DEVELOPMENT OF GEAR SHIFT PATTERNS FOR SIX-SPEED AUTOMATIC TRANSMISSION VEHICLE

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Abstract

The rising index of carbon dioxide emission from the road transport sector and growing level of stringent exhaust emission regulations have urged vehicle manufacturers and designers into developing solutions for more efficient vehicles, where well-designed vehicle transmission and gear shift pattern are some of the alternative approaches used. The present article proposes a design method in developing a suitable gear ratio set and gear-shift patterns for a six-speed automatic transmission. Proton Saga 2019 four-speed transmission model was built on the MATLAB/Simulink platform and the results were validated with experiment data. Then, a six-speed transmission model was developed based on the validated four-speed transmission model to assess the effectiveness of the new selected gear ratio set and gear-shift patterns. All models were simulated based on New European Driving Cycle to study on the fuel consumption and acceleration performance. Three different gear shift patterns were developed and analysed which are sport-mode, eco-mode, and combined-mode patterns. The obtained simulation results indicate that the proposed design method is feasible and effective, in which based on the eco-mode pattern, the six-speed transmission model has a lower fuel consumption rate compared to four-speed transmission model by 2.48%. Moreover, the 6-speed AT vehicle model sport-mode took 2 seconds faster compared to eco-mode which was 13 seconds to complete 0-100 km/h acceleration.

Keywords: Automatic transmission, Fuel consumption, Gear-shifting pattern, Simulation modelling, MATLAB, New European driving cycle.
1. Introduction

The rising index of the global CO$_2$ (carbon dioxide) emissions is always a great environmental concern to the people where the transportation sector has contributed about 24.6 % in the total global CO$_2$ emissions, and it is the second largest emitter compared to other sectors [1]. The road transport sector is known as the largest source of the man-made CO$_2$ emission from the combustion of petroleum-based products, like diesel and gasoline, in internal combustion engine (ICE) [2]. While the recent technology trend is moving towards hybridization and electrification, majority of road vehicles are still powered by ICE.

To minimize the amount of CO$_2$ being generated and released by ICE vehicles, it is important to keep the engine to operate at its most efficient region where a good driving dynamic with an optimum fuel consumption can be achieved. A well-designed vehicle transmission and gear shift pattern are ones of the alternatives to achieve this.

Vehicle transmission with greater number of gears has a greater overall transmission ratio and ratio spread between the gears, which provides better drivability, fuel economy, driving experience and shifting quality. This is due to more closely spaced gear ratios that contributed shorter shift times and lower rotating inertias [3]. Six-speed automatic transmission (6-speed AT) was introduced in 1999 and since then it overcomes four-speed automatic transmission (4-speed AT) in transmission market share. It was projected that 6-speed AT market share was 16% from overall transmission production in 2020 [2].

Technically, 4-speed AT and 6-speed AT shares a similar fundamental working principle which utilize planetary gear set mechanism along with few supplementary components such as shifting elements (clutches and brakes), torque converter, automatic transmission fluid and oil pump to achieve multiple gear ratios. In fact, the major difference between 4-speed AT and 6-speed AT is the design scheme or organization of the planetary gear set. The number of gear ratio available within an AT highly depends on the design configuration of the planetary gears’ arrangement. The popular design used for 4 AT and 6-speed AT were Ravigneaux gear set and Lepelletier gear set respectively [4].

The Ravigneaux gear set has a design configuration of nesting two planetary gear sets that can achieve with a more compact design, lower space requirement and lower manufacturing cost. This is due to the Ravigneaux gear set has two different sun gears and planetary gears that sharing a common ring gear and planetary carrier for operation. On the other hand, the Lepelletier gear set is basically an integration of the Ravigneaux gear set with a simple planetary gear set mounted at the front. An additional clutch is added within the Lepelletier gear set so that six forward gear ratios can be achieved. Thus, modern 6-speed AT vehicles are widely implementing the Lepelletier gear set mechanism as it consumes lesser space and mechanical components that leads to lower manufacturing and maintenance cost.

Gear shift process is controlled by transmission control unit (TCU) according to the real-time driving condition and demand needed from the driver by transmitting and receiving signals from other vehicle’s control units. Thus, gear shift pattern has significant influence on the vehicle fuel and acceleration performance [5].
In the modern 21st century, the development of the advanced automotive technologies is experiencing a fast growth rate where the computer simulation modelling technique has played a significant role. Simulation modelling technique is a computer-based tool that helps to solve real-world problems by using algorithms and equations. With correct and proper analysis method, one can easily obtain insights and reviews for a complex system by using the simulation modelling technique, thus enjoying benefits of design flexibility, time and resources saving, result visualization and optimization [6, 7].

There are various approaches implemented by different researchers in simulating and analysing the gear-shifting pattern based on computer simulation modelling technique. Casavola et al. [8] analysed and compared two gear shifting optimization strategies which were Efficient Gear Actuator (EGA) and Genetic and Fuzzy Algorithm (GFA) in terms of the fuel consumption performance based on NEDC. From the result obtained, it showed that the fuel consumption for an optimized power shift schedule was almost same with an optimized economical shifting schedule but took a shorter trip time compared to optimized economical shifting schedule.

On the other hand, Lu et al. [9] utilized Genetic Algorithm (GA) to optimize a traditional two-parameters shift schedule, by taking the road slope recognition as the optimization parameter. The simulation result showed that the optimized shift schedule was able to reduce the frequent gearshift and enhance drivability on ramp.

Ngo et al. [10] used Dynamic Programming (DP) technique to optimize the shift schedule of an automatic-manual transmission (AMT) vehicle. A fuel economy improvement of 15.4% was achieved by using the optimized shift schedule compared to the original shift schedule based on the NEDC test. Zhao et al. [11] also used the DP algorithm in optimizing the gear shifting strategy for off-road vehicle by considering the trade-offs between two factors which were trip time required and vehicle fuel consumption. Miao et al. [5], also utilized the DP technique and the Moving-Least-Square (MLS) method in their study to generate and optimize the gear-shifting schedule and shifting sequence.

While there are multiple researches conducted to optimize the existing gear shift pattern using different optimization tools, there are only few researchers that focuses on the method to develop gear ratio set and gear shift pattern for a vehicle transmission. Some of the researches are [4, 11-14]. The similarity between them is the development of the gear shift pattern are based on three-parameters which requires extra calibration and complicated procedures compare to two-parameters. Nevertheless, some of their techniques are referred and discussed in Section 2.

Thus, the objective of this study is to propose simple yet feasible two-parameters development methods for gear ratio set and gear shift pattern for 6-speed AT gasoline ICE vehicle. The vehicle powertrain model was developed on the MATLAB/Simulink platform based on model-based design technique. These proposed methods are able to provide easier and thorough understanding especially for those who have general knowledge on gear ratio set and gear shift development.

In this study, three modes of gear shift pattern were developed which are sport-mode for best acceleration performance, eco-mode for best fuel economy, and combined-mode which is the combination of sport and eco-mode. These patterns are based on two input parameters which are throttle opening position and vehicle speed using New European Driving Cycle (NEDC). The effect and significance of the gear
shift pattern on the vehicle fuel consumption and acceleration performance were analysed and studied by using simulation modelling technique.

### 2. Research Methodology

The simulation model was first developed based on Proton Saga 2019 4-speed AT. The simulation results were compared with measured data from chassis dynamometer test for validation purpose. Table 1 shows vehicle parameters and data of Proton Saga 2019. The gear ratios $i_g$ and final drive ratio $i_f$ are considered as confidential data.

<table>
<thead>
<tr>
<th>Vehicle Parameters</th>
<th>Specifications</th>
</tr>
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<tbody>
<tr>
<td>Vehicle Gross Mass, $m_v$ @ 4-speed AT</td>
<td>1485 kg</td>
</tr>
<tr>
<td>Vehicle Gross Mass, $m_v$ @ 6-speed AT</td>
<td>1496 kg</td>
</tr>
<tr>
<td>Engine Max Power, $P_{e_{\text{max}}}$</td>
<td>70 kW @ 5750 rpm</td>
</tr>
<tr>
<td>Engine Max Torque, $T_{e_{\text{max}}}$</td>
<td>120 Nm @ 4000 rpm</td>
</tr>
<tr>
<td>Frontal Area, $A$</td>
<td>2.11 m²</td>
</tr>
<tr>
<td>Tire Rolling Radius, $r_w$</td>
<td>0.261 m</td>
</tr>
<tr>
<td>Drag Coefficient, $C_D$</td>
<td>0.33</td>
</tr>
<tr>
<td>Overall powertrain efficiency, $\eta_T$</td>
<td>0.85</td>
</tr>
<tr>
<td>4-speed AT Gear Ratios, $i_g$</td>
<td>Confidential</td>
</tr>
<tr>
<td>Final Drive Ratio, $i_f$</td>
<td>Confidential</td>
</tr>
<tr>
<td>Gasoline Density, $\rho_g$</td>
<td>0.7475 kg/L</td>
</tr>
<tr>
<td>Tire Rolling Resistance Coefficient, $f_R$</td>
<td>0.02</td>
</tr>
<tr>
<td>Road Traction Coefficient, $\mu_T$</td>
<td>0.85</td>
</tr>
</tbody>
</table>

Figure 1 illustrates the simplified block diagram of 4-speed AT vehicle powertrain model using the model-based design method on MATLAB/Simulink platform. The actual schematic diagram of the model can be referred in Fig. A-1 (Appendix A). The 4-speed AT model consists of several subsystem models which are engine model, torque converter model, transmission model, vehicle body model, gear shifting actuator model, driver model and fuel economy model. The validated 4-speed AT vehicle powertrain model was used as the base model to construct the 6 AT vehicle powertrain model.

#### 2.1. Vehicle body dynamics and modelling

##### 2.1.1. Engine powertrain model

The engine block was driven by an engine power demand function $f(N_e)$, in which the engine provided maximum power available for a given engine speed. The function was normalized to determine the output power delivered by the engine block. The throttle input signal ranged between 0 – 1 was used to control the engine power generation.

\[
P_{e_{\text{max}}} = f(N_{e_{\text{max}}}) \tag{1}
\]

\[
P_e(N_e, T_l) = T_l \times P_{e_{\text{max}}} \tag{2}
\]

\[
\tau_e = \frac{P_e}{\omega_e} \tag{3}
\]
where $P_{\text{emax}}$ is the maximum engine power (W), $N_{\text{emax}}$ is the engine speed that delivers maximum engine power (rpm), $N_e$ is the engine speed (rpm), $\omega_e$ is the engine speed (rad/s), $T_i$ is the throttle input signal (0 – 1), $P_e$ is the engine power (W) and $\tau_e$ is the engine torque (Nm).

**2.1.2. Torque converter model**

The engine was connected to the transmission via torque converter which provided fluid coupling. The torque converter impeller coupled with the engine output while the turbine coupled with the transmission input. The impeller torque and speed had similar value with the engine output. The torque converter model was represented by a static nonlinear input-output model as [15]:

\[
\tau_I = \left(\frac{N_e}{K}\right)^2 \quad (4)
\]

\[
\tau_T = R_T \times \tau_e \quad (5)
\]

where $\tau_I$ is the impeller torque (Nm), $N_e$ is the engine speed (rpm), $K$ is the torque converter capacity factor (Nm/(rad/s)$^2$) and $R_T$ is the torque ratio. The torque ratio and capacity factor are the functions of the speed ratio of the torque converter which were modelled as:

\[
R_T = f_1(R_s) \quad (6)
\]

\[
K = f_2(R_s) \quad (7)
\]

\[
R_s = \frac{N_T}{N_e} \quad (8)
\]

where $N_T$ is the turbine speed (rpm) and $R_s$ is the speed ratio. The torque ratio $R_T$ and torque converter capacity factor $K$ are the functions of the speed ratio which were implemented from standard torque converter characteristics diagram.
2.1.3. Automatic transmission model

The conventional planetary AT has a multiple gear ratio to amplify the transmission input torque. The transmission model was represented as a basic input-output model as:

\[
\tau_{\text{Tout}} = \tau_{\text{Tin}} \times i_g
\]
\[
N_{\text{Tout}} = N_{\text{Tin}}/ i_g
\]

where \(\tau_{\text{Tout}}\) is the transmission output torque (N.m), \(i_g\) is the gear ratio, \(N_{\text{Tout}}\) is the transmission output speed (rpm), \(\tau_{\text{Tin}}\) and \(N_{\text{Tin}}\) are the transmission input torque (Nm) and input speed (rpm) respectively. The transmission input torque and speed were similar as turbine output torque and speed. The transmission output connected to the final drive differential for final amplification with the final gear ratio, \(i_f\). The purpose of the differential is used as the power split devices to deliver transmission output torque to both left and right wheels. The driving torque available at the driving wheel, \(\tau_D\) was determined based on a basic input-output model as:

\[
\tau_D = \tau_{\text{Tout}} \times i_f
\]

2.1.4. Vehicle body model

A basic one-dimensional translational dynamic was considered to study the longitudinal behaviour of the vehicle model. The driving torque delivered from the final drive differential was used to provide traction force to the driving wheels, to overcome the driving resistances acting on the vehicle, such as aerodynamic drag, gradient resistance and tire rolling resistance.

\[
F_D = \frac{1}{2} C_D \rho v^2 A
\]

where \(F_D\) is the aerodynamic drag (N), \(C_D\) is the drag coefficient, \(\rho\) is the air density at the sea level (kg/m³), \(v\) is the vehicle speed (m/s) and \(A\) is the frontal area (m²).

\[
F_R = m_v g f_R \cos \theta
\]

where \(F_R\) is the tire rolling resistance force (N), \(g\) is the gravitational acceleration (m/s²), \(f_R\) is the rolling resistance coefficient, \(m_v\) is the vehicle gross mass and \(\theta\) is the slope angle (º).

\[
F_G = \pm m_v g \sin \theta
\]

where \(F_G\) is the gradient resistance (N). The tractive force at the driving wheels is required to propel the vehicle and overcome the driving resistances.

\[
F_T = \frac{\tau_e i_g i_f}{r_w}
\]

where \(F_T\) is the tractive force (N), \(\tau_e\) is the engine torque (Nm), \(\eta_T\) is the overall powertrain efficiency and \(r_w\) is the tire rolling radius (m). Hence, the vehicle speed could be determined by solving the following differential equation in which the engine speed was calculated.

\[
m_v \frac{dv}{dt} = F_T - F_D - F_R - F_G
\]

\[
N_e = \frac{60 i_g i_f}{2 \pi r_w}
\]
2.1.5. Longitudinal driver model

The working principle of the longitudinal driver model was based on a Proportional-Integral-Derivative (PID) controller. The driver model utilized the reference speed based on NEDC and compared with the vehicle feedback speed. When the vehicle feedback speed was lower than the reference speed, a throttle acceleration signal was generated and when the vehicle feedback speed was higher than the reference speed, a brake signal was generated to apply brake torque to the vehicle.

2.1.6. Gear shifting actuator model

Stateflow feature was used to perform the gear shift. The Stateflow received inputs from the throttle signal and the vehicle feedback speed to determine the shift point based on the gear-shifting pattern defined in the gear shifting control algorithm, then generated an output desired gear number signal to the transmission model.

2.1.7. Engine powertrain model

The fuel economy model calculated the fuel consumption rate of the vehicle model based on the driving cycle simulated. The fuel consumption rate was calculated based on the brake specific fuel consumption (BSFC) contour map, with taking the engine speed and brake mean effective pressure (BMEP) as the inputs.

\[ BMEP = \frac{2\pi n_r \tau_e}{V_d} \times 10^{-5} \]  \hspace{1cm} (18)

where \( n_r \) is the number of crankshaft rotations for a complete engine cycle (2) for a four-stroke Otto cycle engine and \( V_d \) is the total engine displacement volume (m³). The total amount of fuel consumed by the vehicle model throughout the simulation cycle was calculated by using the following equation:

\[ L = \int_0^{N_d} \frac{BSFC \times P_e}{3600 \times \rho_g} \]  \hspace{1cm} (19)

where \( BSFC \) is the brake specific fuel consumption (g/kWh), \( \rho_g \) is the gasoline fuel density (kg/m³), \( L \) is the total amount of fuel consumed by the vehicle (l) and \( N_d \) is the drive cycle simulation time (s). Finally, the fuel consumption rate of the vehicle was calculated with the following expression:

\[ FC = \frac{L}{S} \times 100 \]  \hspace{1cm} (20)

where \( FC \) is the fuel consumption rate (L/100 km) and \( S \) is the total drive cycle distance (km).

2.2. Development of gear ratio set

The gear ratio set design method consisted of three steps which were first gear ratio selection, top gear ratio selection and the intermediate gear selection. The gear ratio set design method used in the present study was referring to the guidelines and suggestion given in [16-18].

2.2.1. First gear ratio selection

The first gear ratio, \( i_L \) is the largest value and was typically designed in a way to ensure the vehicle gradeability performance at the maximum engine output torque.
Based on the road standards design guideline provided, a maximum gradeability of 25% was used for determining the first gear ratio [19].

In a hill climb condition, the vehicle was assumed to be driven at a constant low speed, without acceleration. The aerodynamic drag was neglected. The traction force delivered must be able to overcome the maximum gradient percentage and the driving resistances. The first gear ratio was computed by referring to the Eqs. (13) to (15).

\[
i_L = \frac{\beta m v g (f_R \cos \theta + \sin \theta) r_w}{T_{\text{emax}} \eta_T} \tag{21}
\]

A reservation factor \( \beta \) (larger than 1) was included during the determination of the first gear ratio. The purpose was to provide extra safety measure in case there was greater vehicle load due to passengers, cargo, wind resistance or driving at a road with more than 25% gradeability.

The obtained first gear ratio was substituted in Eq. (15) to ensure that the traction force was not greater than the maximum allowable traction limit to avoid tire slippage. As Proton Saga 2019 is a FWD passenger car type, a 60:40 mass distribution was applied. The normal force, \( W_F \) acting on the front wheels was calculated by using the following equation:

\[
W_F = 0.6 (m v g) = \frac{l_r + h f_R}{(l_f + l_r) \mu_T h} W \cos (\theta) \tag{22}
\]

where \( l_r \) is the distance between center of gravity to rear axle (m); \( l_f \) is the distance between center of gravity to front axle (m) and \( h \) is the height of the center of gravity (m). The traction coefficient value, \( \mu_T \) varies according to the terrain condition in which the vehicle is driving on it. In the present study, the vehicle was assumed to be driving most of the time on the asphalt and tarmac road, with a traction coefficient value, \( \mu_T \) of 0.85 [20]. Thus, the traction force produced at the obtained first gear ratio could be checked by using the following equation.

\[
\frac{T_{\text{emax}} i_L \eta_T}{r_w} \leq \mu_T W_F \tag{23}
\]

2.2.2. Top gear ratio selection

The top gear ratio, \( i_H \) has the smallest value and is typically designed in a way to allow the vehicle to achieve the maximum speed on level ground based on the maximum engine power available. The vehicle was assumed to be cruising at a constant maximum speed without acceleration, resulted in an equilibrium condition. Hence, the maximum vehicle speed, \( v_{\text{max}} \) that can achieve by the vehicle with the engine powerplant was calculated.

\[
\eta_T P_{\text{emax}} = (F_D + F_R) v_{\text{max}} \tag{24}
\]

\[
(0.5 C_D \rho A) v_{\text{max}}^3 + (m v g f_R) v_{\text{max}} - \eta_T P_{\text{emax}} = 0 \tag{25}
\]

After determined the maximum vehicle speed, the top gear ratio was determined by rearranging Eq. (17).

\[
i_H = \frac{\pi N_{\text{emax}} r_w}{30 \eta_T v_{\text{max}}} \tag{26}
\]

The obtained top gear ratio was multiplied with an overdrive factor of 15%. The new overdrive ratio allows the engine to operate in a more fuel-efficient region at
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a lower engine speed and minimize the generated engine noises. When the vehicle is driven with an overdrive ratio, the transmission output rotates with a higher speed than the engine speed. A common practical design value for the overdrive factor has a range between 10–20% [21].

\[ i_H = 0.85 \times i_{H^*} \]  

(27)

2.2.3. Intermediate gear ratio selection

The first gear ratio and top gear ratio define the transmission ratio range, and a set of intermediate gears is required to bridge the gap. To shift from the first gear to top gear, a suitable intermediate gear set is required. Two methods were used to design the intermediate gear ratios which are geometric progression method and progressive design method.

The geometric progression method designs the intermediate gears in a way such that the engine speed is always maintained at a certain range that has an output torque value close to maximum. Typically, this region has a low BSFC and better acceleration performance. The specific speed range was defined by a low boundary and high boundary based on the engine output torque curve map.

The gear shift was assumed to be taken when the engine speed reached the upper boundary speed and dropped to lower boundary speed after shifting. At this point, the vehicle speed for both old and new gears was the same but with different engine speed. By repeating the same procedure for the rest of the gears, a geometric progression constant was obtained by using the following expression.

\[ \frac{N_H}{N_L} = \frac{i_1}{i_2} = \frac{i_2}{i_3} = \frac{i_3}{i_4} = \frac{i_4}{i_5} = C_{gp} \]  

(28)

where \( N_H \) is the upper boundary engine speed (rpm), \( N_L \) is the lower boundary engine speed (rpm) and \( C_{gp} \) is the geometric progression constant. Since the shifting condition is similar for every gear, the geometric progression constant has a constant value. In general, for an \( N \)-speed gearbox, the geometric progression constant and the intermediate gear ratios can be calculated based on the equations.

\[ C_{gp} = \left( \frac{i_1}{i_H} \right) \frac{1}{N-1} \]  

(29)

\[ i_i = i_{i+1} C_{gp} \]  

(30)

where \( N \) is the total number of gears available in a transmission. The progressive design method produces the intermediate gear ratios based on the geometric progression constant obtained from the geometric progression method. The intermediate gear ratios that are developed based on the geometric progression method has a larger speed difference, \( \Delta v \) for higher gears and smaller speed difference for lower gears. For passenger car type application, it is desirable to have a larger speed difference for lower gears and smaller speed difference for higher gears. In progressive design method, the geometric progression constant between each consecutive gear is different. A constant factor, \( k \) which having a typical value range from 0.8–1 was introduced to achieve with the progressive design. In the present study, the \( k \) was assumed as 0.95.

\[ i_i = \prod_{j=1}^{N-1} C_j i_H, \quad i = 1, 2, ..., N - 1 \]  

(31)
\[ C_1 = C_{gp} k^{1 - \frac{N}{2}}, \quad N > 2 \] (32)

\[ C_{i+1} = kC_i \] (33)

where \( C_i \) is the progressive constant between the gears and \( k \) is the progressive factor. By using the geometric progression method and progressive method, two intermediate gear ratio sets were obtained. By combining the first gear ratio and top gear ratio with both intermediate gear ratio sets, two different complete gear ratio sets for the 6-speed AT were obtained.

### 2.2.4. Finalization of gear ratio set

As the 6-speed AT model was developed based on Lepelletier gear set mechanism, the design limit and configuration of Lepelletier gear set must be adhered. The Lepelletier gear set contains a simple planetary gear set and a Ravigneaux gear set. Generally, the ring to sun gear ratio of the simple planetary gear and Ravigneaux gear set must be greater than 1. For the Ravigneaux gear set, it contains two sun gears (small sun and large sun) that share a common ring gear. Hence, the ring to small sun gear ratio must be greater than the ring to large sun gear ratio.

\[ g_1 = \frac{N_{PR}}{N_{PS}} = \frac{1}{i_{3-1}} \] (34)

\[ g_2 = \frac{N_{RR}}{N_{RLS}} = \frac{i_6}{1 - i_6} \] (35)

\[ g_3 = \frac{N_{RR}}{N_{RSS}} = \frac{i_1}{i_3} \] (36)

\[ \therefore g_1, g_2 & g_3 > 1 \; ; \; g_3 > g_2 \]

where \( g_1 \) is the gear ratio for simple planetary ring to sun gear, \( g_2 \) is the gear ratio for Ravigneaux ring to large sun gear, \( g_3 \) is the gear ratio for Ravigneaux ring to small sun gear, \( N_{PR} \) is the number of teeth in the planetary ring gear, \( N_{PS} \) is the number of teeth in planetary sun gear, \( N_{RR} \) is the number of teeth in Ravigneaux ring gear, \( N_{RLS} \) is the number of teeth in Ravigneaux large sun gear and \( N_{RSS} \) is the number of teeth in Ravigneaux small sun gear. After evaluated the feasibility of the developed gear ratio set, the corresponding driving condition and power-speed diagrams were established.

### 2.3. Development of gear-shifting pattern

In the present study, three basic two-parameters gear shift patterns were generated in terms of the best acceleration performance (sport-mode), best fuel economy (eco-mode) and combination of both (combined-mode), with referring to some of the design guidelines and references in [11-14, 17, 21]. The vehicle speed and throttle pedal opening signal were used as the input parameters.

#### 2.3.1. Best acceleration performance shift pattern (sport-mode)

For this shift strategy, an upshift was executed at the point whenever the maximum power of each gear was achieved. The shifting points were determined easily from the constructed power-speed curve diagram, where the upshift speed corresponding to specific throttle pedal opening signal (10\%, 20\%, 30\%, etc.)
were obtained. Thus, a performance upshift lines for each gear were obtained by connecting all the respective points.

If using the same approach, the performance downshift lines would be the same as the performance upshift lines. However, such downshift lines could not be utilized as it would cause frequent gear shifting or gear hunting where the vehicle drivability, fuel consumption and driving comfort would be greatly affected. Thus, a buffer zone between upshift and downshift was introduced to avoid the transmission from shifting back and forth along the lines [17]. In this study, linear convergence algorithm method was used to generate the downshift lines, by introducing a buffer constant, \( B \) that range between 0.3 – 0.55, to provide spacing between upshift line and downshift line [11, 22].

\[
B = \frac{v_{n+1} - v_n}{v_n} \\
v_{n+1} = (1 - B)v_n
\]

(37)

(38)

where \( v_n \) is the upshift speed from \( n \) gear to \( n+1 \) gear at the respective throttle pedal opening signal and \( v_{n+1} \) is the downshift speed from \( n+1 \) gear to \( n \) gear at the respective throttle pedal opening signal. Thus, the comprehensive downshift lines were obtained based on the calculated downshift speed. Hence, a complete sport-mode shift pattern from 10% throttle to 100% throttle was obtained by combining all the upshift and downshift lines into a single diagram.

2.3.2. Best fuel economy shift pattern (eco-mode)

For this shift strategy, an upshift was executed at the point when the BSFC value on a higher gear was less than the current gear. The shifting points were determined by marking the intersection point of the BSFC curve between the gears. The intersection points were obtained from the engine’s BSFC contour map plot provided where the upshift speeds corresponding to specific throttle pedal opening signal (10%, 20%, 30%, etc.) were obtained. Thus, an economy upshift lines for each gear could obtained by connecting all the respective points. The economy downshift lines were obtained by referring to the same linear convergence algorithm method used previously. Hence, a complete eco-mode shift pattern from 10% throttle to 100% throttle was obtained by combining all the upshift lines and downshift lines into a single chart.

2.3.3. Combination shift pattern (combined-mode)

For this shift strategy, it combined the gear shift patterns from sport and eco-mode. The combination method was typically designed in a way to reflect the common driving behaviour and style of an actual driver. This combined-mode shift pattern can provide optimum fuel economy during low-speed driving without losing the dynamic performance when acceleration is needed.

The combined-mode shift pattern divided the shift map into half. For a throttle pedal opening signal less than 50%, eco-mode was executed whereas for a throttle pedal opening signal more than 50%, sport-mode pattern was applied. For the throttle pedal opening signal between 50 – 100%, linearity connecting method was carried out. Thus, the upshift lines for each gear were obtained. The downshift lines were obtained by using the same linear convergence algorithm method. Hence, a complete combination method shift pattern from 10% throttle to 10% throttle was obtained.
Then, the gear ratio set and the gear shift patterns for 6-speed AT were developed. The 4-speed AT vehicle powertrain model shown in Fig. 1 was used to construct the 6-speed AT vehicle powertrain model by replacing the transmission model and gear shifting actuator models to 6-speed AT based. Figure 2 shows the simplified block diagram of 6-speed AT vehicle powertrain model.

Comparing to Fig. 1, all models were similar except for the transmission model and gear shifting actuator models. For the vehicle model, all vehicle parameters were kept constant except its gross mass due to the additional mass from the 6-speed AT itself. The models that were affected are shown in the dashed-line block. The detail schematic diagram of 6-speed AT can be referred in Fig. A-2 (Appendix A). The 6-speed AT transmission model used Lepelletier gear set mechanism and the gear shift patterns were installed within the gear shifting actuator model as the gear shifting control algorithm by using a two-dimensional lookup table method.

2.4. Vehicle data limitations

In the present study, the vehicle powertrain model was constructed to represent an actual Proton Saga 2019. Due to the confidential issue, plenty of sensitive vehicular data regarding the internal combustion engine, torque converter, gear shift pattern and BSFC contour map are unable to include in this document. The data which depicted in Table 1 were allowed to be included and most of the data can be found from public source such as vehicle brochure from Proton official website [23].

In addition, detail stability analysis was not conducted as it required access to the actual powertrain systems and signal monitoring of the frictional and hydraulic components. Thus, to ensure validity and reliability of the proposed development methods for gear ratio set and gear shift pattern for 6-speed AT, the obtained velocity speed profiles were compared with NEDC references speed profile as shown in Fig. 3, validation with 4-speed AT measured data was made as shown in Table 2, and gear-shifting stability was proved in Fig. 8.
3. Results and Discussions

3.1. Simulation validation

The 4-speed AT was simulated based on NEDC to evaluate the validity of the constructed model by comparing the obtained fuel consumption result with actual experimental result. Figure 3 demonstrates the comparison between the obtained velocity speed profile with NEDC reference speed profile. It can be clearly observed that the vehicle model in Fig. 1 drives accordingly to NEDC reference speed profile. The vehicle feedback speed indicates that there is no large fluctuation throughout NEDC cycle, which implies that the developed vehicle model is reliable.

![Figure 3. Comparison between the obtained velocity speed profile and NEDC reference speed profile](image)

The simulation result of the fuel consumption rate of the vehicle model based on NEDC was also obtained and compared with the actual experimental result as shown in Table 2.

<table>
<thead>
<tr>
<th>Result type</th>
<th>Fuel consumption rate</th>
<th>Error Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actual experimental result</td>
<td>6.8 L/100 km</td>
<td>-</td>
</tr>
<tr>
<td>Simulation result</td>
<td>6.934 L/100 km</td>
<td>1.97%</td>
</tr>
</tbody>
</table>

As indicated in Table 2, the 4-speed AT model has a fuel consumption of 6.934 L/100 km based on NEDC, which having an error difference about 1.97%. The error difference is mainly due to the excluded lock-up gear-shifting schedule in the 4-speed AT gearbox model as it would increase design complexity and utilized more computational resources. Besides, the constructed vehicle powertrain model also did not account for the external variables such as engine working temperature, power consumption of electrical auxiliary systems and driving style pattern of test driver. The 1.97% error percentage is considered acceptable hence the 4 AT vehicle model was validated. Then, the gear ratio sets for 6-speed AT were developed by using the mentioned methods.
3.2. Results and discussion for 6-speed AT

Table 3 indicates the results of generated gear ratio sets for 6-speed AT. Both design methods have similar first gear ratio and top gear ratio, but with different intermediate gear ratios. For the progressive method, it has a decreasing ratio spread from lower gear to higher gear. This indicates that a larger speed difference $\Delta v$ in the lower gear and smaller speed difference $\Delta v$ in higher gear, which result in better drivability, acceleration performance and fuel economy.

Table 3. Designed gear ratio sets for 6-speed AT based on two different methods.

<table>
<thead>
<tr>
<th>Gear number</th>
<th>Geometric progression method</th>
<th>Progressive method</th>
</tr>
</thead>
<tbody>
<tr>
<td>$i_1$</td>
<td>4.296</td>
<td>4.296</td>
</tr>
<tr>
<td>$i_2$</td>
<td>2.973 1.445</td>
<td>2.683 1.601</td>
</tr>
<tr>
<td>$i_3$</td>
<td>2.058 1.445</td>
<td>1.764 1.521</td>
</tr>
<tr>
<td>$i_4$</td>
<td>1.424 1.445</td>
<td>1.221 1.445</td>
</tr>
<tr>
<td>$i_5$</td>
<td>0.985 1.445</td>
<td>0.889 1.373</td>
</tr>
<tr>
<td>$i_6$</td>
<td>0.682 1.445</td>
<td>0.682 1.304</td>
</tr>
</tbody>
</table>

The geometric progression method has a constant ratio spread throughout all gears, which provide better driving comfort and smoother shift process. Both gear ratio sets were evaluated to determine whether the gear ratio set adheres to the design configuration and limit of the Lepelletier gear set mechanism. Table 4 displays the evaluation result for both gear ratio set design methods.

Based on Table 4, the gear ratio set generated based on the geometric progression method fails to achieve the design configuration and limit of the Lepelletier gear set mechanism, as the $g_1$ is smaller than 1 and $g_3$ is smaller than the $g_2$. Therefore, the gear ratio set developed by using the progressive method is selected as it has fulfilled the design requirement and limitation of the Lepelletier gear set mechanism.

Table 4. Designed gear ratio sets for 6-speed AT based on two different methods.

<table>
<thead>
<tr>
<th>Gear ratio set</th>
<th>$g_1$</th>
<th>$g_2$</th>
<th>$g_3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometric progression method</td>
<td>0.945</td>
<td>2.145</td>
<td>2.087</td>
</tr>
<tr>
<td>Progressive method</td>
<td>1.309</td>
<td>2.145</td>
<td>2.435</td>
</tr>
</tbody>
</table>

Then, the driving condition and power-speed diagrams of the selected gear ratio set for the 6-speed AT at maximum throttle were constructed, as shown in Fig. 4 and 5 respectively. The dashed-black lines in these figures represent road load resistance and road load power respectively at different road gradients.

As shown in Fig. 4, the developed gear ratio set based on progressive design shows good feasibility. The first gear ratio is able to provide sufficient traction force greater than the initial design gradeability assumption of 25%, with adequate reserve traction force in case extra vehicle load is applied. More importantly, the obtained traction curve for each gear was smaller than the maximum allowable traction limit which ensuring the vehicle can be driven or without slip.
The intersection between dash road resistance line at 0% grade with the sixth gear line indicates the maximum vehicle speed can be achieved, which is close to 160 km/h and it is acceptable for a compact urban car with 1.3L without turbocharger attributes where it can fulfil most of the travelling purpose need of vehicle owners. The launch slip arrow indicates that at that position, the engine cannot provide useful output torque below the idle engine speed, thus creating slippage.

![Diagram showing driving condition and power-speed performance](image)

**Fig. 4. Driving condition diagram of 6-speed AT with the selected gear ratio set at maximum throttle.**

The best acceleration performance upshift points were determined by referring to the power-speed diagram shown in Fig. 5. The red circles are the points where the respective gear reaches its maximum power $P_{max}$ indicated by the straight red line and these points were taken as the upshift points.

![Diagram showing power-speed performance](image)

**Fig. 5. Power-speed diagram of 6-speed AT with the selected gear ratio set at maximum throttle.**

The fuel consumption performance of each gear was evaluated by referring to the engine’s BSFC contour map. Figure 6 illustrates the fuel consumption performance for each gear at the maximum throttle condition. For the best fuel economy shift pattern, the upshift points are determined by the intersection points between BSFC curve lines as indicated by the black circles. The dashed-black lines represent vehicle upshifting and downshifting speed for each respective gear.
The downshift points were then determined after knowing the upshift points for both shift patterns by using linear convergence algorithm method. Then, the complete gear shift pattern in terms of best acceleration performance (sport-mode) and best fuel economy (eco-mode) were developed. The combination gear shift (combined-mode) was also constructed. Figures 7(a) to (c) illustrate the developed gear shift patterns.

The sport-mode shift pattern aims to provide the greatest acceleration performance by delaying the upshifting speeds to maximize the engine output power in accelerating the vehicle. As shown in Fig. 7(a), for the throttle pedal opening signal ranged from 0–40%, the upshift lines were made in a linear form to reflect the desired acceleration performance. Within this region, the more the driver applies the throttle pedal, the longer the vehicle continues running in the current gear which increases the engine speed to achieve better acceleration performance. For 40% throttle pedal opening signal and onwards, the upshift threshold speed for each gear was designed having a same constant value as the upshift threshold speed at maximum throttle pedal condition. As the possibility of a driver to actuate the throttle pedal to the maximum is very low in real life application, therefore, by designing the 40% throttle opening signal is able to provide quick and dynamic response to the vehicle to perform an upshift even though the driver does not apply the throttle pedal to the maximum. However, this does not affect the drivability or maximum acceleration performance of the vehicle as the engine speed is still required to reach the maximum power speed for making a gear shift at 40% throttle pedal opening signal and onwards.

The main objective of eco-mode shift pattern is to promote better fuel savings potential by lowering the sensitivity of the vehicle reaction towards the throttle pedal input. In Fig. 7(b), The upshift threshold lines have a constant upshift speed for throttle pedal signal ranged from 0 – 40% thus lowering the acceleration response. By doing so, earlier upshifts were made as to correspond to lighter throttle pedal openings which allows the vehicle operation status to reach higher gears more quickly, resulting in better fuel economy as the engine operation speed and upshift speed are relatively lower compared to the sport-mode. In case for the driver requires more traction power by applying greater throttle pedal opening more than 40% and onwards, the upshifts are delayed allowing driving wheels to obtain more traction force from the engine as the engine operation speed slowly builds up. However, the amount of traction force able to be generated to the driving wheels
for this mode is limited which the delivered output torque close to but lower than the peak torque value with an engine operation speed lower than 3500 rpm. Therefore, the driver will not obtain the desired feeling of vehicle acceleration capability similar to the sport-mode.

The combined-mode as shown in Fig. 7(c) allows the drive to experience the benefits from sport and eco-mode. This pattern provides optimum fuel consumption at low vehicle speed region without compromising on the vehicle response when acceleration is required. After obtaining all the required gear ratio set and gear-shifting patterns for the 6-speed AT, the effectiveness of the generated gear-shifting patterns were evaluated. The developed gear shift patterns were imported to the gear shifting control algorithm within the gear shifting actuator model by using the two-dimensional lookup table method. The gear ratio calculated from Table 4 was used as the input for the 6-speed AT transmission model. The obtained simulation results for all the gear-shifting patterns were recorded and tabulated in Table 5.

![Diagram](image)

(a) Best acceleration performance shift pattern.  
(b) Best fuel economy shift pattern.  
(c) Combination best performance and fuel economy shift pattern.

Fig. 7. Shift patterns at different modes.
Based on Table 5, all gear shift patterns show good result and workable, as the error percentage between the vehicle feedback speed with the NEDC reference speed profile is less than 1%. The 6-speed AT vehicle model is able to drive according to NEDC reference speed profile when using three different gear-shifting patterns.

Table 5. Overall simulation result for all gear-shifting patterns.

<table>
<thead>
<tr>
<th>Type of shifting pattern</th>
<th>Fuel consumption rate</th>
<th>0 – 100 km/h acceleration</th>
<th>Error in vehicle speed with reference speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Best acceleration performance</td>
<td>9.603 L/100 km</td>
<td>13 s</td>
<td>0.3917%</td>
</tr>
<tr>
<td>Best fuel economy</td>
<td>6.762 L/100 km</td>
<td>15 s</td>
<td>0.6638%</td>
</tr>
<tr>
<td>Combination</td>
<td>7.024 L/100 km</td>
<td>13 s</td>
<td>0.4878%</td>
</tr>
</tbody>
</table>

From the NEDC simulation test, eco-mode has the lowest fuel consumption rate of 6.762 L/100 km, which is about 2.48% lower than the 4-speed AT model. The 4-speed AT model was also using its original calibrated fuel economy eco-shift pattern during the chassis dynamometer test. This proof that the 6-speed AT simulation result with the developed eco-mode has better fuel economy than the 4-speed AT. The sport-mode has the highest fuel consumption rate while normal-mode or combination method lies in between which is about 38.5 % and 1.3 % higher than the 4-speed AT model respectively.

In terms of shifting activity analysis, all the developed gear-shifting patterns exhibit logical and acceptable result as shown in Fig. 8. Based on the Fig. 8, there is no any shifting busyness or frequent shift forth and back from all of the gear-shifting patterns, hence reducing the possibility of having shifting discomfort and extra fuel consumption due to frequent shifting action.

Based on the obtained simulation results, the extra two gears available from the 6-speed AT allow the same vehicle to run at a relatively lower engine speed which promote fuel economy especially driving at the urban area. The obtained 2.48% fuel economy improvement by using 6-speed AT exhibits a good agreement with the experimental results and findings acquired by the National Research Council and International Council of Clean Transportation which reported that a vehicle equipped with 6-speed AT able to reduce the fuel consumption rate in a range of 1.7 – 2.3% relative to the same vehicle that equipped with 4-speed AT [24] while the gear shift pattern has a significant impact on the fuel economy and drivability.

In terms of the cost-based analysis, it is a fact that transmission with greater number of gears has a higher manufacturing cost and heavier weight due to extra mechanical components. According to Corporate Average Fuel Economy (CAFE) standard [24], the direct manufacturing cost for a 6-speed AT gearbox is about 12.87 % relatively higher than the 4-speed AT gearbox. Despite of having higher cost, the 6-speed AT gearbox can bring long-term fuel savings benefit which in turn provides advantage compared to 4-speed AT vehicle owner. In addition, as the proposed method only uses two parameters instead of currently used three parameters, the lead time for development process is reduces as the technical complexity reduces. As the actual test will be conducted using the same facility and same cycle test, no significant cost difference will be incurred.
Development of Gear Shift Patterns for Six-Speed Automatic.

(a) Sport-mode gear-shifting pattern. (b) Eco-mode gear-shifting pattern. (c) Normal-mode gear-shifting pattern.

Fig. 8. Gear shifting analysis.

Due to the unavailable data for the wide-open-throttle test for the 4-speed AT, the present study only studies and compares the dynamic acceleration performance of the 6-speed AT with the designated gear ratio set and gear-shifting patterns. A wide-open throttle analysis is included on the 6-speed AT model to evaluate the dynamic performance of the gear shift patterns. The eco-mode pattern takes longest time to accelerate from 0–100 km/h, which is about 15 s with 2 s slower than the sport-mode and combined-mode. Due to the combined mode has the same shifting pattern design after 50% throttle pedal opening signal section, therefore it has a same 0–100 km/h acceleration result as the sport-mode pattern, which is about 13 s. However, the normal driving gear-shifting pattern in an actual vehicle will not have a same 0–100 km/h acceleration performance compared to sport-mode pattern. As the gear-shifting pattern used in actual vehicle also affects the dynamic performance and response of other modules such as air-intake, fuel-injection, suspension, steering, etc.

3.3. Main achievements

The obtained results proved that the proposed design method is viable yet simple compares to other methods as mentioned in section 1. The advantages of this method are able to reduce development complexity and calibration process. This
two-parameters progressive method is capable to determine the gear ratio sets for 6-speed AT and designing its gear shift patterns, manages to fulfil the design requirement and limitation of the Lepelletier gear set mechanism and able to improve fuel economy by 2.48% relative to the same vehicle that equipped with 4-speed AT which consistent with the experimental results and findings acquired by the National Research Council and International Council of Clean Transportation [24]. The proposed design method is also accurate when applied in 4-speed AT vehicle model with 1.97% accuracy compares to experimental data. The design method performance comparison between 6-speed AT and 4-speed AT is not available due to undisclosed performance data of 4-speed AT.

4. Conclusions

In conclusion, the present article proposes a design method in determining the gear ratio set and gear shift pattern for a six-speed Lepelletier automatic transmission. The simulation results indicate that a similar vehicle that equipped with a 6-speed AT has about 2.48 % better fuel economy than the 4-speed AT under the eco-mode.

In addition, the simulation results also show that the gear-shifting pattern has a significant impact on the vehicle’s acceleration performance, fuel economy and shifting experience, in which eco-mode has the lowest fuel consumption rate, followed by normal-mode and then sport-mode, which has a value of 6.762 L/100 km, 7.024 L/100 km and 9.603 L/100 km respectively. Under the wide-open-throttle simulation test, the sport-mode and normal-mode took about 13 s to accelerate from 0 – 100 km/h while eco-mode took 15 s to complete.

Thus, it is important to design a gear-shifting pattern that can adapt to the driving behaviour or driving style of the actual driver to fulfil the driving demand needed which allow the vehicle to be able to deliver optimum performance in wide variety of driving situations. More importantly, the usage of computer simulation technology in studying the vehicle performance has been proven can be worked effectively and efficiently, with a pre-requirement of having a correct, accurate and validated model.

For the future work, the designated gear ratio set along with the developed gear-shifting logics can be continued with the optimization work to obtain a better result based on the parameters needed. After that, the optimized six-speed Lepelletier gear set can be tested on an actual vehicle, Proton Saga 2019 by using chassis dynamometer to effectively evaluate the real-time performance of the design work for verification purpose.

Nomenclatures

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Frontal area, m²</td>
</tr>
<tr>
<td>B</td>
<td>Buffer constant</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>C_D</td>
<td>Drag coefficient</td>
</tr>
<tr>
<td>C_GP</td>
<td>Geometric progression constant</td>
</tr>
<tr>
<td>C_i, C_j</td>
<td>Progressive constant between the gears at i or j state, where i or j = 1, 2, 3 …</td>
</tr>
<tr>
<td>f_R</td>
<td>Tire rolling resistance coefficient</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$f_i$</td>
<td>Standard torque converter characteristics function</td>
</tr>
<tr>
<td>$F_C$</td>
<td>Fuel consumption rate, L/100 km</td>
</tr>
<tr>
<td>$F_D$</td>
<td>Aerodynamic drag, N</td>
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<tr>
<td>$F_G$</td>
<td>Gradient resistance force, N</td>
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<tr>
<td>$F_R$</td>
<td>Tire rolling resistance force, N</td>
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<tr>
<td>$F_T$</td>
<td>Tractive force, N</td>
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<td>$g$</td>
<td>Gravitational acceleration, m/s²</td>
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<td>Gear ratio for simple planetary ring to sun gear</td>
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<tr>
<td>$g_2$</td>
<td>Gear ratio for Ravigneaux ring to large sun gear</td>
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<td>Gear ratio for Ravigneaux ring to small sun gear</td>
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<tr>
<td>$h$</td>
<td>Height of the center of gravity, m</td>
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<td>Final drive ratio</td>
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<td>$i_g$</td>
<td>Gear ratio</td>
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<td>$i_i$</td>
<td>Gear ratio at $i$ state, where $i = 1, 2, 3…$</td>
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<td>$i_H$</td>
<td>Top gear ratio</td>
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<td>$i_L$</td>
<td>First gear ratio</td>
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<tr>
<td>$k$</td>
<td>Progressive factor</td>
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<td>$K$</td>
<td>Torque converter capacity factor, N.m/(rad/s)²</td>
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<tr>
<td>$l_f$</td>
<td>Distance between center of gravity to front axle, m</td>
</tr>
<tr>
<td>$l_r$</td>
<td>Distance between center of gravity to rear axle, m</td>
</tr>
<tr>
<td>$L$</td>
<td>Total amount of fuel consumed, litre</td>
</tr>
<tr>
<td>$m_v$</td>
<td>Vehicle gross mass, kg</td>
</tr>
<tr>
<td>$n_r$</td>
<td>Number of crankshaft rotations for a complete engine cycle</td>
</tr>
<tr>
<td>$N$</td>
<td>Total number of gears available in transmission</td>
</tr>
<tr>
<td>$N_d$</td>
<td>Drive cycle simulation time, s</td>
</tr>
<tr>
<td>$N_e$</td>
<td>Engine speed, rpm</td>
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<tr>
<td>$N_{emax}$</td>
<td>Engine speed that delivers maximum engine power, rpm</td>
</tr>
<tr>
<td>$N_H$</td>
<td>Upper boundary engine speed, rpm</td>
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<tr>
<td>$N_L$</td>
<td>Lower boundary engine speed, rpm</td>
</tr>
<tr>
<td>$N_{PR}$</td>
<td>Number of teeth in the planetary ring gear</td>
</tr>
<tr>
<td>$N_{PS}$</td>
<td>Number of teeth in planetary sun gear</td>
</tr>
<tr>
<td>$N_{RR}$</td>
<td>Number of teeth in Ravigneaux ring gear</td>
</tr>
<tr>
<td>$N_{RLS}$</td>
<td>Number of teeth in Ravigneaux large sun gear</td>
</tr>
<tr>
<td>$N_{RSS}$</td>
<td>Number of teeth in Ravigneaux small sun gear</td>
</tr>
<tr>
<td>$N_T$</td>
<td>Turbine speed, rpm</td>
</tr>
<tr>
<td>$N_{Tin}$</td>
<td>Transmission input speed, rpm</td>
</tr>
<tr>
<td>$N_{Tout}$</td>
<td>Transmission output speed, rpm</td>
</tr>
<tr>
<td>$P_e$</td>
<td>Engine power, W</td>
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<tr>
<td>$P_{emax}$</td>
<td>Maximum engine power, kW</td>
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<td>$r_w$</td>
<td>Tire rolling radius, m</td>
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<td>$R_s$</td>
<td>Speed ratio</td>
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<td>$R_t$</td>
<td>Torque ratio</td>
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<tr>
<td>$S$</td>
<td>Total drive cycle distance, km</td>
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<tr>
<td>$T_{emax}$</td>
<td>Maximum engine torque, N.m</td>
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<tr>
<td>$T_i$</td>
<td>Throttle input signal</td>
</tr>
<tr>
<td>$v$</td>
<td>Vehicle speed, m/s</td>
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<tr>
<td>$v_{max}$</td>
<td>Maximum vehicle speed, m/s</td>
</tr>
<tr>
<td>$v_n ↑$</td>
<td>Upshift speed from $n$ gear to $n+1$ gear</td>
</tr>
<tr>
<td>$v_{n+1 ↓}$</td>
<td>Downshift speed from $n$ gear to $n+1$ gear</td>
</tr>
</tbody>
</table>
\( V_d \) Total engine displacement volume, \( m^3 \)
\( W_F \) Normal force, \( N \)

**Greek Symbols**
- \( \beta \): Reservation factor
- \( \eta_T \): Overall powertrain coefficient
- \( \rho \): Air density at sea level, \( kg/m^3 \)
- \( \rho_g \): Gasoline density, \( kg/L \)
- \( \tau_e \): Engine torque, \( Nm \)
- \( \tau_D \): Driving torque, \( Nm \)
- \( \tau_I \): Impeller torque, \( Nm \)
- \( \tau_T \): Turbine torque, \( Nm \)
- \( \tau_{Tin} \): Transmission input torque, \( Nm \)
- \( \tau_{Tout} \): Transmission output torque, \( Nm \)
- \( \mu_T \): Road traction coefficient
- \( \omega_e \): Engine speed, \( rad/s \)
- \( \theta \): Slope angle, \( deg \)

**Abbreviations**
- 4-speed AT: Four-speed automatic transmission
- 6-speed AT: Six-speed automatic transmission
- AT: Automatic Transmission
- AMT: Automatic-Manual Transmission
- BMEP: Brake mean effective pressure
- BSFC: Brake specific fuel consumption
- CAA: Clean Air Act
- CAFE: Corporate Average Fuel Economy
- DP: Dynamic Programming
- EGA: Efficient Gear Actuator
- EPA: Environmental Protection Agency
- FWD: Front wheel drive
- GA: Genetic Algorithm
- GFA: Genetic and Fuzzy Algorithm
- ICE: Internal Combustion Engine
- MLS: Moving-Least-Square
- NEDC: New European Driving Cycle
- PGT: Planetary gear train
- PID: Proportional-Integral-Derivative
- TCU: Transmission control unit

**References**

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Appendix A

Detail MATLAB/Simulink Schematic Diagram

Fig. A-1. 4-speed AT vehicle powertrain model.

Fig. A-2. 6-speed AT vehicle powertrain model.