift maps 1 & 2. Gearshift maps 1 & 2 are characterised by a different upshift speed at 100% throttle, 60 km/h instead of 45 km/h. The effect of this is clearly seen in Figure 3-20, where there is a difference in acceleration, due to the difference in wheel torque by
changing gear later. It is clear that due to the torque profile of the 70 kW motor and the gear ratios in the 2SED that a higher wheel torque can be achieved by shifting at a lower vehicle speed as the available torque in 2\textsuperscript{nd} gear is higher than that available in 1\textsuperscript{st} gear.

The ideal shift point can in fact be ascertained by overlapping the maximum available wheel torque in first and second gear. Where the lines overlap gives the optimal shift point, as it indicates where the vehicle can shift gear to maximise the available wheel torque through the transmission.

The full set of results from the 100 \% throttle acceleration tests with the single-speed and the two-speed with the three different gearshift maps are shown in Table 3-1. The results give numerical proof of the conclusions drawn from the previous graphs.
The two-speed vehicle has consistently lower acceleration times than the single-speed vehicle, especially below 45 km/h where the two-speed vehicle is in first gear. The 10-50 km/h times are significantly better than the single-speed vehicle for the two-speed vehicle where it is in first gear, i.e. 5.4%. Whereas, the 40-100 km/h times are similar for the two vehicles as the second gear ratio is lower than the single-speed gear ratio and the single-speed vehicle can ‘catch up’. The maximum vehicle speed is much higher for the two-speed vehicle.

The vehicle model is also capable of simulating driving cycles to measure the energy consumption, either at the motor or at the battery through the comprehensive battery model. Simulating a driving cycle differs from an acceleration test as the acceleration test simply requires a 100% throttle signal at the start point, however the driving cycle utilises the driver model which is a controller to regulate the throttle to maintain the desired vehicle speed, as you do when driving your car.

The vehicle speed trace for the NEDC is shown in Figure 3-21, where the simulated vehicle speed follows the reference velocity very accurately and proves the model to be capable of following drive cycle speed profiles. Specifically, in this case driver simulator keeps the simulated velocity profile within the boundaries for the NEDC, +/- 2kph during the constant velocity phases and +/- 4kph during transients. This is on the limit during the final high speed phase of the NEDC as seen in Figure 3-21.
Four standard driving cycles were simulated with the single-speed and two-speed transmission for each gearshift map and the results are given in Table 3-2.

Table 3-2: Driving cycle battery energy consumption, single and two-speed.

<table>
<thead>
<tr>
<th>Gearshift Map</th>
<th>NEDC</th>
<th>FTP 75</th>
<th>SC03</th>
<th>UDDS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Unladen</td>
<td>2.961</td>
<td>-</td>
<td>1.589</td>
<td>-</td>
</tr>
<tr>
<td>Laden</td>
<td>3.713</td>
<td>-</td>
<td>2.074</td>
<td>-</td>
</tr>
<tr>
<td>STD Two speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Unladen</td>
<td>3.089</td>
<td>4.30%</td>
<td>1.575</td>
<td>-0.88%</td>
</tr>
<tr>
<td>Laden</td>
<td>3.848</td>
<td>3.65%</td>
<td>2.066</td>
<td>-0.36%</td>
</tr>
<tr>
<td>1 Two speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Unladen</td>
<td>3.071</td>
<td>3.70%</td>
<td>1.559</td>
<td>-1.91%</td>
</tr>
<tr>
<td>Laden</td>
<td>3.826</td>
<td>3.05%</td>
<td>2.053</td>
<td>-1.03%</td>
</tr>
<tr>
<td>2 Two speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Unladen</td>
<td>3.102</td>
<td>4.76%</td>
<td>1.567</td>
<td>-1.38%</td>
</tr>
<tr>
<td>Laden</td>
<td>3.852</td>
<td>3.74%</td>
<td>2.058</td>
<td>-0.78%</td>
</tr>
</tbody>
</table>

The results show that for the NEDC the single-speed has a lower energy consumption that the two-speed transmission for all gearshift map combinations. The results also show that the energy consumption increases with gearshifts maps 1 and 2 over the standard gearshift map, so having an upshift at a higher speed increases the energy consumption. This makes sense as upshifting early will promote a larger torque demand on the motor pushing the operating point into an area of higher efficiency.

In contrast, the energy consumption is lower for the two-speed transmission for the other three driving cycles, for all gearshift maps. The literature reviewed in chapter 2 pointed to a reduction in energy consumption for multiple-speed transmissions over single-speed transmissions.

The NEDC differs from the FTP 75, SC03 and UDDS in that it has a lower average velocity and as such the two-speed gear ratios may not be beneficial this driving cycle.
This chapter has explained the development of the novel vehicle and powertrain models. The models are built in the Matlab/Simulink environment out of equations derived from analyzing the vehicle as a system.

The base vehicle model consists of fifteen degrees of freedom, the vertical, longitudinal and rotational movement of the sprung mass and the vertical, longitudinal and rotational movement of each of the four unsprung masses (the wheels). The model considers the resistive forces due to the road grade, aerodynamic resistance and rolling resistance. The suspension dynamics are modeled as spring and dampers and allow the analysis of the vehicle pitch and drivability. Pacejka’s Magic Formula is included to model the tyre slip and improves the quality of the acceleration test results along with the modeling of the half-shaft torsional dynamics and powertrain inertias.

The comprehensive vehicle model is a significant achievement in itself and goes well beyond the complexity of the models adopted in the literature to analyse electric vehicle drivetrain architectures, e.g. Knödel et al. (2010).

The vehicle model was parameterized to match a prototype electric vehicle under development by the industrial partners including the two transmissions. Performance tests and driving cycles were simulated to compare the benefits of the single-speed and two-speed transmissions. The results have shown the two-speed to have significant advantages over the single-speed both in terms of performance and energy consumption.
4 TWO-SPEED TRANSMISSION GEARSHIFT DYNAMICS

4.1 INTRODUCTION

The single-speed transmission represents a single degree-of-freedom and as such can be modelled simply through one governing equation, not accounting for shaft torsion or gear play. The two-speed transmission (2SED) requires the modelling of the gearshift to properly measure the energy consumption as well as investigate vehicle drivability.

The first step in understanding the gearshift control of the 2SED is to derive the equations governing the system dynamics in each of the states during the gearshift process. The three states are the engaged first gear, engaged second gear and the inertia phase as explained below.

Engaged first gear: The transmission is in first gear so the electric motor is kinematically linked to the differential final gear according to the first gear ratio. The sprag clutch is driving the secondary shaft, however the friction clutch can be slipping transferring torque. The system represents a single degree of freedom.

Engaged second gear: Similar to ‘Engaged first gear’ the transmission is kinematically in second gear, however the friction clutch is transferring torque. The sprag clutch is overrunning and as such cannot transfer torque. The system represents a single degree of freedom.

Inertia phase: During the inertia phase the friction clutch is slipping and the sprag clutch is overrunning and so the electric motor and primary shaft can rotate independently from the output shaft. The motor is not kinematically in second gear so the system now represents two degrees of freedom.

The equations governing the dynamics of the transmission in each of the three states are derived in the next section, the fixed gear equations differ slightly to the equations derived previously due to the effect of the slipping clutches. The equations do not consider the torsion of the shafts, the plays within the components or the internal torsional dynamics of the components such as the sprag clutch. The sprag clutch was in fact measured to have a maximum torsion of 2-3 degrees which is negligible compared to the torsion of the half-shafts which is much greater and therefore has a larger impact on the vehicles low frequency drivability. Furthermore, the effect of these transmission characteristics has little impact on the gearshift control or the dynamics of the system in constant gear or during a gearshift.

4.2 GEARSHIFT MODEL

The equations governing the system response in engaged first gear are found through considering the moment balance across each of the transmission shafts. The derivation of the equations governing the transmission dynamics in first gear have already been
Two-speed transmission gearshift dynamics: Gearshift model

explained in equations (60) to (65) however the primary shaft and secondary shaft equations need to be altered to consider the friction clutch slipping torque.

The new torque balance equation about the primary shaft, is given in equation (94). $T_{fc}$ is the friction clutch torque.

$$T_{m,det} - T_{fc} - F_{1} \cdot R_{1} = (J_{m} + J_{1}) \cdot \ddot{\theta}_{1}$$  \hspace{1cm} (94)

The new torque balance equation about the secondary shaft, is given in equation (95).

$$F_{2} \cdot R_{2} \cdot \eta_{2} + T_{fc} - T_{hsl} - T_{hsr} = \left( J_{3} + \frac{1}{2}(J_{hsl} + J_{hsr}) \right) \cdot \ddot{\theta}_{3}$$  \hspace{1cm} (95)

The new resultant differential acceleration in first gear is then given in equation (96).

$$\ddot{\theta}_{\text{diff}} = \frac{T_{m,det} \cdot \tau_{1} \cdot \tau_{\text{diff}} \cdot \eta_{1} \cdot \eta_{3} + T_{fc} \cdot \tau_{2} \cdot \tau_{\text{diff}} \cdot \eta_{2} \cdot \eta_{3} - T_{fc} \cdot \tau_{1} \cdot \tau_{\text{diff}} \cdot \eta_{1} \cdot \eta_{3} - T_{hsl} - T_{hsr}}{(J_{mot} + J_{1}) \cdot \tau_{1}^{2} \cdot \tau_{\text{diff}}^{2} \cdot \eta_{1} \cdot \eta_{3} + (J_{2} + J_{2b}) \cdot \tau_{\text{diff}}^{2} \cdot \eta_{3} + J_{3} + \frac{1}{2}(J_{hsl} + J_{hsr})}.$$  \hspace{1cm} (96)

Whilst the transmission is engaged in first gear the torque is purely being transferred through the sprag clutch and the friction clutch torque is zero. During an upshift the friction clutch is engaged to transfer the torque from the sprag clutch to the friction clutch. Until the sprag clutch is overrun the transmission is kinematically still in first gear so the equation governing the dynamics in first gear includes the friction clutch torque term. The friction clutch torque applied to the system alters the transmission output torque by $T_{fc} \tau_{\text{diff}} \eta_{3}(\tau_{2} \eta_{2} - \tau_{1} \eta_{1})$ as the motor torque is split between the two clutches. The contribution from first gear being, $(T_{m,det} - T_{fc}) \cdot \tau_{1} \cdot \tau_{\text{diff}} \cdot \eta_{1} \cdot \eta_{3}$, and the contribution from second being, $T_{fc} \cdot \tau_{2} \cdot \tau_{\text{diff}} \cdot \eta_{2} \cdot \eta_{3}$.

When the friction clutch has overcome the sprag clutch torque and the sprag clutch is overrunning but the motor is not kinematically in second gear the transmission enters the inertia phase. The motor speed in relation to the wheel speed is neither equal to 1st or 2nd gear due to the slipping clutch. The primary shaft and output shaft governing equations then change as shown below, where equation (97) represents the motor dynamics and equation (98) represented the transmission dynamics. The two equations characterise the two degrees of freedom.

$$\dot{\theta}_{\text{motor}} = \frac{T_{m,det}-T_{fc}}{(J_{2b} / (\tau_{1}^{2} \eta_{1}))J_{m}+J_{1}} $$  \hspace{1cm} (97)

$$\dot{\theta}_{\text{diff}} = \frac{T_{fc} \tau_{2} \tau_{\text{diff}} \eta_{2} \eta_{3} - T_{hsl} - T_{hsr}}{I_{\text{eqv,trans,inertia}}}$$  \hspace{1cm} (98)

$$I_{\text{eqv,trans,inertia}} = J_{3} + \frac{1}{2}(J_{hsl} + J_{hsr}) + J_{1b} \cdot \tau_{2}^{2} \cdot \tau_{\text{diff}}^{2} \cdot \eta_{2} \cdot \eta_{3} + (J_{2}) \cdot \tau_{\text{diff}}^{2} \cdot \eta_{3}$$  \hspace{1cm} (99)
Two-speed transmission gearshift dynamics: Gearshift model

The main difference between the transmission being in a state of no engaged gear is the inertia of the motor is not considered, so the equivalent moment of inertia is much lower. The low moment of inertia results in a higher first natural frequency of the system when in the inertia phase than when in-gear. The higher natural frequency can result in NVH problems and drivability issues if the clutch control is not properly tuned.

The equations governing the system dynamics in second gear are similar to the equations for first gear. The system has one degree of freedom, as in first gear, and therefore the single equation of the system is shown in equation (100)

\[
\dot{\theta}_3 = \frac{T_{m,del} \cdot \tau_2 \cdot \tau_{diff} \cdot \eta_2 \cdot \eta_3 - T_{hsl} - T_{hSR}}{(J_{mot} + J_1 + J_{1b}) \cdot \tau_1^2 \cdot \tau_{diff}^2 \cdot \eta_1 \cdot \eta_3 + 2 J_2 \tau_2^2 \cdot \tau_{diff}^2 \cdot \eta_2 \cdot \eta_3} + J_3 \cdot \eta_3 + \frac{1}{2} \left( J_{hsl} + J_{hSR} \right)
\] (100)

The moment of inertia of the electric motor is the biggest contributing inertia and is much lower when second gear is engaged than first gear, due to the fact it is multiplied by the second gear ratio instead of the first gear ratio.

4.2.1 LINEAR MODEL OF THE SYSTEM

As previously mentioned the system (drivetrain) in the three different states has varying equivalent moment of inertias so should be subject to different first natural frequencies that affect the drivability of the vehicle. To understand the values and to illustrate the differences a linearised model of the system needs to be built to generate Bode plots. The linearised model includes the friction clutch actuation, the tyre relaxation length as well as the tyre slip, the dynamics of the electric motor and the torsional dynamics of the halfshafts.

A linearised model is found through a state-space representation of the model, which is essentially a mathematical model based on first order differential equations. The state-space system, which is continuous linear time invariant (LTI), is represented through the following equations, (101) and (102).

The vector, \( x \), represents the state vector (the drivetrain system being modelled is a LTI so can be written in matrix form). The state vector represents the minimum number of variables needed to model the system, however, formally it consists of variables that have a differential of the same term within the governing equations, for example longitudinal velocity and longitudinal acceleration. The vector, \( y \), is the output vector and is the result of the system. The vector, \( u \), is the input vector and consists of constants in the governing equations which are inputs to the system. The matrix, \( A \), is the state matrix, and consists of the terms that multiply the state vector, \( x \), and as such is a square matrix the same height and width as the length of the state vector, \( x \). \( B \) is the input matrix and consists of the terms that multiply against the input vector, \( u \), for each state equation. The matrix \( C \) is the output matrix, and essentially consists of a row of ones down the diagonal so the output simply equals the input. In the same way, matrix \( D \) is populated by zeros for a standard system.
where there is no direct feed through. However, additional outputs can be added in the output matrices by including equations that consist of terms solved through the state and input matrices.

\[ \dot{x}(t) = Ax(t) + Bu(t) \]  
\[ \dot{y}(t) = Cx(t) + Du(t) \]  

(101)  
(102)

As stated before, the linear system developed accounts for the dynamics of the electric motor/transmission along with various physical phenomena. The equations of the system were derived and are given below, where the equations are rearranged to give the state variable on the left hand side.

The first state variable, \( \dot{T}_{m_{\text{delay}}} \), is the delayed electric motor torque after the motor time constant, \( \tau_m \), is applied, found from the theoretical electric motor torque, \( T_{m_{\text{theor}}} \), as in equation (103).

\[ \dot{T}_{m_{\text{delay}}} = \frac{T_{m_{\text{theor}}}}{\tau_m} - \frac{T_{m_{\text{delay}}}}{\tau_m} \]  

(103)

The second state variable is the differential angular acceleration, equation (104). The state variable the differential angular velocity is simply found through having a one in the A matrix so is equal to the integral, the angular displacement of the differential. In equation (104), \( g \) is the selected gear ratio, \( \eta_g \) is the selected gear efficiency and \( I_{\text{equiv}} \) is the equivalent moment of inertia.

\[ \ddot{\theta}_{\text{diff}} = \frac{(\tau_{\text{diff}} \tau_g \eta_g) T_{m_{\text{delay}}}}{I_{\text{equiv}}} + \frac{(K_{hsf}) \dot{\theta}_w}{I_{\text{equiv}}} + \frac{(B_{hsf}) \dot{\theta}_w}{I_{\text{equiv}}} + \frac{(-B_{hsf}) \dot{\theta}_{\text{diff}}}{I_{\text{equiv}}} \]  

\[ + \frac{(-K_{hsf}) \theta_{\text{diff}}}{I_{\text{equiv}}} \]  

(104)

The wheel rotational acceleration is the next state variable, which considers the half-shaft torque, tyre relaxation length and rolling resistance. The rolling resistance for a non-linear system, \( F_{\text{rolling resistance,non linear}} \), is given in equation (105) and is linearised using Taylor series expansion to give, \( F_{\text{rolling resistance,linear}} \), and is shown in equation, (106).

\[ F_{\text{rolling resistance,non linear}} = F_{t_z} \left( f_0 + f_2 \dot{\theta}_w^2 r_w^2 \right) \]  

(105)

\[ F_{\text{rolling resistance,linear}} = f_0 F_{t_z} + f_2 F_{t_z} \dot{\theta}_w^2 r_w^2 + 2 f_2 F_{t_z} \dot{\theta}_w^2 r_w^2 \Phi \]  

(106)

The final wheel rotational acceleration equation is given in equation (107). Again, the wheel rotational velocity is a state variable but simply equal to its integral, the wheel rotational displacement. \( m_f \) is the vehicle mass on the front axle and \( \dot{\theta}_{w_0} \) is the initial wheel speed.
Two-speed transmission gearshift dynamics: Gearshift model

\[
\ddot{\theta}_w = \frac{(B_{hsf})\dot{\theta}_{\text{diff}}}{2J_w} + \frac{(K_{hsf})\dot{\theta}_{\text{diff}}}{2J_w} - \frac{(2.Rw^3.f_2.m_\tau.g.\dot{\theta}_{w\tau})}{2J_w} - \frac{(R_w^2.f_3.m_\tau.g) - B_{hsf}}{2J_w} \dot{\theta}_w + \frac{(-K_{hsf})\theta_{w\tau}}{2J_w} + \frac{(-R_w)F_{x\text{del}}}{2J_w} \tag{107}
\]

The state variable, delayed tyre longitudinal force, \(F_{x\text{del}}\), which is a function of the tyre damping coefficient, \(B_{\text{tequiv}}\), wheel/vehicle velocities and the tyre time constant, \(\tau_{\text{tyre const}}\), as shown in equation (108).

\[
\dot{F}_{x\text{del}} = \frac{B_{\text{tequiv}}}{\tau_{\text{tyre const}}} \dot{\theta}_v + \frac{B_{\text{tequiv}}}{\tau_{\text{tyre const}}} \dot{\theta}_w - \frac{F_{x\text{del}}}{\tau_{\text{tyre const}}} \tag{108}
\]

The final state variable, vehicle acceleration considers the tyre longitudinal force and the aerodynamic force. Similarly to the rolling resistance, the aerodynamic force needs to be linearised using Taylor Series Expansion to be considered in a linear system.

\[
F_{\text{aerodynamic force, non linear}} = \frac{1}{2}\rho S_v C_d \dot{\theta}_v^2 \tau_w^2 \tag{109}
\]

\[
F_{\text{aerodynamic force, linear}} = \frac{\tau_w^2\rho S_v C_d \dot{\theta}_v \dot{\theta}_v \dot{\theta}_v \dot{\theta}_0^2}{J_v} + \frac{1}{J_v^2}\frac{\tau_w^2\rho S_v C_d \dot{\theta}_v \dot{\theta}_0^2}{J_v} \tag{110}
\]

The final equation which calculates the vehicle acceleration is then given in equation (111). \(\dot{\theta}_v\) is the initial vehicle speed.

\[
\dot{\theta}_v = \frac{\tau_w}{J_v} \dot{F}_{x\text{del}} - \frac{\tau_w^2\rho S_v C_d \dot{\theta}_v \dot{\theta}_v \dot{\theta}_v \dot{\theta}_0^2}{J_v} + \frac{1}{J_v^2}\frac{\tau_w^2\rho S_v C_d \dot{\theta}_v \dot{\theta}_0^2}{J_v} \tag{111}
\]

The outputs of the system which represent the response of the vehicle are the half-shaft torque and the vehicle acceleration. As neither of these variables are represented by state variables they are calculated in the output matrices, C and D using the equations shown below.

\[
T_{hs} = -\frac{(K_{hsf})\theta_w}{J_{\text{equiv}}} - \frac{(B_{hsf})\theta_w}{J_{\text{equiv}}} + \frac{(B_{hsf})\dot{\theta}_{\text{diff}}}{J_{\text{equiv}}} + \frac{(K_{hsf})\dot{\theta}_{\text{diff}}}{J_{\text{equiv}}} \tag{112}
\]

\[
\ddot{\theta}_v = \frac{R_w.\dot{F}_{x\text{del}}}{J_v} - \frac{R_w^4.\rho.\text{Sc}.C_d.\dot{\theta}_v \dot{\theta}_v \dot{\theta}_v \dot{\theta}_0^2}{J_v} + \frac{0.5.R_w^4.\rho.\text{Sc}.C_d.\dot{\theta}_v \dot{\theta}_0^2}{J_v} \tag{113}
\]

The arrays, A, B, C and D are populated by the equations above.

The final bode plot of the half-shaft torque is shown in Figure 4-1, where the dissimilarity in the first natural frequency of the system between first and second gear is due to the lower equivalent moment of inertia of the electric motor at the wheels. The low amplitude of the
Two-speed transmission gearshift dynamics: The gearshift control system

frequency response in fixed gear is due to the system being overdamped which is a consequence of the high time constant of the electric motor. However, if the electric motor had a lower time constant the system would be under damped and there would be peaks at the natural frequency of each of the two fixed gears. During the inertia phase, the first natural frequency is reduced as the equivalent moment of inertia is further reduced when the electric motor is disengaged from the rest of the drivetrain. The amplitude of the frequency response is reduced between 1 Hz and 25 Hz during the inertia phase, owing to the dynamic characteristics (time constant) of the friction clutch actuator. A further study could look into the natural frequency of other components in the drivetrain (mainly the drivetrain mounting system) or vehicle to understand if the natural frequency is similar to the transmission in each state so if any resonance could occur.

![Bode Diagram for Each State](image)

Figure 4-1: Frequency response of the system in first gear, second gear and during the inertia phase for system linearisation, at vehicle speed 10 m/s

4.3 THE GEARSHIFT CONTROL SYSTEM

The 2SED is similar to a DCT in that it has two clutches and as such the gearshift methodology shares some similarities, however only one clutch can be controlled. Furthermore, due to the input being an electric motor not an ICE there are some differences in the control techniques adopted to compensate for the physical changes in the power plant, i.e. time constant and importantly, no idle (i.e. minimum engine speed). The control system adopted utilized a feedforward controller and a PID, the reason being a simple control system is easy to implement in an ECU where processor speed is limited. A feedforward controller was included as well as the PID to ensure the system is not affected by noise and more stable and as such less affected by oscillations.

4.3.1 UPGSHIFT CONTROL

The gearshift methodology for an upshift from first to second gear during power on initially requires the disengagement of the locking ring. The locking ring locks the 1st gear sprag clutch to the secondary shaft so that when the vehicle reverses in first gear the clutch does not over run. The upshift methodology of the 2SED transmission is similar to a DCT in that is
starts with a torque phase which is started by the friction clutch being engaged at a rate dictated by the controller and limited by the physical constraints of the actuator. The transmission is still kinematically in first gear however, the sprag clutch torque is reducing according to the torque being transmitted through the friction clutch. The output of the transmission is therefore progressively changing from the first gear value to the second gear value. The progressive change results in a gearshift without a torque gap which is the main drawback of a manual transmission, where the disengagement of the clutch to switch between gears results in a period of no torque transfer. The torque gap has adverse effects on the vehicle drivability and provokes unwanted drivetrain dynamics, e.g. due to the twisting of the half-shafts.

The torque phase is complete when the torque being transferred by the sprag clutch is zero and this is when the friction clutch torque is equal to that found in equation (114).

\[ T_{fc} = T_m - (J_m + J_1)\dot{\theta}_m - \frac{J_2 b \dot{\theta}_{diff} \tau_{diff}}{\tau_1 \eta_1} \quad (114) \]

\( T_{fc} \) is estimated by the control system from the displacement of the friction clutch actuator. Equation (114) establishes the beginning of the inertia phase of the upshift, during which the electric motor speed has to be reduced from the value for the first gear ratio to the value for the second gear ratio, whilst keeping an adequate vehicle acceleration profile. The principles for the inertia phase control can be derived from equations (97) and (98). During the inertia phase, vehicle acceleration dynamics are controlled by the friction clutch torque (equation (98)), whilst the difference between electric motor torque and friction clutch torque affects the motor dynamics (equation (97)). As a consequence, the two degrees of freedom of the system can be independently controlled, provided that the electric motor drive is not working in conditions of saturation (on its peak torque characteristic, which represents the constraint of the control system). This is the ‘golden rule’ for the control of the inertia phase of the gearshifts in such a system.

During the inertia phase of the upshift for ‘Control 1’, \( T_{fc} \) is ramped up according to an open-loop control system at the same rate as the torque phase to reduce any driveline oscillations during phase transitions. \( T_{fc} \) ramps to a reference level equal to the torque value the electric motor would produce for the actual condition of driver torque demand \( DTD(t) \) (not manipulated by the controller) and electric motor speed. Additional terms compensate for the inertial torque of the main components of the system. \( T_{fc, \text{saturation, IP, US}} \) is given by:

\[ T_{fc, \text{saturation, IP, US}}(t) = T_m \left( DTD(t), \dot{\theta}_m(t) \right) - (J_m + J_1)\dot{\theta}_m \quad (115) \]

In the meantime, electric motor dynamics are controlled by the combination of a feedforward and a feedback (Proportional Integral Derivative - PID) controller, based on a reference speed profile, \( \dot{\theta}_{m, \text{ref}} \), equal to:

\[ \dot{\theta}_{m, \text{ref}} = \dot{\theta}_{diff} \cdot \tau_{diff} \cdot \tau_1 \cdot y(t_{IP}) \quad (116) \]
where the adimensional factor \( y(t_{ip}) \) is a normalisation parameter defining the reference speed profile, according to the qualitative shape in Figure 4-2. \( t_{ip} \) is the output of a counter which is activated by the transmission control unit at the beginning of the inertia phase. The initial value of \( y \) is 1, so that equation (116) provides an initial value of the reference motor speed equal to the actual speed of the unit at the beginning of the inertia phase. The final value of \( y \) is equal to the step ratio \( \tau_2 / \tau_1 \), so that the final value of the reference motor speed is equal to the one required for the synchronization of the friction clutch.

The shape of \( y(t_{ip}) \) is designed so that the electric motor reference speed is at a maximum in the first part of the inertia phase, and reduces to zero (for the specific tuning shown in Figure 4-2) at the time \( t_{ip,\text{end}} \) at the end of the inertia phase. The shape of the profile can be tuned to alter the duration of the inertia phase, depending on the vehicle parameters and required upshift performance, and the amount of perceived discontinuity at the engagement point of the friction clutch at \( t_{ip,\text{end}} \). If the air-gap torque dynamics of the motor drive is modelled through a first order transfer function, the open-loop transfer function of the feedback part of the controller of Figure 4-2 is given in equation (117). \( s \) is a laplace variable, \( \dot{\theta}_m \) is the reference motor speed, \( \dot{\vartheta}_m \) is the motor speed where the horizontal line represents a variable in the frequency domain, \( K_P \) is the PID gain, \( K_D \) is the derivative gain, \( K_I \) is the integral gain and \( \tau_m \) is the motor time constant.

\[
\frac{\dot{\theta}_m(s)}{\dot{\theta}_{m,\text{ref}}(s) - \dot{\vartheta}_m(s)} = \left( K_P + K_D s + \frac{K_I}{s} \right) \frac{1}{J_m s (1 + s \tau_m)}
\]  

Hence the gains of the feedback part of the controller can be tuned according to the well known rules in terms of tracking capability (bandwidth) and phase margin, Ogata (2009).

Figure 4-3 shows an example of possible tuning of the PID control parameters and the relating open-loop (given by equation (117) and closed-loop transfer functions. The transfer functions are affected by the time constant of the electric motor, defined by air-gap torque dynamics (sometimes filtered for anti-jerk purposes). Consequently a sensitivity analysis
Two-speed transmission gearshift dynamics: The gearshift control system

has been added to Figure 4-3 to illustrate the effect of the motor time constant. The PID utilised only a P value with the same value of P used for each controller.

The parameters of the electric motor controller can vary depending on DTD and speed, in order to make the upshift quicker or more comfortable as a function of the specific driving situation. In any case, the motor controller has very low impact on the low frequency vehicle drivability during the upshift, as this is controlled through the friction clutch (equation (115), consistently with the ‘golden rule’. Therefore, the performance of the system is very robust against the variation of the parameters of the feedback motor controller. Tests have been successfully carried out with only the feed-forward system reducing the electric motor drive torque by a constant amount and so with no contribution from the PID, with little variation of the perceivable quality of the achieved results.

![Figure 4-3: Bode diagram of the open-loop and closed-loop transfer functions for the feedback part of the electric motor control loop](image)

At the conclusion of the inertia phase, when the friction clutch engages, the friction clutch actuator is moved to its end stop where the transmissible clutch torque is at the nominal level, depending on the wear condition of the clutch discs.

![Figure 4-4: Possible conditions of an upshift manoeuvre on the electric motor drive torque characteristic, under the hypothesis of constant driver torque demand DTD during the manoeuvre](image)
The upshift control system described until now (‘Control 1’) gives origin to a complete absence of torque gap during the upshift when the initial and the final operating points of the electric motor drive are in the constant torque region of the electric motor. For example, this condition is satisfied for the upshifts from point A to B or point C to D in Figure 4-4. For upshifts in the constant power region of the electric motor drive, such as those from E to F or G to H, the torque phase of the upshift, when operated as described for ‘Control 1’, involves a reduction of wheel torque. For the same conditions, the inertia phase implies a progressive increase of wheel torque, consistent with the reduction of electric motor speed and the subsequent increase of $T_{fc,saturation,IP,US}$ (equation (115)). These variations of wheel torque, especially the increase during the inertia phase, are progressive and provoke lower jerk levels and significantly better acceleration profiles than those experienced in a conventional single-clutch transmission. This is due to the single-clutch transmission suffering from negative values of vehicle acceleration during the phase of the upshift characterised by the disengaged clutch. The control system defined as ‘Control 1’ is reliable, robust and relatively simple; therefore was adopted on the 2SED installed on the HIL test rig.

In order to significantly reduce the torque gap during the inertia phase of the upshift, it is possible to adopt the following control characteristic, here named ‘Control 2’, for the friction clutch saturation level, $T_{fc,saturation,IP,US,C2}$:

$$T_{fc,saturation,IP,US,C2}(t) = T_m(DTD(t), \dot{\theta}_m(t) \cdot \tau_{diff} \cdot \dot{\tau}_2) - (J_m + J_1)\ddot{\theta}_m$$ (118)

The control system provokes the friction clutch to transmit a torque level equal to the value the electric motor would generate if the system was already in second gear. The adoption of the friction clutch torque of equation (118) during the inertia phase of the upshift significantly improves vehicle acceleration, as it eliminates the partial wheel torque gap during the inertia phase of the upshift. However, it generates significant vehicle jerk in the transition between the torque phase and the inertia phase of the upshift. This is due to the fact that the torque phase of the upshift, when implemented according to ‘Control 1’ and ‘Control 2’, intrinsically produces a reduction of the available wheel torque which is progressively recovered in case of ‘Control 1’ but quite abruptly recovered in case of ‘Control 2’. ‘Control 2’ could be adopted as a sport-oriented transmission control algorithm selectable by the driver.

In the case of an upshift in the constant power region of the electric motor, it is possible to compensate for the reduction of wheel torque, induced by the torque shift from the first to the second gear (torque phase of the upshift), by manipulating the electric motor torque demand. This variant of the control system is defined as ‘Control 3’ and is described by equation (119), where $T_{fc,saturation,IP,US,C3}$ is the friction clutch torque defined by ‘Control 3’ and $T_{fc,est}$ is the estimated friction clutch torque:

$$T_{fc,saturation,IP,US,C3} = T_m(DTD, \dot{\theta}_m) + T_{fc,est} \left(1 - \frac{\tau_2}{\tau_1}\right)$$ (119)
Two-speed transmission gearshift dynamics: The gearshift control system

With the implementation of such a control system during the entire torque phase, the disengagement of the sprag clutch would happen at a friction clutch torque level, $T_{fc,\text{dis}}$, equal to:

$$T_{fc,\text{dis}} = \left[ T_m(DTD, \dot{\theta}_m) - (J_m + J_1)\ddot{\theta}_m = \frac{J_2b \dot{\theta}_m \tau_{\text{diff}}}{\tau_1 \eta_1} \right] \frac{\tau_2}{\tau_1}$$

(120)

As a consequence, the final level of the friction clutch torque during the torque phase of the upshift could be higher than the desired friction clutch torque of equation (118 for the inertia phase of the upshift, which is the same for ‘Control 2’ and ‘Control 3’, in case of an upshift carried out in the constant torque region of the electric motor drive. In order to prevent a significant negative jerk (due to the reduction of the friction clutch torque) at the transition between the torque phase and the inertia phase of the upshift, when $T_{fc,\text{est}} > T_{fc,\text{saturation,IP,US,C2}}$, $T_{fc,\text{saturation,IP,US,C3}}$ is switched back to the level imposed by the driver torque demand $DTD(t)$. During this transition careful tuning of the control parameters have to be carried out, paying particular attention to the dynamics of the friction clutch actuator and the electric motor. The system sensitivity to the clutch actuator dynamics has already been demonstrated in Sorniotti et al. (2011).

4.3.2 DOWNSHIFT CONTROL

As a first approximation, the downshift actuation sequence is a reverse of the upshift method, wherein the inertia phase precedes the torque phase. Downshifts in power-on are actuated following significant increases of driver torque demand in order to increase the amount of wheel torque. Due to the significantly lower frequency of downshifts in power-on in comparison with the upshifts (consequence of the usual gear selection algorithms), the control system adopted in the inertia phase is a simplified version of the one presented for the upshifts, even if potentially the control systems could be similar.

Downshifts in power-on are accomplished by initially opening the friction clutch at the rate allowed by the actuator dynamics. When the clutch transmissible torque is lower than the electric motor torque, the clutch starts slipping, giving origin to the inertia phase. The motor torque is kept at the level requested by the user, whilst the friction clutch torque is controlled in order to produce the required acceleration level of the electric motor shaft. At the start of the inertia phase, the friction clutch actuator position should be carefully monitored due to the dynamic friction coefficient of the dry friction clutch being lower than the static friction coefficient. The manipulation of both motor torque and friction clutch torque (according to the ‘golden rule’ presented in section 4.3.1) during the inertia phase of the downshift in power-on would lead to the full controllability of both electric motor dynamics and vehicle acceleration, at the cost of a significantly increased complexity of the control system and the time required to tune its parameters for each vehicle application. The friction clutch can be momentarily re-engaged when the sprag clutch is about to connect (beginning of the torque phase of the downshift), in order to dampen the re-engagement of the sprag clutch and reduce any jerk.
4.4 GEARSHIFT SIMULATION RESULTS

The non-linear vehicle simulation model explained in chapter 3 was utilised to evaluate the gearshift dynamics of the novel transmission system with the transmission block modified to account for the additional degree of freedom. The vehicle data is included in Appendix C.

4.4.1 UPGRADE RESULTS

Figure 4-5 and Figure 4-6 summarise the overall transmission system dynamics for an upshift at 80% of driver torque demand, carried out at a vehicle speed of 75 km/h, in the constant power region of the electric motor drive and using control system ‘Control 3’.

![Speed Dynamics during Upshift - Torque Demand 80%](image1)

![Torque Dynamics during Upshift - Torque Demand 80%](image2)

Figure 4-5: Electric motor drive dynamics and gear input torques during an upshift at 80% of driver torque demand

The phases of the upshift, namely ‘Upshift Request’ followed by the torque phase, inertia phase (defined by ‘Inertia Phase Start’ and ‘Inertia Phase End’) and the final motion of the actuator after the engagement of the second gear (defined by ‘Actuator Stop’), are evident in the graphs. The effect of the efficiencies and the moments of inertia of the components are also visible in the graphs, for example in the marginal difference between the electric motor torque and the input torques transmitted by gear one and two when a gear is engaged in Figure 4-5.
Two-speed transmission gearshift dynamics: Gearshift Simulation Results

During the torque phase the electric motor torque demand is modified according to equation (119) and is increased in order to compensate for the reduction of vehicle acceleration induced by the torque transfer from first to second gear. Due to the sharp gradient in the reference electric motor speed at the start of the inertia phase, the feedforward and feedback controller shown in Figure 4-2 provokes a large decrease in electric motor torque (Figure 4-5). The friction clutch torque is ramped up to make the vehicle acceleration equal to the level of the vehicle acceleration in second gear, according to equation (118). In the second part of the inertia phase the rate of motor reference speed is reduced as shown in the graph of Figure 4-2, and consequently the electric motor torque demand is increased. Moreover as the electric motor speed reduces, the maximum available torque increases due to the torque map of the electric motor (Figure 4-4). This justifies the difference between the first gear torque and the second gear torque in conditions of engaged gear in Figure 4-5, and represents the main peculiarity to be taken into account in the implementation of algorithms for gearshift control of electric powertrains.

The torque actually transmitted by the friction clutch and the maximum torque which can be potentially transmitted (transmissible torque) for an assigned clutch actuator displacement are the same when the clutch is slipping whilst they differ when the clutch is enganged, as in Figure 4-6. This is evident after the inertia phase, when the transmissible torque of the friction clutch is increased due to the change between the dynamic and static friction coefficient of the clutch. Finally the actuator is moved to increase the friction clutch axial force and therefore the transmissible torque. Notably, Figure 4-6 shows that the friction clutch torque transmitted in second gear is lower than that transmitted by the sprag clutch in first gear due to the friction clutch being located on the primary shaft whilst the sprag clutch is located on the secondary shaft.
In addition, Figure 4-6 illustrates that the pressure plate moves when the shift is initiated to take up any play between the clutch plates. After recovering the play, during the torque and inertia phases of the upshift there is an infinitesimal axial movement of the pressure plate due to the high axial stiffness of the clutch plates, although the transmissible torque varies with the friction clutch actuator travel, which depends on the stiffness properties of the Belleville spring.

Figure 4-7: Half-shaft torque dynamics during the same upshift as the previous two figures (constant power region of the electric motor), and during an upshift at 40% of driver torque demand and 30 km/h (constant torque region of the electric motor)

Figure 4-7 plots the time history of half-shaft torques (sum of the torques of the left and right half-shafts) for the same manoeuvre as in Figure 4-5 and Figure 4-6, and during an upshift at 40% of driver torque demand, carried out at 30 km/h, in the constant torque region of the electric motor drive. Both manoeuvres have been simulated by adopting ‘Control 3’. The marginal torque gap induced by the torque phase of the upshift and recovered by the friction clutch control during the inertia phase is evident in the first manoeuvre. The second manoeuvre is characterised by the total absence of any torque gap. This characteristic is common to ‘Control 1’, ‘Control 2’ and ‘Control 3’, when the upshift is requested in the constant torque region of the electric motor unit. The equivalent moment of inertia of the transmission is very high in first gear and very low during the inertia phase and it is this change which provokes some marginal oscillations in the half-shaft torque.

Figure 4-8 compares the vehicle acceleration profiles achievable during two different upshifts at 40% (different manoeuvre from Figure 4-7) and 80% (identical manoeuvre to Figure 4-5 to Figure 4-7) of driver torque demand, both in the constant power region of the electric motor drive. Table 4-1 provides an objective comparison of the three gearshift strategies during each manoeuvre.
Figure 4-8: Upshifts in the constant power region of the electric motor drive, at 40% (different from the upshift in the previous three figures) and 80% (the same as the previous three figures) of driver torque demand: vehicle acceleration profiles.

‘Control 2’ shows most of the benefit from the viewpoint of the acceleration time (a gain of more than 0.2 s when compared to ‘Control 1’), however the mean of the absolute value of jerk (time derivative of vehicle acceleration) during the simulation of the upshift is higher than for ‘Control 1’, with an even more significant disadvantage in terms of peak value of jerk. ‘Control 3’ represents the best compromise between a high performance upshift and the requirement for the expected comfort level. At 80% of torque demand, the benefit of ‘Control 3’ is much more limited than at 40% of torque demand, due to the fact that the torque increase specified by ‘Control 3’ is saturated at the peak torque of the electric motor (at 100% of motor torque demand, ‘Control 3’ produces the same performance as ‘Control 2’).

Table 4-1: Comparison of ‘Control 1’, ‘Control 2’ and ‘Control 3’ during the same upshift manoeuvres of the previous figure (in the constant power region)

<table>
<thead>
<tr>
<th>Torque Demand</th>
<th>40 – 100 km/h</th>
<th>70 – 100 km/h</th>
<th>Upshift time</th>
<th>Mean acceleration during upshift</th>
<th>Mean jerk during upshift</th>
</tr>
</thead>
<tbody>
<tr>
<td>40%</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Control 1</td>
<td>14.91 s</td>
<td>9.55 s</td>
<td>1.11 s</td>
<td>0.76 m/s²</td>
<td>0.95 m/s²</td>
</tr>
<tr>
<td>Control 2</td>
<td>14.71 s</td>
<td>9.34 s</td>
<td>1.35 s</td>
<td>1.00 m/s²</td>
<td>1.18 m/s²</td>
</tr>
<tr>
<td>Control 3</td>
<td>14.68 s</td>
<td>9.32 s</td>
<td>1.39 s</td>
<td>1.01 m/s²</td>
<td>0.41 m/s²</td>
</tr>
<tr>
<td>80%</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Control 1</td>
<td>6.53 s</td>
<td>4.03 s</td>
<td>0.95 s</td>
<td>1.82 m/s²</td>
<td>2.24 m/s²</td>
</tr>
<tr>
<td>Control 2</td>
<td>6.33 s</td>
<td>3.84 s</td>
<td>0.70 s</td>
<td>2.25 m/s²</td>
<td>3.22 m/s²</td>
</tr>
<tr>
<td>Control 3</td>
<td>6.31 s</td>
<td>3.82 s</td>
<td>0.74 s</td>
<td>2.31 m/s²</td>
<td>2.15 m/s²</td>
</tr>
</tbody>
</table>

Jerk is considered as there are two quantifying methods of jerk utilised in the field of automotive engineering to measure drivability, the peak of jerk and the r.m.s. (root mean square) jerk. It is generally felt within the industry that a peak of jerk under 10 m/s² (Hrovat
et al., 1981), is acceptable for the driver. Although the research by Zhou Shouren (1984), suggests that at low frequencies of less than 3Hz, the acceptable amount of jerk is $25.5 m/s^3$.

4.4.2 DOWNSHIFT RESULTS

Figure 4-9 summarises the torque and speed dynamics for a power-on downshift (kick-down) during a tip-in test (a sudden driver torque demand request) from 25 km/h where the final driver torque demand is 80%. The downshift takes place in the constant torque region of the electric motor, and so is performed to provoke an increase in available wheel torque. Figure 4-9 illustrates the oscillations in the speeds due to the effect of the tip-in manœuvre on the torsional dynamics of the half-shafts. The transmitted second gear torque reduces at the start of the inertia phase due to the change in the friction coefficient from the static to the dynamic value. This is also evident in Figure 4-10 which shows that the change is partially compensated by the motion of the friction clutch actuator. At the end of the inertia phase, when the motor speed is at the required level and the sprag clutch is engaged, the friction clutch torque is progressively reduced to zero during the torque phase.

![Motor Speed Dynamics](image-url)

![Torque Dynamics](image-url)

Figure 4-9: Speed and torque dynamics during a downshift in power-on for a tip in test at an initial speed of 25 km/h and a final 80% driver torque demand
Two-speed transmission gearshift dynamics: Concluding remarks

4.5 CONCLUDING REMARKS

The research discussed in this chapter focused on the development of a method for modelling and controlling the gearshift of the 2SED two-speed transmission. This research is particularly novel as the 2SED is a prototype transmission which had not had the gearshift dynamics analysed in simulation.

The gearshift model is based around the transmission being in one of three states: ‘engaged first gear’, ‘engaged second gear’ or ‘inertia phase’ and the equations governing each of these states have been clearly derived. A linear model was developed to find the first natural frequency of the system in each of the three states where the lowest frequency was found to be during the inertia phase as expected, due to the lower equivalent inertia of the system in this state.

A novel controller was developed to manage the motor speed during the inertia phase of the gearshift. The open loop and closed loop transfer functions of the controller were analysed and the closed loop was found to be stable up to 9 Hz which is acceptable for a vehicle and robust for different motor time constants.

Three controllers were designed and tested where ‘Control 1’ was a baseline gearshift controller, ‘Control 2’ reduced the torque gap during the inertia phase and ‘Control 3’ proposed a control strategy for shifts taking place in the constant power region by controlling the driver torque demand. The three alternative gearshift control systems have been outlined, with particular reference to the typical characteristics of electric powertrains, which require novel control algorithms for a seamless management of the upshifts within the constant-power region of the electric motor drive. The effect of the different controllers on the vehicle acceleration times and drivability have been given.
5 HARDWARE-IN-THE-LOOP TEST RIG DEVELOPMENT

5.1 INTRODUCTION

A major novel contribution of the research carried out in this project was the development of a HiL test rig for testing electric vehicle drivetrains.

A HiL system is traditionally a technique used in control to test a physical controller by including the plant under control through a simulation model, as in Figure 5-1. HiL systems aid in the development of controllers and systems by not requiring the whole physical system during testing and development. The automotive industry uses this technique to test ECU (Engine Control Unit) functionality through having a simulation model of the engine and vehicle to represent the response of the vehicle. Through adopting a simulation model of the vehicle instead of using the actual vehicle testing can be carried out quickly and simply, reducing development time and cost.

![Figure 5-1: Traditional HiL schematic](image1)

Essentially, a HiL system is a means of combining simulation with a physical system, in that the plant is represented by a mathematical model in simulation. However, modelling the plant can be difficult for complex non-linear systems such as vehicle drivetrains, due to the torsional dynamics for example. Therefore it is possible to include physical components of the plant into the system where the plant simulation model interacts with the physical components, as shown in Figure 5-2, and is the method adopted herein.

The HiL test rig developed by this project includes the physical controllers of the electric drivetrain, i.e. the electric motor controller, the TCU (Transmission Control Unit) and a simulation model of the vehicle but its novelty is the inclusion of the entire electric vehicle drivetrain in physicality. The drivetrain components of the electric vehicle included on the HiL test rig are the electric motor/inverter, the transmission and the half-shafts.

![Figure 5-2: HiL Test rig schematic](image2)
Hardware-in-the-loop test rig development: Test rig development

The HiL test rig is capable of simulating various tests to analyse a vehicle's response. The main function of the HiL test rig is to simulate driving cycles to understand the impact of different vehicle parameters on energy consumption. The test rig is also designed to carry out performance tests such as full throttle acceleration tests, inclination tests, etc. to analyse a vehicle's performance. The test rig can also carry out drivability tests, such as tip-in/tip-out tests to analyse a vehicle's jerk, etc. The results can be used to validate the simulation models of the vehicle drivetrain, increasing the accuracy of the simulation and validity of the results.

5.2 TEST RIG DEVELOPMENT

The test rig is based around the two main transmissions adopted for the research carried out in this project, the 2SED transmission and its single-speed variant, both designed by Vocis Drivelines and Oerlikon Graziano. The transmissions were developed to be multi-platform, however the case study vehicle used for this research which is being developed as a prototype vehicle by Vocis Drivelines and Oerlikon Graziano is an Mercedes eVito taxi. The powertrain for the eVito taxi consists of the transmission and a 70 kW PMSM motor developed by Zytek Ltd. The university has been provided with both the single-speed transmission and the 2SED transmission along with the 70 kW motor and inverter to be used on the HiL test rig. Furthermore, the half-shafts used on the eVito taxi were included and positioned on the test rig as in the vehicle, i.e. at the correct inclination, as shown in Figure 5-3.

![HiL test rig as installed at the University of Surrey](image)

Figure 5-3: HiL test rig as installed at the University of Surrey

The major components of the test rig are the electric motors installed on each driveshift hub, the ‘hub motors’, which create the resistive torque which would normally be represented by the road load and resistive forces on the vehicle, i.e. aerodynamic drag, rolling resistance, etc. The ‘hub motors’ needed to be sized to accommodate the maximum resistive torque required, which when considering the 70 kW ‘input motor’ which has a maximum torque of 300 Nm with an approximate maximum gear ratio of 15 for first gear, gives a theoretical wheel torque of 4500 Nm not accounting for efficiency. Therefore each
hub motor should have a theoretical maximum torque of 2000 Nm (accounting for a 90% total drivetrain efficiency). The actual torque transmitted through the wheels is limited by the vehicle mass and tyre adhesion which is low for a passenger vehicle, so although 4500 Nm may be available, that which is being transmitted by the tyres to the road is much lower and it is this value which the ‘hub motors’ need to provide. The maximum speed requirement of the ‘hub motors’ is low compared to the ‘input motor’ maximum speed, as with a wheel radius of 0.3 m a wheel speed of 2000 rpm represents a vehicle speed of 226.19 km/h which is well within any standard driving cycle maximum speed limit or sensible family car maximum speed. Two 90 kW motors were selected, where the torque and power profiles are shown in Figure 5-4 and fall well within the maximum torque and speed ranges required.

The test rig requires two separate power sources, one for the ‘hub motors’ and a second to supply the 70 kW motor. The ‘hub motors’ can work as either motors or generators, with symmetric torque characteristics to operate when the vehicle is either in traction or regeneration. The power supply for the hub motors is taken from inverters connected to the national grid and has the additional functionality of generating electricity when in regeneration mode and putting electricity back to the grid. This is a significant cost saving feature as for the majority of the tests when the ‘input motor’ is running in traction and providing a positive torque the ‘hub motors’ are working as generators. If the ‘input motor’ is running through the same power source the DC bus linkages can in fact be connected together so the power being generated can be used to run the input motor and only the power lost due to the efficiencies of the motors, inverters and mechanical system (i.e. transmission, cv joints) is taken from the power source. However, the power supply for the 70 kW motor which acts as the driving motor requires a more complex power supply due to the fact the 70 kW inverter needs a DC power supply as it represents a DC car battery. The battery simulator functioned acceptably, however, a car battery will supply a DC voltage with very little voltage ripple and the battery simulator output had a voltage ripple of a noticeable amount. Therefore a cabinet was built that housed six 0.47 Ohm resistors in parallel that the power supply would pass through which reduced the voltage ripple to a negligible amount. During high current demand periods the resistors would heat up as
expected, so fans were installed along with vents to keep them cool. The power supplies can be seen in Figure 5-6.

The 70 kW ‘input motor’ and inverter are liquid-cooled so a water pump, radiator and fan were installed and connected to the motor and inverter via silicone hoses, as shown in Figure 5-5. This cooling system was found to be adequate to keep both the electric motor and inverter within a reasonable temperature range.

![Motor and inverter cooling system](image)

**Figure 5-5: Motor and inverter cooling system**

![Full HiL test rig schematic including power supplies and controllers](image)

**Figure 5-6: Full HiL test rig schematic including power supplies and controllers**

The HiL test rig itself was designed by Horiba, a UK based engineering company with substantial experience in the drivetrain/engine test rig field. Horiba designed the test rig
base to hold the two ‘hub motors’ along with the base plate to hold the transmission and electric motor in place, as in the vehicle. The manufacturing of the base plate and electric motor mounts were outsourced to a local engineering company, Lunn Engineering.

The test rig has two main controllers, the SPARC Controller developed by Horiba to control the hub motors and a dSPACE MicroAutoBox to run the Simulink/Matlab models which simulate the vehicle model and connects to the SPARC and TCU/Inverter, as shown below.

The dSPACE MicroAutoBox used to control the test rig runs through converting Simulink/Matlab models into c-code. The vehicle models utilised in the dSPACE MicroAutoBox were identical to the models described earlier, however the blocks which represented the drivetrain were removed so that the drivetrain on the test rig could interface with the simulation.

The simulation then sends a torque request to the driving electric motor (70 kW motor) and a speed demand to the hub motors. The SPARC controller which controls the hub motors is in “Speed Control” and as such has its own internal controller to determine how much torque is required at each instance to maintain the required speed. The SPARC Controller connected to speed sensors on the ‘hub motors’ to give some feedback to the simulation. The simulation model described in the previous chapters would calculate the half-shaft torque and then the longitudinal force which would ultimately give the actual vehicle speed but on the HiL the half-shaft torque cannot be calculated so it is measured instead via two torque sensors. The torque sensors are supplied by HMB, and are spec’d to 2000 Nm and are connected to the SPARC which feeds back to the dSPACE MicroAutoBox. The torque sensors accuracy is affected by various factors such as temperature, vibration, humidity and alignment. The alignment with the torque sensor housing is crucial and as such ROTA DS couplings are used on each torque sensor. The couplings are torsionally very stiff but allow some movement to account for vibrations and whirling of the half-shafts, keeping the torque sensor aligned with the base.

The dSPACE communicates with the SPARC Controller to give a required hub speed demand and receives the hub torque/speed signals over one CAN (Computer Area Network). A second CAN connects to the ‘input motor’ inverter ECU for the 70 kW motor to send a required torque demand and receive various data. A schematic of the CAN network is shown below in Figure 5-7.

The CAN signals between the MicroAutoBox and the SPARC Controller were set by configuring CAN transmission and receiver Simulink blocks supplied by dSPACE to be used in the vehicle models. The Simulink blocks can be configured to set the message identifier and ordering type using either little-endian or Motorola, and the start bit/length for each signal. The equivalent settings are input into the SPARC controller. Configuring the CAN between the ECU and MicroAutoBox was a little more complex due to the fact the message order was already set in the ECU so needed replicating in the Simulink environment, i.e. start bit, length, etc. A block was developed to break down a UINT8 signal into separate binary bits which could then be reordered and converted back into UINT8 to be input into the “CAN Tx” blocks or vice versa for the “CAN Rx” blocks. An additional issue with the
inverter ECU CAN was that the signals transmitted on the CAN required a CRC (Cyclic Redundancy Check) to be generated.

The CRC builds an inspection value out of the data bytes, including message counter. The 8-bit CRC method defined in the industry standard regulation SAE J1850 is used based on the polynomial: $x^8 + x^4 + x^2 + x + 1$. The initial value for the calculation is 0xFF, the result is inverted at the end. Usually, the CRC is not calculated directly but taken from a lookup-table that contains all 256 feasible results. Hence, the CRC calculation e.g. for an 8-byte message looks as follows (CRC is in Byte 8):

\[
\begin{align*}
\text{CRC} &= 0xFF \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE1}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE2}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE3}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE4}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE5}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE6}] \\
\text{CRC} &= \text{crctable}[\text{CRC} \oplus \text{BYTE7}] \\
\text{CRC} &= \text{CRC} \oplus 0xFF
\end{align*}
\]

Both sender and receiver of a CAN message generate a CRC, so the receiver can compare both values and check they match. If the messages being sent and received differed, there was some data corruption or loss, and thus the transmitted signals would not be used and a failure condition would result. In case of a failure condition protocols have been programmed into the Simulink model to remove the input torque instantaneously and
ramp down the hub motor speeds at a controlled rate. The failure protocol will result from various scenarios, i.e. low voltage, no power, overheating, uncontrollable torque, etc.

A summary of the CAN signals for the 2SED transmission is shown in Figure 5-8, where the TCU signals would not exist for the single-speed transmission.

The entire HiL test rig is controlled by HMI (Human Machine Interface) software, which links with the dSPACE MicroAutoBox, called ‘Control Desk’. The software pulls fixed initial parameters from the initialisation file for the Matlab/Simulink model ‘m file’ which can be configured in the HMI as modifiable values, i.e. vehicle mass, drive cycle, etc. The software also allows data to be displayed in real time via graphs, dials, and numerical displays which is selected from any input or output of the blocks within the Simulink model so is easily configured. Plots can be overlaid with multiple values to compare results.

The HMI was designed so that tests such as different driving cycles and performance tests could simply be initiated through selecting an option on a drop down box. The vehicle parameters can be selected via the HMI prior to initiating the test. The HMI was also designed to allow individual control of the input torque demand of the electric motor and speed demand of the ‘hub motors’ for static load point testing.

![Figure 5-8: Summary of the CAN signals for the 2SED transmission](image)

5.3 VALIDATION

The initial static tests on the HiL rig to ensure the system was functioning as expected were carried out successfully, with a torque being applied by the 70 kW motor and the hub motors turning at the desired speed.

A series of low torque performance tests were initially carried out to understand the response of the driving motor and check the half-shaft torques against the simulated values. The results of the initial acceleration tests, where throttle is instantly applied at a
set level, showed that the electric motor was characterized by a significant rise time, the simulation model used prior to this had assumed a much lower rise time, i.e. instantaneous. A rise time of 0.19 was calculated and added to the simulation model (simulated through a transfer function) to replicate the actual electric motor response. The new electric motor torque imitated the actual electric motor response almost perfectly, and is illustrated in Figure 5-9, where the increase in the accuracy of the simulation through validating the model against physical testing can be quickly recognised. The simulation does not model the high frequency oscillations due to resonance in the rig and electric motor characteristics.

![Figure 5-9: Electric motor torque comparisons for longitudinal acceleration test - 25% throttle (red - measured, blue - simulation)](image)

The hub torques for the same manoeuvre as in Figure 5-9 are plotted in Figure 5-10, where the simulated half-shaft torque can be seen to be much higher than the simulated half-shaft torque, so required some investigation. The measured half-shaft torque is simply a function of the input torque, gear ratio and the efficiency of the transmission. The gear ratio is fixed, for both the transmission and the simulation model and as a first approximation it can be assumed that the input torque is the same for the simulation model and the test rig. Therefore the difference between the simulation model and the test rig measured value must be due to an incorrect transmission efficiency being used in the simulation model, i.e. too high.

The simulation model has two efficiency maps, as previously mentioned, namely one for the gearbox and one for the differential which need altering to match the characteristics seen on the test rig. However, on the test rig only the efficiency of the whole transmission including CV joints can be measured so the model needed modifying to account for a single efficiency map.
The efficiency of the transmission can be found on the test rig through measuring the power into the transmission and the power out of the transmission. The power can be found by multiplying the torque and the speed, and therefore the efficiency of the transmission, $\eta_{\text{transmission}}$, is given by,

$$\eta_{\text{transmission}} = \frac{P_{\text{out}}}{P_{\text{in}}} = \frac{T_{\text{motor,70kW}}\omega_{\text{motor,70kW}}}{(T_{\text{hub,left}}\omega_{\text{hub,left}}) + (T_{\text{hub,right}}\omega_{\text{hub,right}})}$$ (121)

$P_{\text{out}}$, is the power out of the transmission, $P_{\text{in}}$, is the power into the transmission, $T_{\text{motor,70kW}}$, is the torque of the 70kW motor, $\omega_{\text{motor,70kW}}$, is the angular speed of the 70kW motor, $T_{\text{hub,left}}$, is the torque at the left hub, $T_{\text{hub,right}}$, is the torque at the right hub, $\omega_{\text{hub,left}}$, is the angular velocity at the left hub and $\omega_{\text{hub,right}}$, is the angular velocity at the right hub.

The torque and speed of the 70 kW electric motor are found over the ‘input motor’ inverter ECU CAN. The ‘input motor’ speed is accurately measured according by a hall sensor, however, the electric motor torque is calculated from the current drawn and may not be completely accurate. The hub speeds are measured by independent speed sensors on each electric motor whilst the hub torques are measured by torque sensors supplied by HBM which were calibrated on site and are accurate to +/- 0.1-1 Nm.

A set of test procedures were generated in Simulink that could be easily selected from the HMI to measure the efficiency of the transmission at different load points. The transmission efficiency was measured from 10 to 250 Nm at 10 Nm intervals and from 250 to 7500 rpm at 250 rpm intervals. The data was collected at fixed torques and then the speed ramped up between speed intervals and held at each speed for 5 seconds so an average can be taken improving the accuracy of the results. Each test was saved through the HMI and post-processed via Matlab scripts.
The need to collect the data over a set period and average it is clearly shown in Figure 5-11 and Figure 5-12.

The torque and speed signals can be seen to oscillate at a high frequency so taking an instantaneous value would result in an incorrect value. The traces appear to show a pattern which repeats every one second so although an average over one to two seconds may suffice a five second period was selected to increase reliability as the oscillation period may change for different torque/speed combinations.

The repeatability of the efficiency test procedure was analysed by carrying out multiple tests at the load site in Figure 5-11 and Figure 5-12. The results are shown in Table 5-1 where the efficiency can be seen to remain consistent for each of the first three tests. The result was also attained for averaging the data over a 10 second time period to understand if the result altered, which it did not.
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Hardware-in-the-loop test rig development: Validation Results

Table 5-1: Repeatability test for the efficiency test procedure at 3250 rpm and 70 Nm – single-speed

<table>
<thead>
<tr>
<th>Speed [rpm]</th>
<th>Test rig result</th>
</tr>
</thead>
<tbody>
<tr>
<td>3250</td>
<td>3250</td>
</tr>
<tr>
<td>3250</td>
<td>3250</td>
</tr>
<tr>
<td>3250</td>
<td>3250</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Torque [Nm]</th>
<th>70</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>70</td>
<td>70</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Period [s]</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
</tr>
</tbody>
</table>

| Efficiency [%] | 89 | 88 | 87 | 88 |

The efficiency tests were carried out at each of the torque and speed sites, post-processed and input into an array to create the transmission efficiency map for the simulation model. The acceleration tests were simulated with the new transmission efficiency map and the hub torques were found to be almost identical to the measured values, as shown in Figure 5-13.

Figure 5-13: Hub torque comparison for longitudinal acceleration test – single-speed - 25% throttle (red/blue – measured (left/right), green - simulation)

5.4 VALIDATION RESULTS

Performance tests were carried out at 25 %, 50 %, 75 % and 100 % (full throttle) on the HiL rig and compared against simulation results. The simulations were carried out with the baseline model, and then with the validated motor time constant and experimental efficiency maps. The results are shown in Table 5-2, Table 5-3, Table 5-4 and Table 5-5 respectively.
The results show a clear increase in the accuracy (from the reducing percentage difference) of the predicted simulation acceleration times due to the validation of the electric motor rise time and the experimentally attained efficiency map.

Table 5-2: 25 % Throttle acceleration test results for the HiL test rig and simulation (single-speed)

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 10 km/h</td>
<td>4.83</td>
<td>4.10</td>
<td>15.22</td>
<td>4.32</td>
<td>10.56</td>
<td>4.73</td>
<td>2.17</td>
</tr>
<tr>
<td>0 - 20 km/h</td>
<td>9.54</td>
<td>8.29</td>
<td>13.16</td>
<td>8.51</td>
<td>10.85</td>
<td>9.40</td>
<td>1.52</td>
</tr>
</tbody>
</table>

Table 5-3: 50 % Throttle acceleration test results for the HiL test rig and simulation (single-speed)

<table>
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<th></th>
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<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 10 km/h</td>
<td>2.34</td>
<td>1.855</td>
<td>20.73</td>
<td>2.06</td>
<td>11.97</td>
<td>2.22</td>
<td>5.13</td>
</tr>
<tr>
<td>0 - 20 km/h</td>
<td>4.40</td>
<td>3.73</td>
<td>15.13</td>
<td>3.94</td>
<td>10.47</td>
<td>4.23</td>
<td>3.75</td>
</tr>
<tr>
<td>0 - 30 km/h</td>
<td>6.48</td>
<td>5.63</td>
<td>13.19</td>
<td>5.83</td>
<td>10.03</td>
<td>6.30</td>
<td>2.85</td>
</tr>
<tr>
<td>0 - 40 km/h</td>
<td>8.76</td>
<td>7.75</td>
<td>11.48</td>
<td>7.92</td>
<td>9.59</td>
<td>8.60</td>
<td>1.77</td>
</tr>
</tbody>
</table>

Table 5-4: 75 % Throttle acceleration test results for the HiL test rig and simulation (single-speed)

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 10 km/h</td>
<td>1.58</td>
<td>1.21</td>
<td>23.49</td>
<td>1.40</td>
<td>11.11</td>
<td>1.50</td>
<td>5.08</td>
</tr>
<tr>
<td>0 - 20 km/h</td>
<td>2.91</td>
<td>2.42</td>
<td>16.87</td>
<td>2.62</td>
<td>9.98</td>
<td>2.81</td>
<td>3.44</td>
</tr>
<tr>
<td>0 - 30 km/h</td>
<td>4.25</td>
<td>3.64</td>
<td>14.47</td>
<td>3.84</td>
<td>9.76</td>
<td>4.13</td>
<td>2.82</td>
</tr>
<tr>
<td>0 - 40 km/h</td>
<td>5.71</td>
<td>5.00</td>
<td>12.43</td>
<td>5.17</td>
<td>9.54</td>
<td>5.59</td>
<td>2.19</td>
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<td>0 - 50 km/h</td>
<td>7.65</td>
<td>6.81</td>
<td>11.05</td>
<td>6.93</td>
<td>9.48</td>
<td>7.52</td>
<td>1.70</td>
</tr>
</tbody>
</table>
Table 5-5: 100 % Throttle acceleration test results for the HiL test rig and simulation (single-speed)

<table>
<thead>
<tr>
<th>Test</th>
<th>Physical</th>
<th>Simulation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[Baseline]</td>
<td>[Baseline]</td>
</tr>
<tr>
<td>0 - 10 km/h</td>
<td>1.33</td>
<td>0.98</td>
</tr>
<tr>
<td>0 - 20 km/h</td>
<td>2.24</td>
<td>1.96</td>
</tr>
<tr>
<td>0 - 30 km/h</td>
<td>3.53</td>
<td>2.95</td>
</tr>
<tr>
<td>0 - 40 km/h</td>
<td>4.68</td>
<td>3.99</td>
</tr>
<tr>
<td>0 - 50 km/h</td>
<td>6.14</td>
<td>5.31</td>
</tr>
<tr>
<td>0 - 60 km/h</td>
<td>8.05</td>
<td>6.98</td>
</tr>
</tbody>
</table>

An additional function of the test rig is to measure the energy consumption over standard driving cycles. Through running driving cycles on the test rig the energy consumption predictions for driving cycles will be more accurate than simulation, due to the fact that the non-linear characteristics inherent in a complex system that are difficult to simulate are included in the test rig.

An NEDC driving cycle was initially run with the single-speed transmission installed on the test rig. The challenge for the test rig was to follow the driving cycle speed profile accurately, as the variations in torque input require careful control of the hub torque to maintain the correct speed. The regulations state that at constant velocity the vehicle must remain within +/-2 km/h of the required driving cycle speed profile and +/- 4 km/h in acceleration and deceleration. Figure 5-14 shows the hub speeds during the NEDC for a single-speed transmission test where the speed limits are shown by the red lines. The hub speeds stay well within the required speed limits and follow the drive cycle speed profile accurately despite the high frequency oscillations arising from the electric motor characteristics. This is true, not only during constant speed and acceleration, but also when the system is making an abrupt change from acceleration to braking or a similar manoeuvre.

Regenerative braking is a key feature of an electric vehicle drivetrain as it allows the system to recover energy having a significant reduction in energy consumption. The HiL test rig is capable of allowing the ‘input motor’ to run in regeneration mode as on a vehicle by dissipating the regenerated power via a resistor bank on the battery simulator. The electric motor torque during the NEDC driving cycle is shown in Figure 5-15, where the required
torque demand is accurately followed by the electric motor. The high frequency oscillations are due to vibrations in the rig and the characteristics of the motor.

Figure 5-14: Hub speed comparison for NEDC drive cycle – Single-speed (green/blue = hub speed (left/right), red = hub speed limits)

Figure 5-15: Electric motor torque comparison for NEDC drive cycle - Single-speed (red = requested, blue = actual (from inverter ECU))

Figure 5-16 illustrates a section of an FTP 75 driving cycle showing the power at the electric motor shaft of the test piece and at the wheel hubs, for the simulated model and the test piece on the rig. The difference between the power output of the electric motor and the power at the hubs is due to the friction losses and the inertial torque across the gearbox,
Hardware-in-the-loop test rig development: Validation Results

differential and half-shafts. There is a good match between simulation and experimental testing which further validates the simulation model.

![Graph showing simulated and experimental output powers at the electric motor and the wheel hubs](image)

Figure 5-16: Simulated and experimental output powers at the electric motor and the wheel hubs, (blue = experimental motor power, black = simulated motor power, green = experimental hub power, red = simulated hub power)

The test rig is capable of measuring the energy consumption through reading the voltage and current drawn at the battery simulator power supply. An NEDC was carried out on the test rig with the single-speed transmission and then the same driving cycle was simulated including the model validation described in this section, i.e. motor time constant and validated efficiency map. The results are given in Table 5-6 and shows that the energy consumption calculation from the model was 96% accurate with respect to the data measured from the rig.

<table>
<thead>
<tr>
<th>NEDC</th>
<th>Simulated [kWh]</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.17</td>
<td>3.03</td>
<td></td>
</tr>
</tbody>
</table>

The test rig is also capable of carrying out tests such as tip-ins and tip-offs which provoke driveline stress and vehicle acceleration oscillations to investigate a vehicles drivability. The measurement of the vehicle acceleration and consequently the vehicle jerk is an accurate representation of the drivability of the vehicle. The large vehicle speed variations due to the nature of the tests makes the vehicle speed profiles difficult to follow and were a challenge for the tuning of the test rig hub speed controllers. Figure 5-17 illustrates the hub speeds for a tip-in test carried out at 30 km/h with a 90% throttle demand where the hub speed can be seen to follow the required speed and remain within the speed limits.

The difficulty in maintaining the required hub speed lies in the fact there is a large torque variation despite the wheel speed remaining constant, i.e. 0-100% throttle. This is not an issue on an actual vehicle due to the large equivalent inertia of the vehicle at the wheel resisting any wheel speed variation, unless wheel slip occurs. The SPARC controller contains
the software to convert the hub speed demand into a torque demand which initially consisted of a feed-back controller using a PID which was found to be insufficient. The source code was altered and some data sent to the SPARC from the dSPACE controller to give an element of feed-forward control, i.e. allow the system to pre-empt the torque demand, and resulted in the acceptable behaviour seen in Figure 5-17.

![Figure 5-17: Hub speeds during tip-in test at 30 km/h for a 90 % throttle application (green/blue = hub speeds (left/right), red = hub speed limits)](image)

The validated motor time constant and efficiencies were added to the simulation model to ascertain the peak of jerk for a standard tip in test at 10 km/h for 100 % throttle, as seen in Figure 5-18. The peak of jerk can be seen to be highest for first gear of the 2SED which has the highest overall transmission ratio of the three fixed gear case studies. Furthermore, the test conducted in second gear of the 2SED which has the lowest overall transmission ratio has the lowest peak of jerk.

![Figure 5-18: Peak of jerk comparison](image)

5.5 GEARSHIFT ANALYSIS

The two-speed transmission was tested extensively on the HiL test rig in fixed gear, carrying out performance tests and drive cycles as shown in the previous sections. However, to test
the 2SED or in fact any multiple-speed transmission for a standard acceleration test, performance manoeuvre or driving cycle it is necessary to perform a gearshift.

To properly control the test rig whilst a gear change is taking place it is necessary to understand the drivetrain dynamics throughout a gear change. During a gearshift for a manual transmission the clutch needs to be disengaged to allow the synchronisers to be moved which creates a torque interrupt at the wheels. This can be felt during an upshift as a lurch forward as the vehicle momentarily decelerates when the clutch pedal is depressed. This leads onto the most important factor to consider during the gearshift; that on a vehicle, despite the torque interrupt, the wheel rotational acceleration will remain relatively constant due to the inertia of the vehicle and the adhesion between the tyre and the road surface resisting the change in torque. On the HiL test rig, the inertia of the vehicle is not physically represented so the hub motors have to be properly controlled to maintain the required speed despite the large variations in torque and is in itself a major challenge.

The torque interrupt with a manual transmission is severe due to the complete disengagement of the clutch, and consequently a gap in engine/motor torque transfer. However the 2SED boasts a seamless gearshift similar to a DCT (as explained earlier) so is easier to control as the variation in torque at the wheels during the shift is significantly reduced. It should be noted that the 2SED design can provoke a sharp variation in torque when the one way sprag clutch is engaged as the teeth lock up. Nevertheless, accurate control of the friction clutch during the downshift will make this phenomenon less harsh and perceivable.

The HiL test rig uses a SPARC controller to control the hub motors as previously explained and simply receives a wheel speed request from the dSPACE MicroAutoBox. The SPARC controller uses a PID system and internal controllers to convert the required wheel speed sent from dSPACE into a required motor torque to achieve and maintain the required speed. In this way, the SPARC controller is working in “Speed control” however is capable of functioning in “Torque control” where the SPARC receives a torque signal from the dSPACE and simply forwards it directly to the motor, however then the speed controller would need to be implemented in the dSPACE MicroAutoBox.

The gearshift request is sent from dSPACE to the 2SED TCU and in initial tests the SPARC was not made aware of if or when a gearshift was taking place. The SPARC controller would simply react to the change in wheel torque whilst the wheel speed demand remained constant via the internal controllers. An initial test is illustrated in Figure 5-19 for a first to second gear upshift where the motor speed and equivalent motor speeds in first and second are shown. The figure shows a gear shift where the hub speed is “dragged down” by the hub torque remaining too high for too long into the upshift.

During an upshift the hub torque reduces due to a change in gear ratio and this reduces in the 2SED proportionally to the increasing clutch force (or reducing displacement as shown in Figure 5-20). The SPARC controller needs to ask for enough torque to oppose the transmission output torque and maintain the required hub speed. In Figure 5-19 the shift fails as the SPARC does not react quickly enough to the change in hub torque. The SPARC is
initially requiring enough torque to maintain the required wheel speed in first gear but as
the hub torque reduces to the second gear value the hub torque does not reduce to the
lower second gear value quickly enough and remains at the first gear value pulling the
wheel speed down.

![Figure 5-19: Illustrating incorrect hub speed control during an upshift at 100 Nm and 1000 rpm. (red = motor speed, black = equivalent motor speed in first gear, green = equivalent motor speed in second gear)](image)

The SPARC controller was modified to allow the SPARC controller to pre-empt hub torque
variation and maintain the desired hub speed. A feedforward value calculated in the
MicroAutoBox from the friction clutch position in the 2SED (read from the TCU CAN) was
sent to the SPARC. A shift request signal was also generated so the SPARC controller would
know when to use the feedforward signal. The feedforward signal was generated from the
electric motor torque, the gear ratio and the clutch displacement signal received from the
2SED TCU. The electric motor torque during the experimental upshift is illustrated in Figure
5-21 and shows the momentary drop in requested motor torque during the inertia phase of
the upshift allowing the motor to reach the equivalent second gear speed quicker. The
inertia phase lasts approximately 0.25 seconds. The hub torques and feedforward signal are
shown in Figure 5-22 where the feedforward signal follows the hub torque accurately. In Figure 5-22 the feedforward signal does not line up exactly with the actual hub torque, this is primarily due to the efficiency of the transmission, inertias and clutch efficiency not being included in the feedforward controller equations. Any discrepancies between the required torque and feedforward torque are controlled by the SPARC PID and internal controllers (i.e. the feedback).

Figure 5-21: Electric motor torque during an upshift at 100 Nm and 3000 rpm. (black = actual motor torque, red = requested motor torque)

Figure 5-22: Hub torques during an upshift at 100 Nm and 3000 rpm. (red/black = hub torques (left/right), green = feedforward hub torque)

In addition to the feedforward signal HORIBA visited the test rig and retuned the PID values in the SPARC Controller along with adding some additional code to reduce the reaction time. The result of the improvements can be seen in Figure 5-23 below where the upshift
results in only minimal variations in the wheel speed. Only a small spike in wheel speed is evident which is due to the hub torque reducing before the sprag clutch disengages.

![Graph showing hub speeds during an upshift at 100 Nm and 3000 rpm.](image)

**Figure 5-23**: Hub speeds during an upshift at 100 Nm and 3000 rpm. (red = motor speed, black = equivalent motor speed in first gear, green = equivalent motor speed in second gear)

## 5.6 HYBRID APPLICATION

The test rig was used to validate a model used in an activity to investigate the drivability of a TTRP hybrid electric vehicle.

Drivability is an important factor that needs to be considered when developing the drivetrain, ICE/electric motor control, gearshift control, suspension system and powertrain mounting system of any vehicle. The low frequency drivability of a vehicle is generally measured through acceleration and jerk (rate of acceleration) profiles during set manoeuvres such as acceleration and tip-in tests (Dorey and Holmes, 1999; Sorniotti, 2008), with a magnitude of jerk over 10 m/s³ being considered unacceptable according to some sources (Huang and Wang, 2004). Some authors also consider the frequency of the jerk oscillations and the RMS (Root Mean Square) jerk, stating that a value of jerk up to about 25 m/s³ can be tolerated for frequencies less than 3 Hz (Shouren, 1984).

A conventional parallel HEV layout consists of an ICE and an electric motor placed on the same axle with the torque of each power source driving the same wheels (Arata et al., 2011). A significant body of literature has been published concerning the control of the mode transitions and gearshifts within parallel HEVs (for example, Gupta, Landge and Seth, 2009; Shin et al., 2010). Kim et al. (2009) explains that the HEV drivability can be improved through controlling the clutch slip.

The drivability in conditions of constant gear is particularly relevant when considering a TTRP HEV as two power sources (one for each axle), each with different steady-state and dynamic characteristics, are both simultaneously providing torque, and are coupled to transmission systems with different parameters (gear ratios, inertias and torsion stiffness).
In a TTRP HEV each axle is driven, and thus the vehicle is All-Wheel-Drive (AWD), consequently benefitting from increased traction capabilities. A HEV in this configuration can work in three modes: FWD driven by the ICE in the case study vehicle, Rear-Wheel-Drive (RWD) driven by the electric motor (centrally located and connected to the wheels through half-shafts in the specific application of this article), and AWD driven by both axles in a parallel layout. This arrangement allows each driving mode to be adopted for specific driving conditions and the electric motor to be utilised to reduce the ICE fuel consumption through the improvement of the location of the ICE operating points and the implementation of brake regeneration. Moreover, the electric motor improves vehicle performance and/or allows the ICE to be downsized. The work of Sorniotti et al. (2011) deals with the seamless gearshift control of an electric axle for fully electric vehicles or TTRP HEVs.

As the ICE and electric motor have different torque and power curves and efficiency maps, the operating points need to be carefully controlled. The ideal electric motor torque characteristic (as a function of motor speed) is theoretically very favourable from the viewpoint of vehicle response, due to the constant torque achievable at low motor speeds, going into a constant power region for high values of electric motor speed. The lack of combustion and consequent torque fluctuations typical of internal combustion engines forego the need of a clutch damper. However, the lack of the clutch damper eliminates the main damping component within the transmission, and can give rise to non-optimal drivability, especially if there are significant plays within the transmission system (Amann, Böcker and Prenner, 2004). In addition, the ease of control and typically low reaction time of electric motor drives allow an effective implementation of anti-jerk and motor torque control algorithms to reduce driveline oscillations. The potentially low reaction time of the electric motor drive also improves the driver’s subjective rating of vehicle responsiveness when an abrupt driver torque demand is applied, but can also excite undesired drivetrain oscillations.

Non-linear powertrain models were developed to simulate each separate drivetrain component of the TTRP HEV, along with a non-linear vehicle chassis model to analyse vehicle motion and pitch dynamics. The overall model is characterised by 16 degrees of freedom.

The ICE drivetrain model (from the engine to the half-shaft) has two degrees of freedom due to the non-linear torsional dynamics of the clutch damper. An additional degree of freedom can be represented by the internal dynamics of the differential gearset, which gives rise to a different angular speed on the left and right sun gears. This is relevant only in case of uneven friction coefficients on the two tyres of the same axle, or asymmetric half-shafts.

The rear axle consists of the 2SED and 70kW PMSM motor used throughout this project. The equations governing the vehicle model dynamics are very similar to those explained previously in this report.
The front ICE-driven axle was validated against data from a FWD test vehicle, and the rear axle against results attained from the 2SED installed on the Hardware-in-the-Loop (HIL) test rig.

The test data for the ICE-driven vehicle have been collected from a FWD vehicle fitted with a drivetrain comprised of an ICE and a five-speed automated manual transmission. The ICE test vehicle parameters are in Appendix D. The test vehicle carried out tip-in tests in conditions of fixed gear with a starting speed ranging from 12 km/h for first gear to 45 km/h in fifth gear, in order to perform the manoeuvres at similar initial values of engine speed. During the initial part of a tip-in test the vehicle maintains a constant speed. This requires a torque demand, expressed as a percentage of the maximum engine torque, which has a value that is a function of the gear ratio and initial vehicle speed. Then the driver applies a throttle input at a set rate, ideally according to a step input. In practice, the actual torque applied by the driver is not a step input but has been recorded during the experimental tests.

The results from the simulation model and the test data have been overlapped in Figure 5-24 for a first gear tip-in test, comparing the engine torque, vehicle speed, acceleration and jerk. The same torque demand profiles as the ones recorded during the experimental tests were adopted during the simulations. The figures show that the simulated results accurately follow the results of the test data, applying the correct amount of engine torque, and provoking the correct vehicle speed, acceleration and jerk response. A similar fit between the experimental and simulation results has been achieved for all the tip-in manoeuvres in the different gear ratios.
Hardware-in-the-loop test rig development: Hybrid Application

The rear axle of the TTRP HEV has been validated using a similar methodology to the front ICE-driven axle. The TTRP HEV simulation model was adapted to only be driven by the rear axle, with the synchronisers of the ICE transmission open, removing any engine torque from the front wheels. The electric rear axle was validated against the HiL test rig by simulating tip-in tests and then carrying out tip-in tests on the test rig.

The tip-in tests were simulated at varying initial speeds, 10 km/h, 30 km/h, 50 km/h and 70 km/h in both gears on both the model and the HiL test rig. An example of a test is shown in Figure 5-25, where the simulated results accurately overlap the experimental results. The vehicle model is identical for both the simulation and the model employed in the HiL test rig.
Through proving the accuracy of each drivetrain model separately following the validation methods presented previously, the whole TTRP HEV model can be thought to be reliable and suitable for predictive analysis and anti-jerk control design. The main vehicle parameters of the case study adopted in this section are reported in the Appendix D.

The drivability of the TTRP HEV has been analysed by simulating tip-in tests to consider the drivetrain response and vehicle acceleration in the time domain. In addition, a sensitivity analysis was carried out to research the effect of the torque distribution between the front and rear axles on the drivability, for the same overall value of wheel torque. In fact, depending on the energy efficiency maps of the two powertrains, driving mode, state of charge of the battery, the TTRP HEV supervisory controller (energy management system) can decide to split the torque demand between the two axles in a variety of possible distributions, provided that the torque demand is not so high as to require full torque from both powertrains. As a consequence, the average steady-state vehicle acceleration will be the same for whatever torque split, however the jerk dynamics during the transient may be very different, depending on the torque distribution and the selected gear for each axle. In order to calculate the necessary torque demands to give the required wheel torque distribution and overall torque, the model initially computes the required steady-state values of the total front and rear wheel torques during a manoeuvre for the assigned accelerator pedal input. The model then backwards calculates the ICE and electric motor torque demands in steady-state conditions to provide the target wheel torque distribution level whilst retaining the required total steady-state wheel torque value during the manoeuvre.

The acceleration profiles of a tip-in test carried out at 27 km/h (initial speed), with the ICE transmission in third gear and with the rear transmission in first and second gear, are presented in Figure 5-26 and Figure 5-27 respectively, for different wheel torque distributions. Both figures show an increase of the vehicle longitudinal acceleration oscillations for an increase of the engine driven axle torque. In particular, the electric axle of the specific vehicle application does not give rise to any significant oscillations during the second gear test, and low amplitude oscillations during the first gear test, whilst the engine-driven axle always produces significant oscillations. The experience of the author of this contribution is that the electric axle results can significantly vary depending on the set of vehicle parameters. In particular the response time and the mass moment of inertia of the electric motor drive are different, for example, for permanent magnet and switched reluctance machines.

In Figure 5-26 and Figure 5-27, the variety of the TTRP HEV responses for the same steady-state value of longitudinal acceleration level is very wide. The rear drivetrain wheel torque is lower when in second gear, which gives rise to lower acceleration values, as can be seen in Figure 5-27.
Hardware-in-the-loop test rig development: Hybrid Application

Figure 5-26: Sensitivity analysis of the front/rear axle torque distribution during a tip-in test at 27 km/h; front transmission: third gear, rear transmission: first gear

Figure 5-27: Sensitivity analysis of the front/rear axle torque distribution during a tip-in test at 27 km/h; front transmission: third gear, rear transmission: second gear

The response seen in Figure 5-26 and Figure 5-27 is a function of the vehicle parameters. For example, in a parallel research activity conducted at the University of Surrey based on the linear model of a TTRP HEV in the frequency domain, Morina (2010) has observed that the first natural frequencies of the two powertrains are very close to each other.

The results of the two powertrain validation tests and of the TTRP HEV sensitivity tests demonstrate the need for an anti-jerk controller, which is required to dampen the oscillations and make vehicle response consistent for the different combinations of wheel torque demands and gear ratios on the two axles. A first example of model-based anti-jerk control system was designed and implemented in the TTRP HEV model to improve the vehicle response during abrupt torque demands, in conditions of constant gear ratio. A schematic of the anti-jerk control system is shown in Figure 5-28. The system only modifies the electric motor torque demand as the typically fast rise time and high level of torque controllability of the electric axle result in a more effective anti-jerk control system than one based on the control of the ICE. The ICE keeps the empirical anti-jerk control (not detailed here) of the original engine control unit (based on the operating parameters of the
ICE), whilst the electric axle implements the supervisory anti-jerk control function. In the diagram, $T_{\text{wheel\_demand}}$ is the wheel torque demand, $T_{\text{demand}}$ is the engine torque, $T_{m\_demand}$ is the electric motor torque, $T_{\text{etheor}}$ is the theoretical engine torque, $T_{m\_theor}$ is the theoretical electric motor torque, $T_{\text{hsf\_est}}$ is the estimated front half-shaft torque and $T_{\text{hrs\_est}}$ is the estimated rear half-shaft torque.

Two extended Kalman filters, based on the simplified models of the electric axle and the internal combustion engine powertrain, estimate the half-shaft torque on each axle, as demonstrated in Amann, 2004 and in Bottiglione, 2011 (the proposed filter has been adopted for this activity). The vehicle supervisory controller distributes the driver torque demand to the front and rear axles depending on a bias based on the operating conditions of the vehicle (powertrain temperatures, battery state of charge, etc.). The TTRP HEV control proposed in this article calculates the overall reference half-shaft torque for the combined front and rear axle, starting from the driver torque demand, the engine speed and the electric motor speed, through the torque characteristics of the two propulsion units. The total reference and estimated half-shaft torques are then compared and the difference is the input into a PID (Proportional, Integral, Derivative) controller to modify the electric motor torque demand. The demand modified to eliminate the typical oscillations, caused by the torsion dynamics in the drivetrain, which usually affect the vehicle acceleration response during tip-in tests.

![Diagram of anti-jerk control system](image)

Figure 5-28: Simplified schematic of the anti-jerk control system

Figure 5-29 shows the Bode plots for the open-loop and closed-loop transfer functions ($O\text{LT}F$ and $C\text{LT}F$ respectively) for the electric motor PID and the vehicle system. The formulations for the open-loop and closed-loop transfer functions are given in equations (122) and (123). The frequency response of the front and rear half-shaft torques has been obtained through linearisation of the equations of the system which have been implemented in a state-space formulation. Within Equations (122) and (123) and Figure 5-29, the transfer functions resulting from the linearisation of the extended Kalman filters
Hardware-in-the-loop test rig development: Hybrid Application

have been neglected due to their fast dynamics. The conventional rules for tuning the PID feedback control system can be applied in order to achieve the desired tracking bandwidth. $G_{PID,TTRP}$ is the total PID value.

$$OITF = G_{PID,TTRP}(s)\left(\frac{T_{hsf}}{T_{mdemand}}(s) + \frac{T_{hsr}}{T_{mdemand}}(s)\right)$$  \hspace{1cm} (122)

$$CLTF = \frac{G_{PID,TTRP}(s)\left(\frac{T_{hsf}}{T_{mdemand}}(s) + \frac{T_{hsr}}{T_{mdemand}}(s)\right)}{1 + G_{PID,TTRP}(s)\left(\frac{T_{hsf}}{T_{mdemand}}(s) + \frac{T_{hsr}}{T_{mdemand}}(s)\right)}$$  \hspace{1cm} (123)

Figure 5-29: Examples of Bode diagrams of the open-loop and closed-loop transfer functions for the feedback part of the electric motor based anti-jerk controller, for increasing values of the proportional gain P (indicated by the direction of the arrow in the figure)

Particular care must be taken when selecting the gains of the controller, in order to prevent frequent saturations of the electric motor drive and their effect on the integral part of the PID controller. In this respect, an anti-wind up layout will have to be evaluated in a future upgrade to the PID controller.

To determine the effectiveness of the motor PID controller the frequency response of the system considering the vehicle acceleration was analysed. The transfer function showing the vehicle acceleration for a requested combination of engine and electric motor demands is given below, where $\ddot{x}$ is the longitudinal acceleration of the vehicle, $\tau_{gf}$ is the total front transmission ratio and $\tau_{gr}$ is the total rear transmission ratio.

$$\ddot{x} = T_{edemand} \frac{\ddot{x}}{T_{edemand}}(s) +$$

$$\frac{G_{PID,TTRP}(s)\left(T_{edemand}\tau_{diff, f} + T_{mdemand}\tau_{diff, r} + T_{edemand}T_{mdemand}\tau_{gf}\right)}{T_{edemand}G_{PID,TTRP}(s)\left(\frac{T_{hsf}}{T_{edemand}}(s) + \frac{T_{hsr}}{T_{edemand}}(s)\right) + T_{mdemand}} -$$

$$\frac{\ddot{x}}{T_{mdemand}}(s) + G_{PID}(s)\frac{T_{hsr}}{T_{mdemand}}(s)$$  \hspace{1cm} (124)
Hardware-in-the-loop test rig development: Hybrid Application

For example, Figure 5-30 illustrates the overall frequency response of the vehicle acceleration with a basic proportional controller included, compared with the frequency response of the TTRP HEV with no PID anti-jerk controller, except the standard anti-jerk controller of the engine. It is supposed that the two inputs (torque demands) to the system are phase synchronous. A sensitivity analysis of the effect of the value of the proportional gain \( P \) of the controller on the vehicle acceleration is carried out, where the proportional controller assumes the values \( P^*, P^*/2 \) and \( P^*/4 \). The figure shows that a benefit is achieved by implementing an anti-jerk controller on the electric motor, due to the theoretically flat frequency response when the anti-jerk controller is implemented for a proportional gain value of \( P^*/4 \). This ideal behaviour is not realistic when considering the torque saturation of the electric motor which prevents achievement of the same response, however the frequency response characteristic provides a good insight into controller gain design. The transfer functions are affected by the time constant \( \tau_m \) of the electric motor air-gap torque dynamics (sometimes filtered at the power electronics control level for anti-jerk purposes).

The acceleration profile for a tip-in test with the anti-jerk control system is compared with the same manoeuvre without the anti-jerk control system in Figure 5-31. The gains adopted on the PID controller are relatively high, however the model considers the torque saturation of the motor drive and the benefits of the controller are evident. Proper gain scheduling will be required for a consistent application of this controller to any driving condition. The anti-jerk control system can be seen to significantly reduce the oscillations in the acceleration profile, increasing driver comfort.

![Figure 5-30: Frequency response of the system with and without the PID including a sensitivity analysis of the proportional gain, at 27 km/h, with a 50/50 front/rear percentage torque distribution (Front transmission: third gear, Rear transmission: first gear)](image-url)
Hardware-in-the-loop test rig development: Concluding remarks

Figure 5-31: Acceleration profile comparison with and without the anti-jerk control system during a tip-in test at 27 km/h, with a 50/50 front/rear percentage torque distribution (front transmission: third gear, rear transmission: first gear)

5.7 CONCLUDING REMARKS

The development of the hardware-in-the-loop electric drivetrain test rig was defined as a key objective at the outset of the PhD program. Work on the test rig has been ongoing for the duration of the project, where the first year focused on the delivery and installation of the components and the subsequent years spent on development, testing and tuning. At this point in time the test rig has been developed to the point of being operational with extensive functionality.

The test rig is capable of carrying out standard driving cycles and measuring the energy consumption successfully. The hub speeds can follow the drive cycle profile and stay within tolerances defined by the EU regulations. Furthermore, performance tests can be carried out including full throttle acceleration tests where the hub speeds follow the speed profile adequately.

The test rig has been used to validate the simulation model, defining the correct motor time constant and measuring the efficiency of the transmission.

A gearshift was successfully carried out on the test rig where the hub motors were properly controlled to accurately represent the required hub wheel speeds. To achieve this, the SPARC controller was modified heavily in a joint effort with Horiba along with the generation of a feedforward signal based on the motor torque demand, current gear and clutch position.

The future development of the test rig should focus on tuning the SPARC Controller to account for the inertial changes during a gearshift so driving cycles can be run with multiple-speed transmissions in all conditions. This would lead to the development of gearshift controllers, gearshift map validation and further studies on multiple speed transmissions which are a key topic of research for electric vehicles.

With the TTRP drivability study a comprehensive non-linear model of the longitudinal dynamics of a through-the-road parallel hybrid-electric vehicle has been developed and presented. Each drivetrain model was validated against real world test data in a wide range
Hardware-in-the-loop test rig development: Concluding remarks

of tip-in manoeuvres, proving the accuracy of the modelling methodology. A sensitivity analysis was then carried out through the variation of the torque distribution and gear ratios of the front and rear axles. A novel and significant contribution of the work has been the results of the sensitivity analysis which showed a major modification of vehicle acceleration and jerk dynamics induced by the variation of the torque distribution between the front and rear axles, for the same steady-state value of vehicle longitudinal acceleration. This could provoke a sense of inconsistent drivability and discomfort in the driver and the passengers. Moreover, the electric axle can induce larger driveline oscillations due to the potentially very low rise time of the electric motor drive, which is further aggravated by the lack of a clutch damper. A novel anti-jerk control system has been successfully proposed and implemented to improve the overall vehicle acceleration profile through the modification of the electric motor torque demand.
6 NOVEL 4-SPEED DUAL-MOTOR TRANSMISSION FOR ELECTRIC VEHICLES

6.1 INTRODUCTION

An additional area of research carried out during this project focused on a novel multiple speed transmission for electric vehicles. Rinderknecht, Meier and Fietzek, (2011), developed a seamless transmission system concept based on the adoption of two electric motor drives, each of them can be connected to either one of two gear ratios, giving origin to nine states. The drivetrain can also be characterised by an internal combustion engine for power regeneration or to provide a tractive torque, if a friction clutch is implemented allowing the internal combustion engine to be connected to/disconnected from the system. The original paper by Rinderknecht, Meier and Fietzek, (2011) explains the basic layout of the system concept and provides some hints of the possible advantages over single-motor electric drivetrains, however it does not supply any analytical tool or experimental proof of the actual achievable benefit. For reasons of standardisation and cost-effectiveness, the authors of Rinderknecht, Meier and Fietzek, (2011) suggest the adoption of the same electric machines and gear ratios on each primary shaft of the system, without any presentation of actual quantitative evaluation or design optimisation.

This section explains the mathematical equations and the analytical instruments required to evaluate the potential energy efficiency and performance benefits obtainable through this transmission concept. Also, the author provides an insight into the possible automated model-based design methodologies for the selection of the optimal state and torque distribution maps. Finally, the system will be evaluated for two case study vehicle applications, characterised by very different data sets, and compared with single-speed and two-speed single-motor drivetrains. The industrial companies involved in this project developed a physical prototype of this novel transmission concept (Bologna, Everitt and Fracchia, 2011).

6.2 THE NOVEL TRANSMISSION CONCEPT

Figure 6-1 is a schematic of the electric powertrain including the novel transmission, patented in Bologna, Everitt and Fracchia, (2011). This is characterised by an ‘odd’ electric machine, which is connected to the ‘odd’ primary shaft and, through a dog clutch, to either gear 1 or gear 3, and an ‘even’ electric machine, which is connected to the ‘even’ primary shaft and, through a dog clutch, to either gear 2 or gear 4. The gearshifts can be entirely operated through the control of the electric motor drive torques and the position of the electro-mechanical dog clutch actuators which drive barrel cams to select the gears. The high controllability inherent to electric motor drives permits the actuation of the gearshifts without the need for synchronisers, as the synchronisation is carried out electrically. This transmission can be coupled to a torque vectoring differential, therefore providing the energy efficiency benefit of a multiple-speed transmission and the vehicle dynamic performance of individual wheel powertrains, which have the packaging and weight-related
constraint of being characterised by a single-speed transmission. The dual-motor layout of this novel drivetrain concept allows a high load factor of the electric machines, when they are operated singularly, with a further potential increase of the overall energy efficiency depending on the motor characteristics. In the figure, $J_{\text{mot,odd}}$ is the moment of inertia of the ‘odd’ motor, $J_{\text{mot,even}}$ is the moment of inertia of the ‘even’ motor, $J_{1,\text{odd}}$ is the moment of inertia of the ‘odd’ primary shaft, $J_{1,\text{even}}$ is the moment of inertia of the ‘even’ primary shaft, $\tau_{g,1}$ is the first gear ratio, $\tau_{g,2}$ is the second gear ratio, $\tau_{g,3}$ is the third gear ratio, $\tau_{g,4}$ is the fourth gear ratio, $\eta_{g,1}$ is the first gear efficiency, $\eta_{g,2}$ is the second gear efficiency, $\eta_{g,3}$ is the third gear efficiency, $\eta_{g,4}$ is the fourth gear efficiency, $\tau_{\text{FRR}}$ is the final reduction ratio and $\eta_{\text{FRR}}$ is the final reduction gear efficiency.

Nine possible states characterise the system operating conditions: 1) only first gear engaged; 2) only second gear engaged; 3) only third gear engaged; 4) only fourth gear engaged; 5) first and second gears engaged; 6) second and third gears engaged; 7) third and fourth gears engaged; 8) first and fourth gears engaged; and 9) no engaged gear. However, the prototype transmission is incapable of operating in state 8, first and fourth gear, although it is considered for the research presented in this paper.

The adoption of this transmission layout implies a significant increase in flexibility when selecting the electric motor drives operating points. For example, Figure 6-2 plots the theoretical wheel torque obtainable in steady-state conditions (i.e., neglecting the angular acceleration of the drivetrain components and hence their inertial effects), at the peak torque of the electric motor drive/s, as a function of vehicle longitudinal velocity, for each possible state of the dual-motor drivetrain for a case study vehicle. Transmission efficiency has been neglected for simplicity in this single figure (but it will be considered in the calculations presented in the next sections of the contribution). The wheel torque characteristic can be subdivided into fifteen different areas (from A to O), each of which can be covered by a different number of states. The higher the number of states which can generate the same wheel torque and vehicle speed combination, the larger the chance of being able to achieve a higher operating efficiency of the overall system. The number of
alternative operating states of the drivetrain for each area is outlined in Table 6-1. In particular, for low torque and low speed conditions, the transmission permits the alternative selection of eight states (area H), whilst a significant number of alternatives is also allowed in further driving conditions. For example, in the torque envelope enclosing areas D-E-F-G-H-L-N, covered during normal driving conditions, at least three alternative states are selectable for each operating point. Due to the constant power characteristic of the electric motor drives, ‘even’ operating points located at the peak torque levels, such as those in the areas J, K and M, can be covered by multiple alternative states.

Figure 6-2: Theoretical wheel torque as a function of vehicle velocity for the different transmission states

Table 6-1: Number of possible states (second row of the table) which can generate the operating condition outlined in each sub-area (identified by the first row of the table) of Figure 6-1.

<table>
<thead>
<tr>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
<th>I</th>
<th>J</th>
<th>K</th>
<th>L</th>
<th>M</th>
<th>N</th>
<th>O</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
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<td>4</td>
<td>5</td>
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<td>5</td>
<td>4</td>
<td>5</td>
<td>2</td>
<td>3</td>
<td>1</td>
</tr>
</tbody>
</table>

6.3 THE ELECTRIC DRIVETRAIN DYNAMIC MODEL

The transmission can work in conditions of: i) no engaged gear on either primary shaft; ii) one engaged gear on one primary shaft and no engaged gear on the other primary shaft; and iii) two engaged gears, one per each primary shaft. As a consequence, each primary shaft is characterised either by an engaged gear and a disengaged gear or by two disengaged gears. If a gear is engaged, the primary shaft on which that gear is located will rotate with a constant gear ratio relative to the other components of the transmission, in particular the secondary shaft of the transmission and the differential case (if we neglect the mechanical play within the system). If both gears of the same primary shaft are disengaged, the electric motor shaft and the transmission primary shaft will rotate independently from the rest of the transmission. Therefore the overall system can be characterised by up to three degrees of freedom, one in case of an engaged gear for each
primary shaft, and three in case of a condition of two disengaged gears on each primary shaft.

The torque balance equation of the generic primary shaft (including the electric motor) when it constitutes an independent degree of freedom (two disengaged gears on that shaft) is given below. \( J_{\text{mot,odd/even}} \) is the moment of inertia of the 'odd'/‘even’ motors, \( J_{1,\text{odd/even}} \) is the moment of inertia of the ‘odd’/‘even’ primary shafts and \( \dot{\theta}_{\text{mot,odd/even}} \) is the rotational acceleration of the ‘odd’/‘even’ electric motors.

\[
T_{\text{mot,odd/even}} - T_{\text{loss,odd/even}} = (J_{\text{mot,odd/even}} + J_{1,\text{odd/even}})\dot{\theta}_{\text{mot,odd/even}}
\]  

(125)

The electric motor torque contribution \( T_{\text{loss,odd/even}} \) is particularly important, as it determines the decay rate of motor speed when either motor is disengaged from the rest of the drivetrain. The electric motor drive torque \( T_{\text{mot,odd/even}} \) takes into account the air gap torque dynamics with respect to the reference theoretical air gap torque (function of torque demand and electric motor speed) through a second order differential equation (or transfer function), and also the contribution caused by the windage losses of the electric motor drive, \( T_{\text{windage,odd/even}} \), which are expressed by a look-up-table as a function of motor shaft speed, as defined in equations (126) and (127). \( T_{\text{mot,del,odd/even}} \) is the delayed electric motor torque of the ‘odd’/‘even’ electric motors, \( T_{\text{windage,odd/even}} \) is the torque windage losses of the ‘odd’/‘even’ electric motors and \( \dot{\theta}_{\text{mot,odd/even}} \) is the rotational velocity of the ‘odd’/‘even’ electric motors. \( T_{\text{mot,ref,odd/even}} \) is the reference ‘odd’/‘even’ electric motor torque, \( DTD_{\text{mot,odd/even}} \) is the driver torque demand, \( \zeta_{\text{mot}} \) is the damping ratio of the electric motor air gap torque characteristic and \( \omega_{n,\text{mot}} \) is the natural frequency of the electric motor air gap torque characteristic.

\[
T_{\text{mot,odd/even}} = T_{\text{mot,del,odd/even}} - T_{\text{windage,odd/even}}(\dot{\theta}_{\text{mot,odd/even}})
\]  

(126)

\[
T_{\text{mot,del,odd/even}} = T_{\text{mot,ref,odd/even}}(\dot{\theta}_{\text{mot,odd/even}}, DTD_{\text{mot,odd/even}})
\frac{1}{1 + \frac{2\zeta_{\text{mot}}}{\omega_{n,\text{mot}}}s + \frac{s^2}{\omega_{n,\text{mot}}^2}}
\]

(127)

During a gearshift, when the angular speed difference between the electric motor shaft and the differential (referred back to the motor shaft) remains close to zero for a required amount of time, the gear is engaged and the system loses a mechanical degree of freedom, therefore equation (125) becomes irrelevant to the system dynamics, as it is incorporated into the overall torque balance equation at the differential. If the gear is disengaged during a gearshift, equation (125) is relevant again and is re-activated by resetting the initial conditions of the integral operator which calculates the angular velocity of the primary shaft starting from its acceleration, by using the last value of the motor shaft speed (calculated from the differential speed) before the disengagement of the dog clutch. The dog clutch actuator position is modeled through a time delay and a first order transfer function, equation (128). A first order transfer function is used to accurately simulate the physical properties of the actuator experimentally attained on the prototype transmission.
\( x_{\text{gear,act}} \) is the actual position of the gear actuator, \( x_{\text{gear,ref}} \) is the reference position of the gear actuator, \( \tau_{\text{gear,act}} \) is the time delay of the gear actuator and \( T_{\text{gear,act}} \) is the time constant of the gear actuator.

\[
x_{\text{gear,act}}(t) = x_{\text{gear,ref}}(t - \tau_{\text{gear,act}}) \frac{1}{1 + \frac{T_{\text{gear,act}}}{s}}
\]  

(128)

\( T_{\text{gear,act}} \) assumes different values depending on the direction of motion of the actuator, mimicking the experimental measurements carried out by the industrial partners of the project on the transmission prototype.

Equation (129) is the first approximation torque balance equation of the transmission components rotating together with the differential. Only one comprehensive formulation of the equation is reported here, despite each of the nine states of transmission operation requiring a unique variation of this equation. The variables \( \text{flag}_{\text{sel.even/odd}} \) are adopted to indicate whether a gear is engaged on the ‘even’ and/or ‘odd’ primary shaft, in order to include (in case of engaged gear) or exclude (in case of disengaged gear) the relating electric motor torque and the inertias of the motor, engaged gear and primary shaft within equation (129).

\[
T_{\text{mot.even}} T_{g,\text{sel.even}} \eta_{g,\text{sel.even}} T_{FRR} \eta_{FRR} \text{flag}_{\text{sel.even}} + \\
+ T_{\text{mot.odd}} T_{g,\text{sel.odd}} \eta_{g,\text{sel.odd}} T_{FRR} \eta_{FRR} \text{flag}_{\text{sel.odd}} + \\
- T_{hs_i} - T_{hs_r} = \\
= \left[ J_{\text{diff}} + J_2 T_{FRR}^2 \eta_{FRR} \eta_{g,\text{sel.even}}^2 + \right. \\
\left. \frac{\sum J_{g,\text{unsel.even}} T_{g,\text{unsel.even}} T_{FRR}^2 \eta_{FRR}}{\eta_{g,\text{sel.even}}} + \\
\frac{\sum J_{g,\text{unsel.odd}} T_{g,\text{unsel.odd}} T_{FRR}^2 \eta_{FRR}}{\eta_{g,\text{unsel.odd}}} + \\
\left( J_{\text{mot.even}} + J_{1,\text{even}} + J_{g,\text{sel.even}} \right) \text{flag}_{\text{sel.even}} T_{g,\text{sel.even}}^2 \eta_{g,\text{sel.even}}^2 T_{FRR}^2 \eta_{FRR} + \\
\left( J_{\text{mot.odd}} + J_{1,\text{odd}} + J_{g,\text{sel.odd}} \right) \text{flag}_{\text{sel.odd}} T_{g,\text{sel.odd}}^2 \eta_{g,\text{sel.odd}}^2 T_{FRR}^2 \eta_{FRR} \right] \dot{\theta}_{\text{diff}}
\]

(129)

\( T_{g,\text{sel.even}} \) is the gear ratio of the selected gear on the ‘even’ primary shaft, \( \eta_{g,\text{sel.even}} \) is the efficiency of the selected gear on the ‘even’ primary shaft, \( T_{g,\text{sel.odd}} \) is the gear ratio of the selected gear on the ‘odd’ primary shaft, \( \eta_{g,\text{sel.odd}} \) is the efficiency of the selected gear on the ‘odd’ primary shaft, \( J_{g,\text{unsel.even}} \) is the moment of inertia of the unselected gear on the ‘even’ primary shaft, \( T_{g,\text{unsel.even}} \) is the gear ratio of the unselected gear on the ‘even’ primary shaft, \( \eta_{g,\text{unsel.even}} \) is the efficiency of the unselected gear on the ‘even’ primary shaft, \( J_{g,\text{unsel.odd}} \) is the moment of inertia of the unselected gear on the ‘odd’ primary shaft, \( T_{g,\text{unsel.odd}} \) is the gear ratio of the unselected gear on the ‘odd’ primary shaft, \( \eta_{g,\text{unsel.odd}} \) is the efficiency of the unselected gear on the ‘odd’ primary shaft.
 shaft, \( \tau_{g,\text{unsel,odd}} \) is the gear ratio of the unselected gear on the ‘odd’ primary shaft, 
\( \eta_{g,\text{unsel,odd}} \) is the efficiency of the unselected gear on the ‘odd’ primary shaft.

The efficiency map of the transmission to be adopted in equation (129) has been derived from detailed models of the different transmission efficiency contributions available at the industrial company supporting this research and experimentally validated on other transmission systems with comparable mechanical characteristics. This model includes the contributions deriving from the bearings, the gear meshing, the windage and churning, and also the actuation losses. However, the efficiency values adopted for equation (129) do not include the losses due to the electro-mechanical actuation of the dog clutches, which are localised during the gearshift actuation phase. The efficiencies in equation (129) represent equivalent values and are split between the ‘odd’ side of the gearbox, the ‘even’ side of the gearbox and the final reduction gear. The efficiency of each contribution is computed as a function of the respective input torque to the transmission, primary shaft angular speed and operating temperature. The efficiencies in equation (129) have been considered for traction conditions of the powertrain, and can be reversed in case of a different sign of the input torque to the gear couplings. A lumped parameter model (i.e. equivalent thermal capacity with internal heat generation and heat exchange with the external ambient and adjacent components) of the transmission and each electric motor drive permits the estimation of the temperature dynamics of the system.

The transmission system model has been coupled with the vehicle longitudinal dynamics model explained in Chapter 2. The quality of the gearshift can only be evaluated through a model which at least considers the first order drivetrain torsion dynamics. In conditions of engaged gears, the overall drivetrain can be thought of as a system of one equivalent inertia (from the motors to the differential) connected to the wheel inertia (the second inertia of the overall system, an equivalent wheel inertia per axle can be considered) through the half-shafts, modeled as torsion springs and dampers. The plays in the drivetrain system are distributed between the dog clutches, the gear between the primary and secondary shaft, the gear between the secondary shaft and the differential case, the differential mechanism (planetary gears and sun gears) and the constant velocity joints. The plays in the different components have the same order of magnitude (a few decimals of a degree), but their significance is higher when they are located in close proximity to the wheels. The dynamic model implemented here considers an equivalent play of the transmission, located at the transmission output, between the inner constant velocity joints and the half-shaft. \( \Delta \theta_{\text{diff-w,eq}} \) is the equivalent angular torsion angle between the wheel and the differential, \( \Delta \dot{\theta}_{\text{diff-w,eq}} \) is the equivalent angular speed between the wheel and the differential, \( \Delta \theta_{\text{play}} \) is the equivalent angular backlash play in the transmission, at the output port, \( \theta_w \) is the rotational angle of the wheel and \( \theta_{\text{diff}} \) is the rotational angle of the differential.

In formulas:

\[
T_{hs,l/r} = k_{hs,l/r} \Delta \theta_{\text{diff-w,eq}} + \beta_{hs,l/r} \Delta \dot{\theta}_{\text{diff-w,eq}}
\]  

(130)
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where:

\[ \Delta \theta_{diff-w,eq} = \begin{cases} 0, & \text{if } |\theta_w - \theta_{diff}| < \Delta \theta_{play} \\ \left( |\theta_w - \theta_{diff}| - \Delta \theta_{play} \right) \text{sign}(\theta_w - \theta_{diff}), & \text{if } |\theta_w - \theta_{diff}| > \Delta \theta_{play} \end{cases} \]  

The dual-motor drivetrain can be coupled to either an open differential or a torque vectoring differential. Differential dynamics can be included or excluded depending on the purpose of the specific simulation run. Tyre longitudinal dynamics are modelled as explained in Chapter 3.

If either the ‘even’ or ‘odd’ primary shaft is not characterised by an engaged gear, the equivalent mass moment of inertia of the drivetrain is subject to a reduction, as it loses the contribution related to that side of the transmission. During the transition between the different transmission states, the mass moment of inertia of the wheel and the torsion dynamics (due to the stiffness and marginally the damping coefficient) of the half-shaft remain the same, however the variation of the equivalent inertia of the drivetrain provokes a variation of the dynamic response of the system. This variation is not so evident when considering a conventional manual transmission of an internal combustion engine driven vehicle, due to the fact that in the dual-motor drivetrain at least one electric motor drive (the motors are the major contributors to system inertia) usually remains engaged to the transmission output during a gearshift.

In the next paragraphs the drivetrain system will be simulated and tested on two case study vehicle applications, whose data sets are in Appendix E. The two vehicles are a rear wheel driven high performance sedan (case A) and a front wheel driven city car (case B) where each vehicle is equipped with very different motor drives. The first vehicle (case A) is equipped with an electric motor drive characterised by a limited angular speed range (maximum motor speed of 5,000 rpm) and a very wide constant torque region, whilst the second vehicle (case B) is equipped with an electric motor drive characterised by a high value of maximum speed (14,000 rpm) and a limited extension of the constant torque region. Table 6-2 reports the value of the equivalent mass moment of inertia (and the relating contribution to the vehicle apparent mass) of the dual motor drivetrain for each state, for the two case study vehicles.

Table 6-2: Mass moment of inertia (first row for each vehicle, expressed in kgm^2) of the rotating components of the electric drivetrain referred to the differential (and subsequent variation of vehicle apparent mass in kg)

<table>
<thead>
<tr>
<th>State</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case study vehicle A</td>
<td>7.5</td>
<td>4.8</td>
<td>2.7</td>
<td>2.7</td>
<td>10.1</td>
<td>5.3</td>
<td>3.2</td>
<td>8.0</td>
<td>2.2</td>
</tr>
<tr>
<td></td>
<td>70</td>
<td>45</td>
<td>25</td>
<td>25</td>
<td>95</td>
<td>49</td>
<td>30</td>
<td>75</td>
<td>20</td>
</tr>
<tr>
<td>Case study vehicle B</td>
<td>7.6</td>
<td>6.5</td>
<td>3.9</td>
<td>2.6</td>
<td>11.8</td>
<td>8.1</td>
<td>4.3</td>
<td>8.0</td>
<td>2.2</td>
</tr>
<tr>
<td></td>
<td>79</td>
<td>67</td>
<td>40</td>
<td>27</td>
<td>123</td>
<td>85</td>
<td>45</td>
<td>83</td>
<td>23</td>
</tr>
</tbody>
</table>
Note. In the second row for each case study vehicle, expressed in kg), as a function of the drivetrain state (transmission efficiency has been neglected in this calculation, but is considered in the formulas adopted in the simulator)

In low gear conditions for vehicle B, characterised by a low mass and a low torque electric motor drive, the contribution of the drivetrain rotating components to the vehicle apparent mass is particularly relevant.

A linearised and simplified (e.g. first order dynamics for the electric motor drive) model of the system has been implemented for each operating state, according to a state-space formulation (Nise, 2004). The main non-linearity to be considered is the longitudinal tyre response characteristic. Therefore the longitudinal slip stiffness is calculated for each operating state, considering the value of vertical load and slip ratio (a function of the expected wheel torque) for the specific linearisation point, through the Pacejka tyre model. A first linearisation of the tyre longitudinal force vs. slip ratio characteristic is carried out, followed by a second linearisation of the slip ratio as a function of wheel speed and vehicle equivalent angular speed. $T_{T,\text{theor}}$ is the theoretical tyre torque without considering any dynamics, $F_{x,\text{theor}}$ is the theoretical tyre force without considering any dynamics, $C_{\phi}$ is the tyre longitudinal slip stiffness, $\dot{\theta}_w$ is the angular velocity of the wheel, $\dot{\theta}_v$ is the equivalent angular velocity of the vehicle. A subscript, 0, denotes an initial condition.

In formulas:

$$T_{T,\text{theor}} = F_{x,\text{theor}}R_W$$

$$\equiv F_{x,0,\text{theor}}R_W + C_{\phi} \left( \frac{\partial_{\psi} - \dot{\phi}_w}{\partial_{\psi}} \right) R_W$$

$$\equiv F_{x,0,\text{theor}}R_W + C_{\phi}R_w\sigma_0$$

$$+ \frac{C_{\phi}R_w}{\partial_{\psi}} \left( \frac{\partial_{\psi} - \dot{\phi}_w}{\partial_{\psi}} \right)$$

(132)

The tyre relaxation parameter is considered constant in the linearised model. In actual operating conditions (excluding cornering), this parameter is a function of the slip ratio and tyre vertical load. The equations of the matrices derived for the state-space formulation are summarised in Appendix F.

Figure 6-3, Figure 6-4 and Figure 6-5 plot the adimensional frequency response characteristics of vehicle acceleration, where the adimensionalisation has been carried out through the steady-state value of the response. As the system is characterised by multiple inputs, the derivation of the vehicle acceleration frequency response is carried out through the combination of the resultant responses caused by the single inputs to the system. In particular, the reference ‘odd’ motor torque and the reference ‘even’ motor torque are combined through equation (133). The motor reference torques are considered to be synchronous. $A_{x}$ is the vehicle longitudinal acceleration, $n_{\text{input}}$ is the number of inputs to the state space system, $A_{x,i}$ is the vehicle longitudinal acceleration contribution due to the $i^{th}$ input to the state space system, $\psi_i$ is the phase angle of the vehicle acceleration due to the $i^{th}$ input to the state space system and $u_i$ is the $i^{th}$ input to the state space system.
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\[
|A_x| = \sqrt{\left( \sum_{i=1}^{n_{input}} |A_{x,i}| u_i \cos \varphi_i \right)^2 + \left( \sum_{i=1}^{n_{input}} |A_{x,i}| u_i \sin \varphi_i \right)^2}
\]  \hspace{1cm} (133)

Figure 6-3: Frequency response characteristic of the adimensional vehicle acceleration for states 1) - 9) of case study vehicle A, for the same operating condition (500 Nm of wheel torque at a vehicle speed of 10 m/sec)

The response of the system is strongly underdamped, with a variation of the first natural frequency (Figure 6-3) consistent with the values of the equivalent mass moment of inertia of the drivetrain shown in Table 6-2. The value of the damping ratio of the first mode of the system is an increasing function of the vehicle longitudinal velocity and wheel torque (which provokes a variation of the linearised longitudinal slip stiffness).

Figure 6-4: Frequency response characteristic of the adimensional vehicle acceleration for state 1 of case study vehicle A, for different values of vehicle longitudinal velocity and 1,000 Nm of wheel torque where the motor torque referred to the wheels is the same

The frequency response of the system is substantially independent from the torque distribution between the two electric motor drives, provided that their air gap torque dynamic characteristics are not significantly different. This property differentiates the dynamic characteristics of this novel drivetrain from those of a typical parallel hybrid
electric vehicle, where the torque distribution between engine and electric motor significantly affects the drivability response.

Figure 6-5: Frequency response characteristic of the adimensional vehicle acceleration for state 1) of case study vehicle A, for different values of wheel torque, at a vehicle speed of 10 m/sec where the motor torque referred to the wheels is the same for each motor

In state 9 of Figure 6-3 (disengaged gears on both drivetrain sides) the first natural frequency of the system is beyond the scale of the graph, at a higher level than the one usually considered for low frequency drivability analyses.

6.4 GEARSHIFT CONTROL

This section describes the seamless gearshift dynamics of the system, implemented on the non-linear model described in the previous section. In particular, several cases can be outlined: i) power-on upshift from a dual gear condition to another dual gear condition (i.e., from first and second gear to second and third gear); ii) power-off upshift in dual gear condition; iii) power-on downshift in dual gear condition; iv) power-off downshift in dual gear condition; v) the same cases as in i)-iv) in conditions of single-gear. The transitions from a dual gear condition to a single-gear condition and vice versa are a particular variant of case i). The next section will provide a detailed analysis of case i) and the equivalent of i) for a single-gear to single-gear state shift.

Initially, in the condition of engaged gear (or gears), an energy management system (EMS) calculates the torque demand of each electric powertrain depending on the current drivetrain state and driver torque demand. The energy management system achieves this through initially setting a theoretical reference wheel torque $T_{w,ref}(V,TD)$ as a function of vehicle velocity and driver torque demand, in percentage of the maximum available wheel torque. The maximum available wheel torque is computed from a look-up-table containing the envelope of the characteristics in Figure 6-2. For each reference wheel torque $T_{w,ref}$ and vehicle velocity, $V$, a look-up-table outputs the selected reference torque $T_{ref,odd,EMS}^*$ for the ‘odd’ electric motor drive, which can be tuned off-line according to an energy efficiency criterion. The reference torque $T_{ref,even,EMS}$ calculated by the EMS on the ‘even’ motor drive is:
The gearshift strategy controls the position of the gear actuators and, during gearshift actuation, supersedes the electric motor torque demands $T_{\text{ref,odd,EMS}}$ and $T_{\text{ref,even,EMS}}$ calculated by the energy management system.

### 6.4.1 UPSHIFT FROM DUAL GEAR STATE TO DUAL GEAR STATE IN POWER-ON

An upshift from a dual gear to another dual gear state in power-on is presented in Figure 6-6 (regarding a 1st-2nd to 2nd-3rd upshift), in which the manoeuvre is split into functional phases (from A to G). Firstly (phase A), the system carries out a torque roll-off phase (at a rate which depends on the tuning of the controller for the specific vehicle application) on the electric motor drive on the transmission side involved in the gearshift (the ‘odd’ motor drive in Figure 6-6). This is compensated by a torque increase on the other electric motor drive (the ‘even’ motor drive in the specific manoeuvre), with the aim of providing the desired vehicle acceleration profile during the upshift. In particular, the compensation takes into account the difference between the reference torque contributions at the wheel for the electric motor drive involved in the gearshift, $T_{\text{w,ref, mot gearshift,EMS}}$, and the actual estimated wheel torque $T_{\text{w,roll-off}}$ transmitted by the motor unit to the wheel. $T_{\text{ref,mot constant gear,roll-off comp}}$ is the reference torque to the motor side of the transmission in conditions of constant gear during the roll-off phase of the upshift and $\tau_{g,mot constant gear}$ is the gear ratio of the transmission in condition of constant gear during the upshift.

In formulas:

$$
T_{\text{ref,mot constant gear,roll-off comp}} = T_{\text{w,ref,mot gearshift,EMS}}(V,TD) - T_{\text{w,roll-off}} + T_{\text{w,ref,mot constant gear,EMS}} \tau_{g,mot constant gear,T_{FRR}}
$$

A detailed estimation of the actual wheel torque at the wheels can include the inertial parameters of the transmission (the same statement is valid for the torque contributions in equation (134); however for a basic implementation of the system, the inclusion of the gear ratio only in the estimation process is a sufficient approximation for an acceptable gearshift. Once the electric motor drive torque on the transmission side involved in the upshift has gone to zero, the reference signal is sent to the respective gear actuator (phase B of Figure 6-6). Once the dog clutch has been disengaged (phase C of Figure 6-6), a combination of a feedforward and a Proportional Integral Derivative (PID) controller is used for the speed control of the electric motor drive on the drivetrain side involved in the gearshift.
The dynamic torque balance equation of the electric motor drive can be linearised by modeling the windage loss (on the motor and transmission primary shaft) contribution as a viscous damping contribution with damping coefficient $b_{mot}$. The resulting loop-gain transfer function for the feedback control system is shown below. $T_{motor, theor}$ is the theoretical motor torque, $G_{PID}$ is the transfer function of the Proportional Integral Derivative controller of the electric motor drive speed and $b_{mot}$ is the equivalent damping.
coefficient of the electric motor drive and the transmission primary shaft in condition of both disengaged gears on the primary shaft.

\[
\frac{\dot{\theta}_{\text{mot}}}{T_{\text{motor, theor}}} = G_{\text{PID}}(s) = \frac{J_{\text{mot}}}{\omega_{n, \text{mot}}^2} s^3 + \left( \frac{2\zeta_{\text{mot}} J_{\text{mot}}}{\omega_{n, \text{mot}}} + \frac{b_{\text{mot}}}{\omega_{n, \text{mot}}^2} \right) s^2 + \left( J_{\text{mot}} + \frac{2\zeta_{\text{mot}} b_{\text{mot}}}{\omega_{n, \text{mot}}} \right) s + b_{\text{mot}}
\]  

(136)

The gains of the feedback controller can be tuned by using the conventional methodologies based on stability (gain margin and phase margin) and performance (tracking bandwidth). A sensitivity analysis of the system response to the variation of the proportional gain of the motor speed controller is shown in Figure 6-7. The structure of the motor speed controller is only marginally relevant, as the controller has an impact on the motor speed dynamics when the respective dog clutch is disengaged. As a consequence, the resulting dynamics do not directly affect vehicle response.

![Figure 6-7: Bode plots of the loop-gain transfer function and closed-loop transfer function for different tuning parameters (proportional gain) of the electric motor drive PID controller (case study vehicle A)](image)

When the difference between the actual motor speed and the reference motor speed in the new gear (gear 3 in Figure 6-6) is within a threshold (for example, 75 rpm), a counter is started (phase D of Figure 6-6). After the error between the reference motor speed and the actual motor speed remains within the threshold for a sufficient amount of time (for example, 100 msec), the dog clutch actuator is re-engaged on the next gear (phase E of Figure 6-6). Once the actuator has reached the reference position (new gear engaged), the reference motor torque on the drivetrain side involved in the gearshift is ramped up to the value specified by the energy management system, whilst equation (134) is used for the derivation of the motor drive torque on the other side of the drivetrain.
Figure 6-8 is the comparison of the acceleration profiles during 1\textsuperscript{st}-2\textsuperscript{nd} to 2\textsuperscript{nd}-3\textsuperscript{rd} upshifts for different driver torque demands, 30% and 50%, and different distributions of the torque between the two electric powertrains. In particular, a 50/50 distribution at the wheels (‘50% Distribution’ in the figure) is compared with a wheel torque distribution directly proportional to the respective gear ratio (‘GR Distribution’ in the figure). The latter permits the system to achieve the same motor torque demand for the two drivetrain halves before and after the gearshift manoeuvre, in the constant torque region of the two identical machines. The plot shows that the initial and final torque distributions do not affect the gearshift dynamics of the system, i.e. the system is robust against the motor torque distribution variations specified by the energy management system. Moreover, it is evident that for higher torque demands a torque gap is generated during the gearshift because of the saturation of the torque on the electric motor drive that compensates the torque roll-off phase of the powertrain half subject to the upshift.

6.4.2 UPSHIFT FROM SINGLE-GEAR STATE TO SINGLE-GEAR STATE IN POWER-ON

Figure 6-9 is an example of an upshift from a single-gear condition to another single-gear condition (from gear 1 to gear 2 in the specific case). The first step (phase A) in the procedure is the speed control (electric synchronisation) of the electric motor drive which is going to be characterised by the final gear ratio, through the same combination of feedforward and feedback control of the motor discussed for the previous manoeuvre. Once the error between the reference and the actual motor speed is within a threshold (phase B) for a specified amount of time, the dog clutch actuator can be moved (phase C) to engage the new gear, following which the reference torque level of the electric motor drive on the new gear side can be ramped up (phase D), whilst the reference torque of the electric motor drive on the other side of the transmission is ramped down, similarly to what is presented in equation (134).

Both upshift manoeuvres of Figure 6-6 and Figure 6-9 are characterised by a substantially seamless actuation, which is evident from the speed and acceleration profiles in the respective figures. Seamless upshifts in dual gear operating conditions can be achieved only when the system operates at a significantly lower torque demand than the maximum allowed level. Similar control methodologies have been applied to the control of the other possible combinations of upshifts and downshifts.
6.5 ENERGY EFFICIENCY AND VEHICLE PERFORMANCE EVALUATION

This section provides an insight into the methodologies used for the evaluation of the overall vehicle performance in conjunction with the adoption of the novel dual-motor drivetrain. The results are compared with those of other electric drivetrain configurations, such as the commonly adopted single-speed and two-speed electric drivetrains with central electric motor drive and differential. Firstly, the methodology implemented for the selection of the most efficient state (i.e. the equivalent of the gearshift map for a single-motor multiple-speed drivetrain) and torque distribution between the two electric machines for each driving condition is presented. Then simulation results and performance metrics are analysed and discussed.

6.5.1 STATE SELECTION

This paragraph explains the automated off-line procedures, partially summarised in Figure 6-10, which have been developed for the selection of the optimal (i.e. the most energy efficient) operating state and torque demand distribution between the two electric motor drives, for assigned values of wheel torque demand and vehicle velocity.

For a value of wheel torque, vehicle speed, drivetrain thermal condition (transmission and electric motor/s temperatures) and drivetrain state, the routine estimates the value of vehicle acceleration for the analysed road grade. Road grade is assumed equal to zero in the results presented in this article. If road grade can be estimated on-line during vehicle operation, the procedure should be repeated for the range of different road grades, otherwise the road grade can be neglected, as it only affects the estimated vehicle acceleration and drivetrain inertial contributions.

In case of states 1)-4) (single gear), a backward calculation is adopted for deriving the input power of the active electric machine, through the drivetrain components efficiency maps (transmission and motor drive), and by taking into account the relevant inertial contributions deriving from the acceleration of the rotating parts of the system. Finally, the input power $P_{\text{input}}$ to the drivetrain can be calculated including or excluding the energy storage unit efficiency properties. In this respect, the model described in (Gao, 2002) has been adopted. In case of states 5)-8) (two gears engaged), characterised by the cooperative action of two electric motor drives, it is necessary to impose the air gap torque of one of the two electric motor drives (e.g. $T_{\text{mot,odd}}$ in Figure 6-10) and calculate the required torque of the other motor (e.g. $T_{\text{mot,even}}$ in Figure 6-10). For each wheel torque, vehicle speed and transmission state (and thermal condition in a second approximation analysis), this calculation has to be repeated for the possible range of torque distributions between the two electric motor drives, in order to select the most efficient condition between those giving origin to the same net transmission output torque. In case of significant absolute values of wheel torque or vehicle speed, some of the states or torque distributions will not be able to generate those conditions (i.e. an assigned $T_{\text{mot,odd}}$ will provoke a $T_{\text{mot,even}}$ exceeding the motor limits) and therefore will not be considered as viable alternatives for that specific operating condition.
Figure 6-9: Example of upshift from 1st to 2nd in conditions of 30% driver torque demand for vehicle A

Once the lowest input power to the electric drivetrain has been computed for each possible state, the most efficient state for that transmission output can be selected. For example, Table 6-3 includes the comparison of the electric motors input power between the different
Novel 4-speed dual-motor transmission for electric vehicles: Energy efficiency and vehicle performance evaluation

states for a wheel torque of 600 Nm and vehicle speed of 70 km/h, for vehicle A. The difference in power demand for the possible states fully justifies the adoption of this multiple-speed drivetrain.

Table 6-3: Electric motor/s estimated input power comparison (in absolute value and percentage difference from the optimal state) for the different possible states of the dual-motor drivetrain, for a wheel torque of 600 Nm and vehicle speed of 70 km/h (vehicle A)

<table>
<thead>
<tr>
<th></th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Percentage diff. [%]</td>
<td>4.03</td>
<td>0.00</td>
<td>4.16</td>
<td>4.67</td>
<td>2.63</td>
<td>0.50</td>
<td>1.04</td>
<td>4.81</td>
</tr>
</tbody>
</table>

In Figure 6-10, \( T_w \) is the wheel torque, \( T_{\text{resist}} \) is the resistive wheel torque, \( T_{\text{min}} \) is the minimum electric motor torque at the current speed, \( T_{\text{max}} \) is the maximum electric motor torque at the current speed, \( T_{1,\text{even, output}} \) is the ‘even’ primary shaft torque and \( f_{\text{lag/o/1, odd/even}} \) is a Boolean variable equal to 0 or 1 depending on the state before and after the gearshift.

Figure 6-10: Simplified flow chart of the procedure adopted for the computation of the input power to the electric powertrains for states 5)-8), within the state and torque distribution optimisation procedure

The whole procedure is repeated for the possible range of wheel torques, vehicle speeds and thermal conditions for each state, according to a ‘brute force’ algorithm, as the
computational effort is still compatible with the capability of a personal computer. An optimisation run for the whole set of states, torques and speeds with reasonable parameter discretisation can be completed within 36 hours by a personal computer with 4 GB RAM and a dual-core 3 GHz processor.

The output of the routine is constituted by two multi-dimensional look-up-tables: i) the look-up-table providing the most efficient drivetrain state for each wheel torque demand and vehicle speed (and, optionally, road grade and thermal condition); ii) the look-up-table providing the most efficient torque distribution for the two electric machines, as a function of the same input parameters as the look-up-table in i). The look-up-tables can be ‘smoothed’ using an interpolation function to improve the driveability and a logic system can be adopted to reduce the number of spurious state changes. The look-up-tables in i) and ii) can be used both in case of a backward facing simulator, in which the time history of wheel torque during a driving cycle is assigned, and in case of a forward facing simulator or an actual vehicle implementation, as the driver request in the transmission controller is expressed in the form of a wheel torque demand. The state selection procedure implemented for this contribution does not consider the losses relating to tyre slip dynamics, however their inclusion is straightforward for more detailed studies.

Figure 6-11 illustrates a typical map of the optimal values of ‘even’ motor torques as functions of the vehicle operating conditions. The authors have tested the optimisation procedure on several vehicle data sets in addition to case A and case B, and have noticed that the optimal states selected by the procedure often imply the adoption of a single-motor state in traction and a dual-motor state in regeneration. This is due to the fact that in a single-motor state the vehicle is characterised by a lower value of apparent mass than in conditions of dual-motor operation, and therefore requires less input power to accelerate, but provides less regenerative power. The confirmation of this statement derives from the application of the procedure to case study drivetrains with symmetric efficiency maps of the electric motor drives. In fact, when the contribution of the inertial terms is neglected within the procedure, the ideal states and torque distributions are symmetrical in
regeneration and traction. The driving cycle simulation, even when adopting simplified backward facing models, needs to take into account the energy contribution relating to the gearshift dynamics, as the energy required for the second electric motor drive to be electrically synchronised with the transmission is supplied by the energy storage unit. This energy can be particularly relevant, for example in case of a gearshift between two single-gear states, as one of the motors will have to be accelerated from an initial standstill condition. This contribution is automatically taken into account in forward facing simulators, such as the one adopted for gearshift dynamics analysis in the previous section, whilst this aspect is usually neglected in backward facing simulators, which are the common simulation solution for gear ratio and state optimisation along driving cycles, because of their high computational efficiency. In a backward facing simulator, the energy balance $\Delta E_{\text{Gearshift}}$ during a gearshift can be estimated by:

$$\Delta E_{\text{Gearshift}} = k_{\text{equiv,odd}} \left[ f_{\text{lag}} g_{1,\text{odd}} \left( \frac{1}{2} I_{\text{mot,odd}} \dot{\theta}_v^2 \tau_{g,\text{odd,prev}}^2 \tau_{FRR}^2 \right) - f_{\text{lag}} g_{0,\text{odd}} \left( \frac{1}{2} I_{\text{mot,odd}} \dot{\theta}_v^2 \tau_{g,\text{odd,prev}}^2 \tau_{FRR}^2 \right) \right]$$

$$+ k_{\text{equiv,even}} \left[ f_{\text{lag}} g_{1,\text{even}} \left( \frac{1}{2} I_{\text{mot,even}} \dot{\theta}_v^2 \tau_{g,\text{even,prev}}^2 \tau_{FRR}^2 \right) - f_{\text{lag}} g_{0,\text{even}} \left( \frac{1}{2} I_{\text{mot,even}} \dot{\theta}_v^2 \tau_{g,\text{even,prev}}^2 \tau_{FRR}^2 \right) \right]$$

where $k_{\text{equiv,odd/even}}$ represents the equivalent efficiency (or the reverse of it) of the electric motor drive and (optionally) the energy storage unit during the manoeuvre. $\Delta E_{\text{Gearshift}}$ is added to the energy consumption estimation of the backward facing model along the driving schedule.

The conclusion is that the off-line methodology for the selection of the most energy efficient states described in this section provides optimal results when the vehicle is operated in a constant wheel torque and state condition, however the procedure gives origin to sub-optimal (but still indicative) results during a driving schedule. During a driving schedule, techniques such as dynamic programming and model predictive control can be used for the identification of the optimal sequence of states and a smooth transition (Beck, 2005).

### 6.6 RESULTS

This paragraph deals with the comparison of the dynamic performance and energy consumption characteristics provided by single-speed single-motor drivetrains, two-speed single-motor drivetrains and four-speed dual-motor drivetrains installed in the case study vehicles A and B. Each of the main vehicle drivetrain parameters were optimised (from the viewpoint of energy efficiency) through backward facing simulations. In particular, the single-speed and two-speed vehicle drivetrains have been optimised according to the procedures described in (Sorniotti, 2010) and (Sorniotti, 2011).
Novel 4-speed dual-motor transmission for electric vehicles: Results

The main results are reported in Table 6-4 and Table 6-5. The overall dynamic performance of the dual-motor four-speed system exceeds the dynamic performance of the other two more conventional drivetrain options. The single-speed drivetrain is incapable of providing acceptable dynamic performance, especially for vehicle A. Moreover, the energy consumption of the dual-motor drivetrain is consistently lower than the two-speed single-motor drivetrain, for a percentage between 3.2% (FTP75) and 4.8% (NEDC) for vehicle A, and a percentage between 3.8% (NEDC) and 0.5% (FTP75) for vehicle B. The energy consumption values reported in the table have been obtained through a backward facing simulator and include the gearshift actuation energy contribution $\Delta E_{Gearshift}$ for the dual-motor drivetrain, whilst they neglect the same contribution (which has little relevance, as it can be deduced in Sorniotti 2010) for the two-speed drivetrain.

Table 6-4: Performance comparison for vehicles A and B

<table>
<thead>
<tr>
<th>Case A – Unladen</th>
<th>Single-Speed</th>
<th>Two-Speed</th>
<th>Two vs Single-speed [% diff.]</th>
<th>Four Speed</th>
<th>Four vs Single-speed [% diff.]</th>
<th>Four vs Two-Speed [% diff.]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vmax [km/h]</td>
<td>170</td>
<td>216</td>
<td>26.67</td>
<td>268</td>
<td>57.57</td>
<td>24.39</td>
</tr>
<tr>
<td>0-10 km/h [s]</td>
<td>0.72</td>
<td>0.43</td>
<td>-40.28</td>
<td>0.42</td>
<td>-41.67</td>
<td>-2.33</td>
</tr>
<tr>
<td>0-30 km/h [s]</td>
<td>2.16</td>
<td>1.29</td>
<td>-40.28</td>
<td>1.28</td>
<td>-40.74</td>
<td>-0.78</td>
</tr>
<tr>
<td>0-60 km/h [s]</td>
<td>4.35</td>
<td>2.60</td>
<td>-40.23</td>
<td>2.57</td>
<td>-40.92</td>
<td>-1.15</td>
</tr>
<tr>
<td>0-100 km/h [s]</td>
<td>7.35</td>
<td>4.48</td>
<td>-39.05</td>
<td>4.81</td>
<td>-34.56</td>
<td>7.37</td>
</tr>
<tr>
<td>70-120 km/h [s]</td>
<td>3.82</td>
<td>3.45</td>
<td>-9.69</td>
<td>3.45</td>
<td>-9.69</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case B</th>
<th>Single-Speed</th>
<th>Two-Speed</th>
<th>Two vs Single-speed [% diff.]</th>
<th>Four Speed</th>
<th>Four vs Single-speed [% diff.]</th>
<th>Four vs Two-Speed [% diff.]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vmax [km/h]</td>
<td>165</td>
<td>165</td>
<td>0.01</td>
<td>166</td>
<td>0.69</td>
<td>0.68</td>
</tr>
<tr>
<td>0-10 km/h [s]</td>
<td>0.79</td>
<td>0.48</td>
<td>-39.24</td>
<td>0.45</td>
<td>-43.04</td>
<td>-6.25</td>
</tr>
<tr>
<td>0-30 km/h [s]</td>
<td>2.39</td>
<td>1.45</td>
<td>-39.33</td>
<td>1.37</td>
<td>-42.68</td>
<td>-5.52</td>
</tr>
<tr>
<td>0-60 km/h [s]</td>
<td>4.84</td>
<td>3.3</td>
<td>-31.82</td>
<td>3.12</td>
<td>-35.54</td>
<td>-5.45</td>
</tr>
<tr>
<td>0-100 km/h [s]</td>
<td>9.11</td>
<td>8.26</td>
<td>-9.33</td>
<td>7.47</td>
<td>-18.00</td>
<td>-9.56</td>
</tr>
<tr>
<td>70-120 km/h [s]</td>
<td>7.05</td>
<td>7.98</td>
<td>13.19</td>
<td>7.26</td>
<td>2.98</td>
<td>-9.02</td>
</tr>
</tbody>
</table>
The main results are reported in Table 6-4 and Table 6-5. The overall dynamic performance of the dual-motor four-speed system exceeds the dynamic performance of the other two more conventional drivetrain options. The single-speed drivetrain is incapable of providing acceptable dynamic performance, especially for vehicle A. Moreover, the energy consumption of the dual-motor drivetrain is consistently lower than the two-speed single-motor drivetrain, for a percentage between 3.2% (FTP75) and 4.8% (NEDC) for vehicle A, and a percentage between 3.8% (NEDC) and 0.5% (FTP75) for vehicle B. The energy consumption values reported in the table have been obtained through a backward facing simulator and include the gearshift actuation energy contribution $\Delta E_{\text{Gearshift}}$ for the dual-motor drivetrain, whilst they neglect the same contribution (which has little relevance, as it can be deduced in Sorniotti 2010) for the two-speed drivetrain.

### 6.7 CONCLUDING REMARKS

This chapter explained the research carried out during the project based around a novel 4-speed dual-electric motor transmission. The equations governing the operation of the transmission were first derived which allowed the modeling of the gearshift and performance tests and driving cycles to be carried out.

The key advantage of the transmission, which is that multiple operating states can be selected, provides flexibility for which electric motor operating point can be selected for each vehicle operating condition. A novel procedure was developed to select which state to be in for each vehicle speed and wheel torque, considering the efficiency maps and component inertias. The procedure resulted in a look-up table for the transmission state, similar to a gearshift map along with a map to define the optimal torque demand of one of the motors.

Two case study vehicles were considered with the novel multiple-speed transmission for various performance tests and driving cycles. The results were compared against single-speed and two-speed variants which had a gear ratio optimisation carried out. The results
show a large improvement over the single-speed case study for both vehicles, however the improvement over the two-speed case study was negligible. Therefore it would have to be investigated if the marginal gains would be worth the additional cost of a more complex transmission and electric motor drivetrain layout.
7 CONCLUSION AND FUTURE WORK

The research carried out during the project is summarised in this chapter along with suggested further research.

7.1 CONCLUSIONS

The report begins with a comprehensive review of the state-of-the-art research concerning electric vehicle power trains and specifically multiple-speed transmissions. The review initially showed that whilst the torque characteristic of an electric motor (where torque is available from zero speed) permits a single-speed transmission to be utilised, the adoption of a multiple-speed transmission has the potential to benefit an electric vehicles energy consumption and performance. However, the limited research in this field, predominantly carried out in Knödel (2009 and 2010) and Ren, Crolla and Morris (2009), utilised basic simulation techniques to make comparisons between different transmission architectures for electric vehicles. The conclusions were either drawn from graphical comparisons, Knödel (2009) or extremely basic vehicle models not considering any drivetrain optimisation to allow a fair comparison to be made, Knödel (2010) and Ren, Crolla and Morris (2009). Similar findings were published by authors in the field of automotive research proving this to be a topic of significant importance which is a conclusion further confirmed by the results of a questionnaire at the 8th International CTI Symposium in Berlin in 2009, Rhinderknecht and Meier (2011). It was this initial finding which drove the author to pursue this particular research topic as it was felt that through applying novel complex vehicle models with validated data improved results could be attained and enhanced conclusions could be made. The decision to focus the project on this topic required further review into multiple-speed transmissions and this was carried out by analyzing the state-of-the-art transmissions designed for electric vehicles. A further review was made of the gearshift methodologies adopted in the multiple-speed transmissions utilizing electric motors as a power source as the lack of an idle speed and the faster reaction time results in different methodologies being used. Furthermore, a review of techniques being used to build gearshift maps and gear schedules was carried out as it was necessary to utilise a method in simulation to allow fair comparisons to be made.

The industrial partners for the research were Vocis Drivelines and Oerlikon Graziano who had developed a prototype single-speed and a prototype two-speed transmission. The two transmissions were the focus of the majority of the research carried out, initially through comparison in simulation due to the lack of a mule car at the outset of the project. A complex vehicle model was developed in Matlab/Simulink which considered the resistance forces on the vehicle whilst modelling the vehicle suspension, tyre dynamics, half-shaft torsion and pitch dynamics. The vehicle powertrains were modelled for the electric motor, transmission and differential including the electric motor time constant and motor delay. The modelling of the single-speed transmission was carried out through deriving the governing equations and applying them to simulation, where the efficiency maps were
added as look-up tables. The two-speed transmission was modelled in a similar fashion however initially the gearshift dynamics were omitted from simulation.

The initial vehicle models were used to simulate performance tests and standard driving cycles with both the single-speed and two-speed transmissions. The vehicle case study adopted was based on Mercedes Vito taxi which was being developed at Vocis Drivelines as a test vehicle using a 70 kW PMSM and the two transmissions. The initial simulation results showed that there were significant performance gains to be found through adopting a multiple-speed transmission. The performance gains are predominantly due to a higher first gear being selected which increases the available wheel torque. A relatively heavy vehicle was considered so the increased wheel torque could be utilised without unwanted tyre slip. The vehicle models were used to simulate standard driving cycles to measure the energy consumption at the battery, as a comprehensive battery model was included in the vehicle model. The driving cycle results showed there to be a significant reduction in the energy consumption for the two-speed over the single-speed for the majority of driving cycles, however for the NEDC the energy consumption was slightly higher which may be due to the gear ratios not being properly optimised.

To correctly simulate the 2SED it required the understanding of the gearshift methodology and subsequently the modelling of the gearshift dynamics. To permit modelling of the gearshift, three states were considered and the governing equations for each state derived. In addition, a linear model was created to find the first natural frequencies in each state. A robust controller was developed to manage the motor speed during the inertia phase of the gearshift. Three different controller architectures were created based on different motor speed targets. The effect of the different controllers on the acceleration times and drivability was analysed and found to improve with a more aggressive engine speed target.

A key novel accomplishment of the project was the installation, commissioning and development of a hardware-in-the-loop test rig at the University of Surrey. The HiL test rig was designed to test electric vehicle drivetrains through having the entire system, from motor to transmission and half-shafts, installed on the test rig. The road load is simulated by two ‘hub motors’ which provide a resistive torque representing the resistance due to the vehicles inertia, rolling resistance and aerodynamic drag. The HiL was successfully shown to perform acceleration tests, drivability tests and driving cycles. In addition, the electric motor time constant was found along with the correct transmission efficiency map, which validated the model.

A final accomplishment of the project was the analysis of a novel powertrain for electric vehicles patented by the industrial partners. The transmission consists of four gears but two input shafts which attach to separate electric motors, allowing for eight different states to be utilised during traction and braking. The transmission was successfully modelled in simulation with particular attention being paid to understanding the gearshift dynamics between the different states. A novel state selection methodology was developed to allow the selection of the optimal state, from the view point of minimising energy consumption, during driving cycles. Two case study vehicles were considered and the 4-speed dual motor
transmission was seen to have an improvement in both performance and energy consumption over the single-speed along with marginal gains over the two-speed. Furthermore, as a gear ratio optimisation procedure was carried out for the two case study vehicles with the single-speed and two-speed transmissions the energy consumption was seen to improve for the NEDC and other drive cycles.

7.2 FUTURE WORK

The research carried out in this project has shown that a multiple-speed transmission can benefit the energy consumption and performance of an electric vehicle over a single-speed transmission. However, the research was limited to analysing the transmissions provided by the industrial partners, i.e., the single-speed, two-speed and novel four-speed dual-motor transmissions. It would be beneficial to analyse further transmission designs to understand the benefits in terms of energy consumption over driving cycles and vehicle performance as suggested below:

- Three-speed layshaft transmission;
- Multiple speed transmissions capable of seamless shifts;
- Transmission utilising planetary gear sets;
- Multiple-speed transmissions using electric synchronisation (dog clutch with no synchronisers);
- Power split hybrid transmissions.

The TTRP model can be utilised to develop power split control strategies which is currently a very relevant topic of research due to the influx of hybrid vehicles. The model can be adapted to vary the hybrid powertrain architectures and analyze the benefit of each in terms of energy consumption, vehicle performance and drivability. Furthermore, anti-jerk controllers could be developed for hybrid vehicles.

The controller for the gearshifts in both the dual-motor four-speed transmission and two-speed transmission utilised rule-based controllers with PID and feedforward elements. It would be interesting to investigate different robust controllers such as model predictive control, sliding mode control, etc, to understand if the drivability can be improved.

The optimisation strategies adopted in the research for gear ratio optimisation and the state/torque split of the two-motor/four-speed transmission utilised brute force algorithms. The algorithms could be further developed to use genetic algorithms or other non-linear optimisations, or advanced controllers to minimise the computational time.

The gearshift models of the two-speed transmission are advanced and could be adapted for DCT transmissions to optimise gearshift strategies and gearshift maps.

The HiL rig should be used to understand the effect of using different components on the efficiency of the transmission as this was not conducted within the research presented. In particular, the test rig could be used to understand the effect of bearing dimensions/types on transmission drag torque. The test rig can also be used to optimise and validate gearshift
Conclusion and future work: Future work

maps over standard drive cycles due to the fact the gearshift takes place physically and the non-linearities which take place during shifts are difficult to model.

The test rig could incorporate further sensors such as temperature sensors to develop and validate a clutch temperature model which is a useful tool to incorporate into gearshift controllers.

NVH and transmission noise are a relevant and current area of research, and as the transmission is very accessible on the test rig, microphones and accelerometers could be incorporated to focus on components or speed/torque ranges where NVH is a particular issue.


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APPENDIX

APPENDIX A

The DC motor is the default motor used in the majority of electronic applications including automotive applications. This is due to the torque characteristics being favourable to electric drivetrains whilst the electronic systems are simple due to a brushed mechanical commutator being used as the inverter. However, due to the commutator being used to direct the current between the rotor poles it is prone to wear and thus the reliability is compromised and repairs are not easily carried out. Consequently, brushless alternating current (AC) motors are generally preferred due to the increased reliability but for lower power systems the DC motor is still a viable option.

The first option of the three main AC motors is the induction motor (Zeraolia, Benbouzid and Diallo, 2006), which is characterised by having windings on both the rotor and stator. Current is applied to the stator windings intermittently by the power electronics providing a rotating magnetic field, current is then induced in the rotor windings similarly to a power transformer. The torque is created by the magnetic field in the rotor, which is created to oppose the current in the rotor windings, rotating the rotor until the magnitude of induced current and torque balances the load. If the rotor was to rotate synchronously with the stator magnetic field rotation no torque would be created so the rotor always rotates slower than the stator magnetic field and it is this slip which creates the torque. The benefits of the IM are higher reliability than the DC motor due to the lack of a mechanical commutator, low maintenance, low cost and robustness. A further advantage of an IM is the large region beyond base speed, which is comprised of a constant power region and a further extended region beyond a critical speed where a breakdown torque is reached. However, the IM does have some disadvantages such as low efficiency (particularly when compared to a Permanent Magnet Synchronous Motor (PMSM) due to there being copper losses in the rotor windings), low power factor and low inverter efficiency which is especially relevant when considering a high power high speed unit. A typical efficiency map for a squirrel cage induction motor is shown in Figure A-1. The IM has been adopted in several HEVs such as the Renault Kangoo, DaimlerChrysler Durango and BMW X5.

The permanent magnet synchronous motor (PMSM) differs from the IM in that permanent magnets are installed on the rotor in place of the windings. The permanent magnets can either be surface mounted or buried, and whereas the surface mounted configuration is more cost effective as it utilises less magnets the buried design creates a higher air-gap flux density. The rotating electromagnetic field is created through power electronics controlling the currents passing through windings in the stators which accelerates the rotor and creates torque. A distinct feature of the PMSM is a shorter constant torque region as the permanent magnets do not allow for any field weakening to take place. However, the constant power region can be extended through either modification of the conduction angle or through a field winding which can reduce the air-gap field generated by the permanent magnets at high speed.
Figure A-1: Analytical iso efficiency contour for a 4 kW, 400 V, 1500 rpm induction motor not including the inverter losses. (Stockman et al., 2010)

Figure A-2 illustrates two efficiency maps for a PMSM with 8 pole pairs and the left figure was constructed with concentrated windings and the right with distributed windings (Finken, Hombitzer and Hameyer, 2010). Each windings configuration has different characteristics, for example concentrated windings have a higher power density whereas the eddy-current losses are reduced for a distributed windings layout.

Figure A-2: Iso-efficiency maps for a PMSM, where the left figure is for a concentrated windings layout and the right is for a distributed windings layout, both motors have eight pole pairs (Finken, Hombitzer and Hameyer, 2010)

Stockman et al. (2010) show that there is an improvement in the efficiency of a PMSM over an induction motor as shown in Figure A-3. IE2 refers to the IEC 60034-30 standard. (Rotating electrical machines – Part 30: Efficiency classes of single-speed, three-phase, cage-induction motors (IE-code). IEC Standard 60034-30, (2008 – 10), 2008.)
The PMSM has been utilised in several Hybrid Electric Vehicle (HEV) drivetrains including the Honda Insight, Nissan Tino and the well-known Toyota Prius.

The final popular option for a brushless motor is the switched reluctance motor (SRM), which is unique in its design as it operates through reluctance torque. The SRM consists of a rotor which does not contain permanent magnets, or any windings, it is made from a soft magnetic material. The stator contains windings which are controlled through power electronics and when power is applied to each stator the rotor's magnetic reluctance pulls the rotor in line with the stator. The SRM can also have different numbers of stator and rotor poles, such as 4 rotor poles and 6 stator poles and it is more effective in generating a constant torque if the power to the current stator poles overlaps the power to the next stator poles. Figure A-4 illustrates a typical efficiency map for a SRM.
The SRM has advantages such as good reliability due to the simple design, but suffers such disadvantages as high acoustic noise and torque ripple. Australian car manufacturer Holden used the SRM in the ECOmmodore, (GM, 2015).

The efficiency map of the IM shown in Figure A-1 has the high efficiency region at approximately 50% of maximum motor torque and 50% of maximum motor speed with quite a narrow high efficiency area. The location of the high efficiency region of the PMSM varies significantly according to which winding architecture is adopted as shown in Figure A-2, with the high efficiency area being at 50% of maximum motor torque for each architecture but at low speed (25% of maximum motor speed) for concentrated windings but at the mid speed range (50% of maximum motor speed) for distributed windings. The efficiency map of the SRM shown in Figure A-4 has the high efficiency region at very low speed (20% of maximum motor speed) and 75% of maximum motor torque. The efficiency of the SRM reduces considerably as the speed increases so suggests that a high number of gears would be required to keep the motor in the high efficiency region during a driving cycle. The very different efficiency maps of each motor suggest that for any drivetrain the gear ratios/gearshift maps would have to be tailored specifically for each motor and a specific transmission would not be interchangeable between each motor.

Figure A-5, Knödel et al (2009), illustrates the different generic efficiency and power factors of the different motors. Note. The induction motor is referred to as the asynchronous motor in this figure.

![Efficiency and Power Factor](image.png)

Figure A-5: Comparison of electrical specifications of various electric motor types (Knodel et al, 2009)
<table>
<thead>
<tr>
<th>Variable</th>
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<th>Value</th>
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<td>Unladen vehicle mass (single-speed)</td>
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<td>[kg]</td>
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<tr>
<td>Front-to-rear payload distribution</td>
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<td>Battery capacity</td>
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<td>Number of battery modules</td>
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<td>[V]</td>
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<tr>
<td>Module resistance in parallel with the capacitor</td>
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<tr>
<td>Module capacitor capacitance</td>
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<tr>
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<td>Heat transfer coefficient motor/separation surface</td>
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</tr>
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</tr>
<tr>
<td>Second gear ratio (two-speed)</td>
<td>[-]</td>
<td>63/19</td>
</tr>
<tr>
<td>Primary shaft moment of inertia (two-speed)</td>
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<td>Friction clutch moment of inertia (two-speed)</td>
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<td>Secondary shaft moment of inertia (two-speed)</td>
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<tr>
<td>First gear outer moment of inertia (two-speed)</td>
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<td>Aluminium specific heat</td>
<td>[J/(kg·K)]</td>
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<td>Oil specific heat</td>
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<td>---------------------------------------------------------------------------</td>
<td>---------------</td>
<td>-----------</td>
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<tr>
<td>Torsional stiffness of the long half-shaft (connected to wheel hub)</td>
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<tr>
<td>Torsional stiffness of the short half-shaft</td>
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</tr>
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Note. Any parameters not mentioned can be taken from Appendix A.

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<tr>
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<td>Torsional stiffness of the right half-shaft</td>
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### APPENDIX D

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<th>FWD ICE Test Vehicle</th>
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<td>Vehicle Mass [kg]</td>
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<td>[%\text{front}/%\text{rear}]</td>
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<td>ICE - Peak Power [kW]</td>
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<td>69</td>
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<tr>
<td>ICE - Peak Torque [Nm]</td>
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<td>ICE Trans. – 1\text{st} Gear</td>
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<td>ICE Trans. – 2\text{nd} Gear</td>
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<td>ICE Trans. – 3\text{rd} Gear</td>
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<td>ICE Trans. – 5\text{th} Gear</td>
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<td>ICE Trans. – 6\text{th} Gear</td>
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<td>EM – Peak torque [Nm]</td>
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## APPENDIX E

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<th>Four-Speed</th>
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<th>Case B</th>
<th>Four-Speed</th>
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<tr>
<td><strong>Height of Centre of Gravity [m]</strong></td>
<td>0.28</td>
<td></td>
<td></td>
<td>0.32</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Aerodynamic Drag Coefficient [-]</strong></td>
<td>3.2</td>
<td></td>
<td></td>
<td>2.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Frontal Area of the Vehicle [m²]</strong></td>
<td>0.327</td>
<td></td>
<td></td>
<td>0.31</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Wheel Radius [m]</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX F

State vector

\[
X = \begin{bmatrix}
T_{\text{mot.del,odd}} \\
T_{\text{mot.del,even}} \\
\theta_{\text{diff}} \\
\dot{\theta}_{\text{diff}} \\
\theta_{w} \\
\dot{\theta}_{w} \\
\dot{v} \\
T_{T,\text{del}}
\end{bmatrix}
\]

Input vector

\[
U = \begin{bmatrix}
T_{\text{mot.theor,odd}} \\
T_{\text{mot.theor,even}} \\
\frac{R_{m}mg}{L} (C_{0,\text{front}} - C_{2,\text{front}} R_{w}^{2} \ddot{v}_{t,0}) \\
\frac{\rho SC_{d} R_{w}^{2} \dot{v}_{t,0}^{2}}{2F_{X,0} + 2C_{s} \sigma_{0}}
\end{bmatrix}
\]

State matrix

\[
A = \begin{bmatrix}
a_{1,1} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & a_{2,2} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & a_{3,4} & 0 & 0 & 0 & 0 \\
a_{4,1} & a_{4,2} & a_{4,3} & a_{4,4} & a_{4,5} & a_{4,6} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & a_{5,6} & 0 \\
0 & 0 & a_{6,3} & a_{6,4} & a_{6,5} & a_{6,6} & 0 & a_{6,8} \\
0 & 0 & 0 & 0 & 0 & 0 & a_{7,7} & a_{7,8} \\
0 & 0 & 0 & 0 & 0 & a_{8,6} & a_{8,7} & a_{8,8}
\end{bmatrix}
\]

where:

\[
a_{1,1} = -\frac{1}{\tau_{\text{mot,odd}}} \quad a_{2,2} = -\frac{1}{\tau_{\text{mot,even}}} \quad a_{3,4} = 1
\]

\[
a_{4,1} = \frac{\tau_{g,\text{sel,odd}} \eta_{g,\text{sel,odd}} F_{RR} \eta_{FRR}}{J_{\text{equiv.}}} \quad a_{4,2} = \frac{\tau_{g,\text{sel,even}} \eta_{g,\text{sel,even}} F_{RR} \eta_{FRR}}{J_{\text{equiv.}}}
\]

\[
a_{4,3} = \frac{K_{HS}}{J_{\text{equiv.}}} \quad a_{4,4} = \frac{c_{HS}}{J_{\text{equiv.}}} \quad a_{4,5} = \frac{K_{HS}}{J_{\text{equiv.}}} \quad a_{4,6} = \frac{c_{HS}}{J_{\text{equiv.}}}
\]

\[
a_{5,6} = 1 \quad a_{6,3} = \frac{K_{HS}}{2J_{w}} \quad a_{6,4} = \frac{c_{HS}}{2J_{w}} \quad a_{6,5} = \frac{K_{HS}}{2J_{w}}
\]
\[ a_{6,6} = - \frac{c_{HS}}{2J_w} \frac{R_w mg a}{L} \left( \frac{c_{1,\text{rear}} R_w + c_{2,\text{rear}} R_w^2 \dot{\phi}_{v,0}}{2J_w} \right) \]
\[ a_{6,8} = - \frac{1}{2J_w} \]
\[ a_{7,7} = - \frac{\rho \omega_c R_w^2 \dot{\phi}_{v,0}}{2J_w + m R_w^2} \frac{R_w mg a}{L} \left( \frac{c_{1,\text{front}} R_w + c_{2,\text{front}} R_w^2 \dot{\phi}_{v,0}}{2J_w + m R_w^2} \right) \]
\[ a_{7,8} = \frac{1}{2J_w + m R_w^2} \]
\[ a_{8,6} = \frac{2C_s R_w \dot{\phi}_{v,0}}{l_{\text{tyre}}/\dot{\phi}_{v,0}} \]
\[ a_{8,7} = - \frac{2C_s R_w}{l_{\text{tyre}}/\dot{\phi}_{v,0}} \]
\[ a_{8,8} = \frac{1}{l_{\text{tyre}}/\dot{\phi}_{v,0}} \]

**Input matrix**

\[ B = \begin{bmatrix} b_{1,1} & 0 & 0 & 0 & 0 \\ 0 & b_{2,2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & b_{6,3} & 0 & 0 \\ 0 & 0 & 0 & b_{7,4} & 0 \\ 0 & 0 & 0 & 0 & b_{8,5} \end{bmatrix} \]

where:

\[ b_{1,1} = \frac{1}{\tau_{\text{motor,odd}}} \]
\[ b_{2,2} = \frac{1}{\tau_{\text{motor,even}}} \]
\[ b_{6,3} = - \frac{1}{2J_w} \]
\[ b_{7,4} = \frac{1}{2J_w + m R_w^2} \]
\[ b_{8,5} = \frac{R_w}{l_{\text{tyre}}/\dot{\phi}_{v,0}} \]