MODELING AND SIMULATIVE ANALYSIS OF SHIFT CONTROL FOR AT POWERTRAIN ON THE CAR

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Abstract - The problem with this power transmission system is related to increased fuel consumption, and small performance. The above issues are decisive for the management of the AT transmission system. In other words, determining when to shift gears is one of the factors that determine the advantages of this system. In this paper, the authors focus on building a model consisting of the engine, hydraulic torque converter, gearbox, final drive and differential, drive wheels, and vehicle body dynamic system and propose a gear shift control algorithm for AT based on the throttle opening level (%) and the longitudinal velocity of the car. Finally, the effectiveness and advantage of the proposed control schemes are indicated by illustrative results through MATLAB/SIMULINK. This approach is the basis for research on improving performance and optimizing fuel for automotive powertrains.

Key words - Automatic Transmission; powertrain; vehicle dynamic

1. Introduction

One of the important systems in the car is the power transmission system with functions such as the transmission, torque conversion, and rotational speed from the engine to the active wheels with each vehicle's operating mode as well as the resistance generated on the car during movement process [1], [2]. Depending on the engine's power transmission method, the powertrain systems can be divided into: Mechanical Transmission System (MT) [3], [4], Automatic Transmission System (AT) [5], [6]; The power transmission system uses Dual Clutch Transmission [7], [8]; Continuously Variable Transmission [9].

Thanks to its relatively large transmission efficiency, gently control operations, and performance of the correct gear shifting, the AT automatic transmission system is commonly used on most modern cars, making gear selection determined based on the gearshift pattern of the vehicle speed and engine throttle opening. Currently, there is constant and significant growth and improvement in the good scope for vehicles with automatic gearboxes when it comes to gearshift. Controlling and handling a gearbox switching components, such as freewheels and the clutches, has a significant impact on gearshift comfort. In which automatic transmissions using planetary gear pairs are still the majority. Currently, automatic transmission continues to be researched completely in the direction of increasing the number of levels and improving control quality with new algorithms. However, domestic automatic transmission is still limited, especially control algorithms and gear shift controllers. An electronic control unit is implemented to perform this control, which operates according to a complex control algorithm based on many different input parameters. However, the manufacturer doesn't publish details of the control algorithm, so researching and accessing control technology on automatic transmissions is difficult. Within the limits of research, the article proposes a gear-shifting control algorithm on a fixed-gear automatic transmission (using a planetary gear pair) based on the accelerator pedal level signal (throttle opening degree) and car speed. However, this power transmission is related to fuel consumption when using hydraulic torque converters, heat release from hydraulic oil flow, and friction increase engine temperature and reduce thermodynamic efficiency leading to increased fuel consumption rate. Therefore, building a dynamic model and simulating an automatic transmission system to see the system's ability to respond when shifting gears and starting. At the same time, it is the basis for designing an automatic control system of the torque converter for this powertrain system to help overcome the above disadvantages. When the vehicle runs stably, the torque converter will be locked to increase performance and improve performance system operability. Building an automatic power transmission system model through a mass moment of inertia can be found in [6], [10]-[15]. Research content uses a gear shift controller model in an automatic transmission built using Matlab – Stateshow simulation software based on the proposed control algorithm. Then combine and inherit with simulation models such as a heat engine, gearbox models, torque converters, etc. from Matlab to get a complete system.

In this study, the authors build a dynamic model and simulate the process of changing speed based on the gear shifting control algorithm of the AT automatic transmission system, the dynamic model takes into account the torsional elasticity of the axles, the torque converter system can react slowly to sudden speed changes during the starting and shifting phases.

2. AT powertrain's dynamic model

A normal powertrain's design as in Figure 1 includes, in general, the following components: engine, clutch, transmission, final drive, driveshaft, and tires. The AT powertrain system still consists of these parts. But instead of a clutch, a torque converter is used to engage and disengage the power from the engine to drive parts.

The fundamental assumptions used to build a dynamic model of the AT powertrain system are: neglecting the motion of the engine on its suspension, assuming the distribution of torque to the drive wheels on each side is similar, considering the rotating components are absolutely stiff and have determined torque, and supposing each part of the system is a lumped mass-spring-damper model. Based on the mentioned reasoning, the whole general dynamic model was easily built and described in Figure 2.

Where: T_e, T_b, T_t, T_r are correspondingly engine torque, pump torque, turbine torque and load torque; J_1 is the equivalent mass moment of inertia of the engine and pump; J_2 is the equivalent mass moment of inertia of the turbine, shaft and primary transmission shaft; J_3 is the equivalent mass moment of inertia of the secondary transmission shaft; J_{41}, J_{42} are respectively the equivalent mass moment of inertia of the final drive and differential unit in the two branches; J_{51}, J_{52} are respectively the equivalent mass moment of inertia of the half-shaft and wheels in the two branches; J_6 is the mass moment of inertia of the vehicle mass at drive wheels, $J_6 \triangleq m_v r_w^2$, where r_w is the radius of the wheels; K_1, C_1 are respectively the equivalent stiffness and damping coefficient of the input shaft; K_2, C_2 are correspondingly the stiffness and damping coefficient of the output shaft; $K_{31}, K_{32}, C_{31}, C_{32}$ are correspondingly the stiffness and damping coefficient of the half-shaft in the two branches; i_g, i_0 are respectively the transmission and final drive ratio.

The model shown in Figure 1 is too complicated to be useful in the study and design of an automatic shifting controller for the system. To brief this model, two sides of the driveline are considered perfectly symmetric, the tires have a perfect adherence, and no transitory effect on tire ground contact is present.

Owe to the mentioned assumptions, two branches of the driveline can be collapsed into one having: $J_4 = J_{41} + J_{42}$; $J_5 = J_{51} + J_{52}$; $K_3 = K_{31} + K_{32}$; $C_3 = C_{31} + C_{32}$.

Moreover, consider the following assumptions: (1) the tires have a perfect adherence and no transitory effects on tire-ground contact, (2) the input shafts and output shafts of the two sub-gearboxes are infinitely rigid. The general model can be simplified by collapsing the driveline downstream of the torque converter into a simple linear spring-damper system. The combined parts operation can be described as corresponding with the final drive and differential. The detail of the simplified model is shown in Figure 3 and defined by the following lines of reasoning below.

In which, $J_e = J_1$, $J_g = J_2.(i_g.i_o)^2 + J_3.i_o^2 + J_4$; $J_v \triangleq J_5 + J_6 = J_5 + m_v r_w^2$; $i_\Sigma = i_g.i_o$; $K_d = K_1.i_g^2.i_o^2 + K_2.i_o^2 + K_3$; $C_d = C_1.i_g^2.i_o^2 + C_2.i_o^2 + C_3$.

Applying D'Alembert's principle to the model, the dynamic equation describes the engine and torque converter's pump can be written as:

$$J_e\dot{\omega}_e = T_e - T_b \tag{1}$$

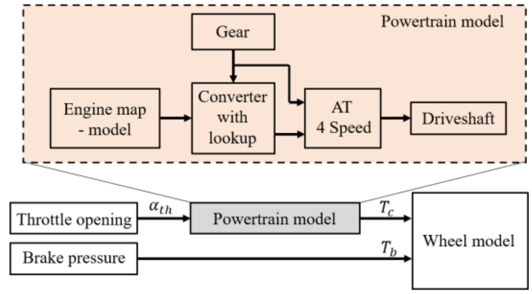

Figure 1. *Structure of the vehicle validation model*

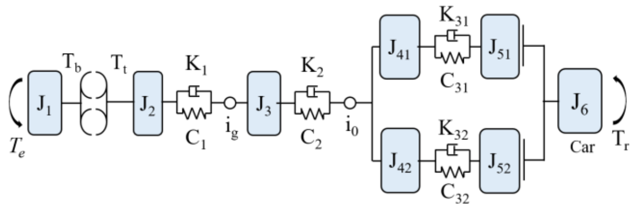

Figure 2. *General dynamic model of the AT powertrain system*

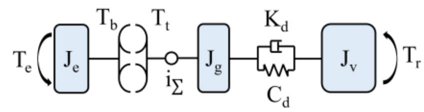

Figure 3. *The simplified dynamic model of the AT powertrain*

The dynamic equation demonstrates the parameters converted into one combination and described respectively as the final drive and differential can be written as:

$$J_g\dot{\omega}_g = T_t - [K_d.(\omega_g - \omega_v) + C_d.(\theta_g - \theta_v)] \tag{2}$$

and ω_v, θ_v respectively the speed and torsion of the wheels.

The dynamic equation demonstrates the wheels and vehicle body can be written:

$$J_v\dot{\omega}_v = \underbrace{K_d.(\omega_g - \omega_v) + C_d.(\theta_g - \theta_v)}_{T_x} - T_{br} - (F_a + F_r + F_g).r_{bx} \tag{3}$$

Where: T_x, T_{br} - correspondingly are the torque conveyed to drive wheels and brake torque. Resistance torque is determined as the following equation: $T_r = T_{br} - (F_a + F_r + F_g).r_{bx}$. The relation between velocity and wheel acceleration is: $a_v = \dot{v}_v = \dot{\omega}_v.r_{bx}$.

Gathering (1), (2), (3). The differential equations describing the dynamics of the simplified model are given by:

$$\begin{cases} J_e\dot{\omega}_e = T_e - T_b \\ J_g\dot{\omega}_g = T_t.i_\Sigma - [K_d.(\omega_g - \omega_v) + C_d.(\theta_g - \theta_v)] \\ J_v\dot{\omega}_v = K_d.(\omega_g - \omega_v) + C_d.(\theta_g - \theta_v) - T_r \end{cases} \tag{4}$$

The simplified dynamic model of the AT powertrain captures the essential part of the dynamic behavior of the driveline.

2.1. Engine model

The still model of the engine is used in this paper. The engine's torque T_e is considered as a function of engine speed ω_e and throttle position α in the percentage of opening and is defined by the looking-up table method.

$$T_e \triangleq f_1(\alpha, \omega_e) \tag{5}$$

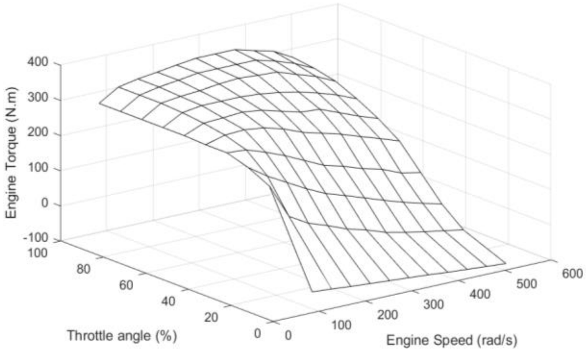

Figure 4. *Engine torque looking-up map*

2.2. Torque converter model

The main components of the torque converter include the impeller and turbine. The direction of power flow is always supposed to be from the impeller to the turbine. The impeller torque can be described by the following formula:

$$T_b = k(\phi)\rho_h d^5 \omega_1^2 \tag{6}$$

where: ρ_h is the stand for the density of the converter fluid; d is the pump diameter; $\phi = \frac{\omega_2}{\omega_1}$ is the speed ratio of the torque converter.

The turbine torque as in figure can be defined by the function below:

$$T_t = \zeta(\phi) T_b \tag{7}$$

The parameters $k(\phi)$ and $\zeta(\phi)$ are experimentally determined.

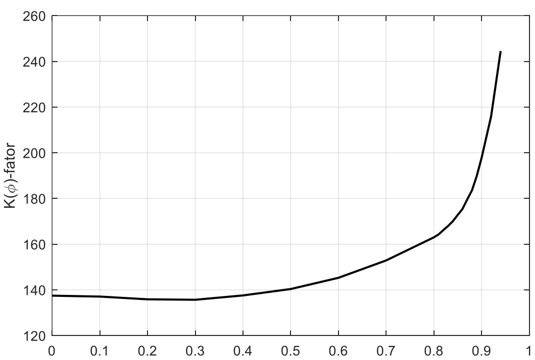

Figure 5. *Characteristics of the torque converter for* $k(\phi)$

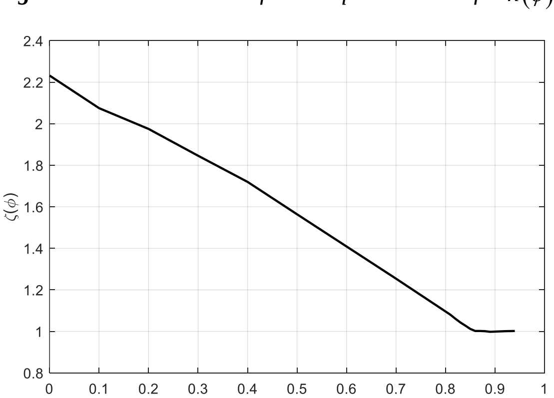

Figure 6. *Characteristics of the torque converter for* $\zeta(\phi)$

2.3. Resistance torque of the vehicle

The resistance torque delivered to drive wheels consists of the rolling resistance force F_r, the aerodynamic resistance force F_a, and the uphill driving force caused by gravity when driving on non-horizontal roads F_g.

The aerodynamic resistance force is approximated by:

$$F_a = 0.5.\rho_a.A_f.C_a.v_v^2 \tag{8}$$

The rolling resistance force is often modeled as:

$$F_r = m_v.g.\cos\beta.f \tag{9}$$

Uphill driving force induced by gravity when driving on a non-horizontal road is:

$$F_g = m_v.g.\sin\beta \tag{10}$$

where ρ_a is the density of the atmosphere; A_f is the frontal area of the vehicle; C_a is the aerodynamic drag coefficient; v_v is the vehicle speed; g is the acceleration of gravity, β is the slope angle of the road; f is the rolling friction coefficient.

3. Formulation of shifting schedule

This paper offers an algorithm for the gear-shifting

procedure on the 4-speed AT transmission. The AT powertrain gear-shifting process is controlled by a control unit. Its operation complies with a complex algorithm based on many various input parameters consisting of throttle and vehicle speed signals.

Assuming the case of shifting gears when the transmission is set to automatic shifting mode (D- Drive).

3.1. Up-shift schedule generation

The upshifting control algorithm is built by comparing the true or false conditions of two parameters, the throttle and the vehicle speed signals. The control unit receives the signals from sensors and compares them with the values of throttle and vehicle speed signals which are calculated and integrated into the engine control maps. The outline of the gear upshifting logic is shown in Figure 7.

When the system starts, it determines the gear-shifting control process based on the throttle position. In case the throttle position is higher than *a%*, the system continues to rely on the vehicle's velocity to determine the timing of gear shifts. If the automotive speed exceeds the v_{1a} threshold, the controller shifts the transmission from first gear to second gear. The system continues turning the gear positions to third gear if the velocity of the vehicle is over the v_{2a}. Once the speed is greater than v_{3a}, the controller regulates the transmission turned to fourth gear and maintains this position.

Similar to other throttle positions (*a, b, c, d…*) the different gear shifting timing will be decided by the controller. As mentioned above, the throttle (*a%, b%, c%...)* and velocity *(* v_{1a}, v_{2a}, v_{3a} *...)* thresholds used to control the upshift in the diagram in Figure 1 are calculated and integrated into the engine control map.

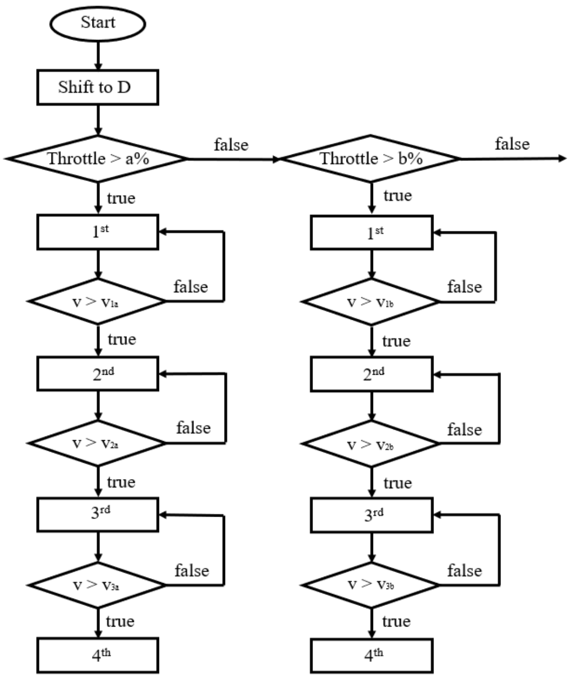

Figure 7. *The outline of the gear upshifting logic*

3.2. Down-shift schedule generation

The downshifting control algorithm is also built by comparing the true or false conditions of two parameters, the throttle and the vehicle speed signals. The control unit receives the signals from sensors and compares them with the values of throttle and vehicle speed signals which are calculated and integrated into the engine control maps. The outline of the gear downshifting logic is described in Figure 8.

Similar to other throttle positions (y, z, …) the different gear shifting timing will be decided by the controller.

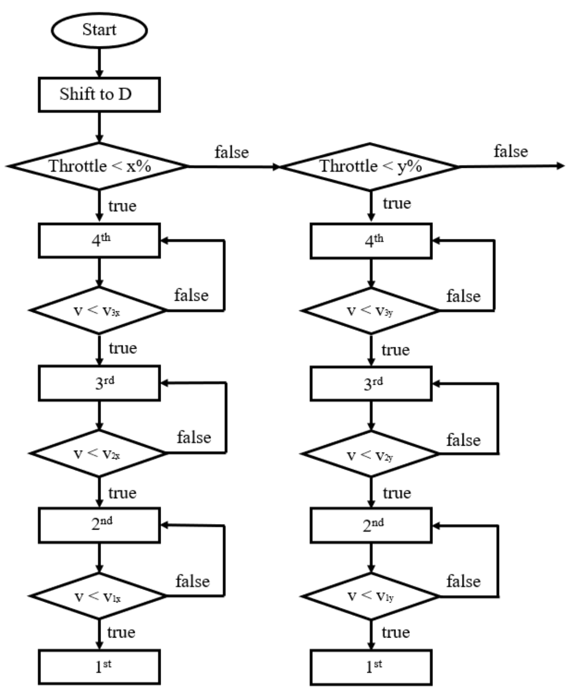

Figure 8. *The outline of the gear downshifting logic*

3.3. Analysis shifting process

The controller model is divided into two sections, one describes the speeds of the transmission. Another demonstrates the gear-shifting conditions. The timing to shift gears is interpolated from the percentage of the throttle opening and presented in Figure 9.

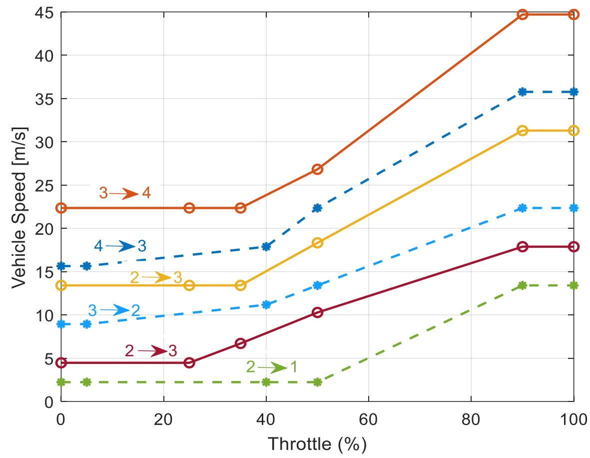

Figure 9. *The gear-shifting timing map*

For example, the first to second gear shifting will be conducted by the controller if: the vehicle throttle position is at 25%, and the vehicle velocity reaches nearly 5 m/s.

In case the throttle threshold reaches 50% and the velocity is 10 m/s, the controller is also going to shift from the first to the second gear.

Similar to the 90% throttle position, the first to second shifting will be accomplished when the vehicle velocity reaches 17 m/s.

The second to first-gear shifting will be carried out at the following times:

If the throttle position is *90%* and vehicle velocity is 14m/s, the second to first gear shifting procedure will be conducted.

Much the same as when the automotive reaches the 50% throttle threshold and the *2.5 m/s* vehicle velocity at the same time, the process is also going to take place.

The second to third and the third to fourth gear shifting procedures as well as their reverse processes comply with the same principle based on the map presented in Figure 9.

4. Illustrative results

In this paper, dynamic modeling of the automatic powertrain system using a torque converter. The dynamic model of the automatic transmission system is constructed and simulated using Matlab/Simulink software, as shown in Figure 10. The simulation results are described from Figure 11 to Figure 17.

The throttle valve opening controlled by the driver is depicted in Figure 11. When *t = 0* seconds, *60%* input throttle is given by the driver, The result is that the low engine speed leads to high engine torque as represented in Figure 12 and Figure 13. Vehicle automation could be efficiently achieved by maximizing the relation between the torque and the speed of the engine.

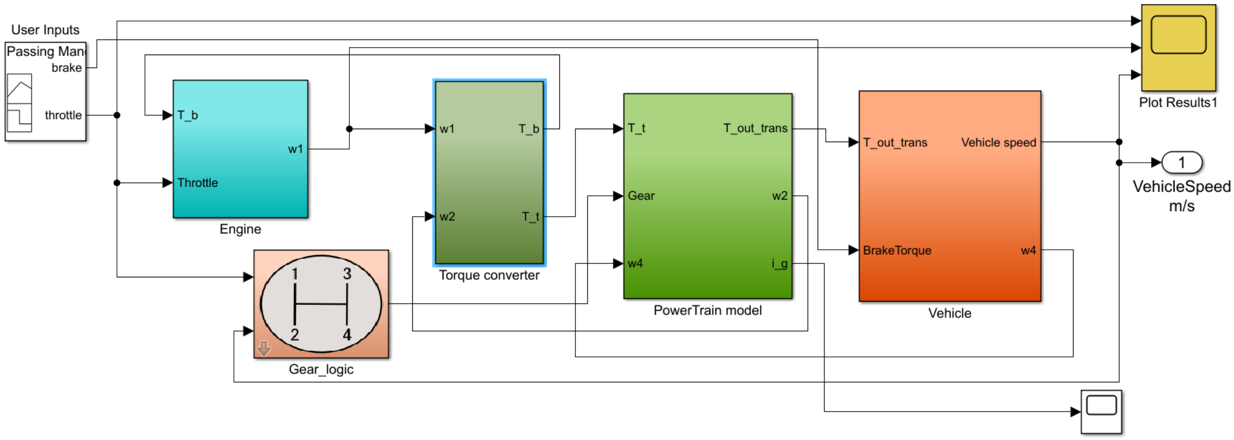

Figure 10. *The Simulink representation of AT powertrain*

Table 1. *Parameters of the model [14]*

Symbol	Value	Unit
$J_e; J_g; J_v$	0.1106; 19.086; 163.601	$kg.m^2$
C_d	45234.443	$N.m/rad$
K_d	4558.458	$N.m.s/rad$
i_0	3.07	--
r_w	0.32	m
$i_g^{min}; i_g^{max}$	0.67; 3.0	--

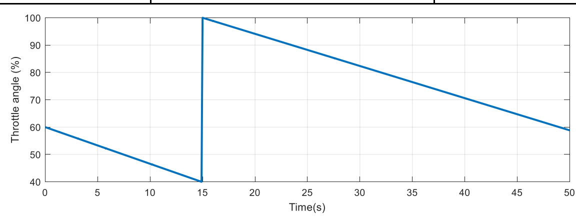

Figure 11. *Vehicle input throttle graph*

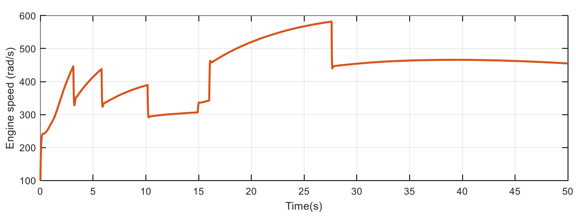

Figure 12. *Engine speed*

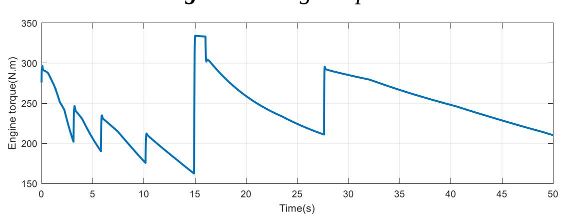

Figure 13. *Engine torque*

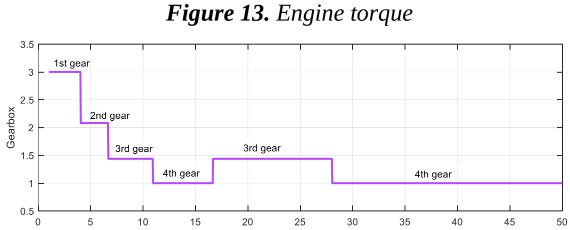

Figure 14. *The gear shifting process*

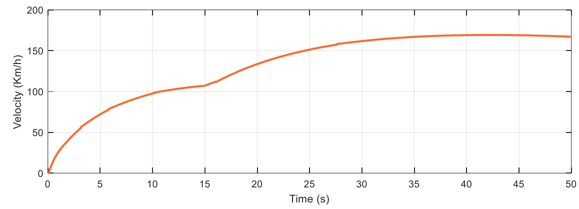

Figure 15. *Car speed*

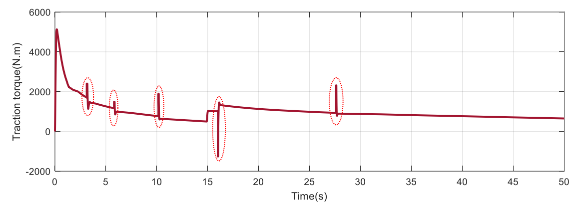

Figure 16. *Tractiontorque*

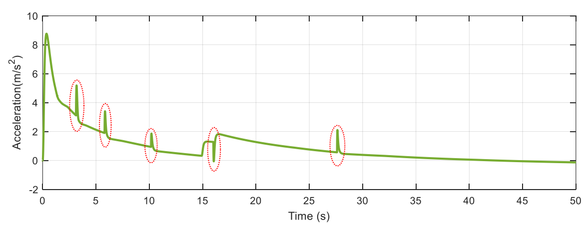

Figure 17. *Acceleration vehicle*

The gear-shifting process is depicted in Figure. 14. The car and its engine obtained an increase in speed before time *t = 3* seconds. The vehicle shifts into the first gear, at that time the engine speed has decreased significantly, and it achieves a high acceleration value once again.

At 6 and 11 seconds, upshift points 2-3 as well as 3-4 occurred. At this time, the vehicle's speed remains constant. At *t=15* seconds, the throttle is fully open, and the transmission is switched to the third gear in the car. Increasing the load leads to changes in the speed of the vehicle as well. As the throttle is depressed, the engine accelerates to *160 km/h* and then shifts to overdrive at *t= 28* seconds as shown in Figure 15. The car could shift into fourth gear.

Traction torque and acceleration are depicted in Figure 16 and Figure 17. The red-circled areas indicate the acceleration capabilities of the vehicle and its engine, corresponding to the gear-shifting process of the gearbox. It demonstrates a smoother gear-shifting process.

5. Concluding remarks

In this study, the mathematical models and the simulation models were successfully built including the AT powertrain dynamic model, engine model, and torque converter model. In addition, a gear-shifting control algorithm and an automatic controller based on that algorithm were also developed. The shifting timing was decided by the different conditions integrated into the controller. The presented results demonstrated the shifting principle and models offered were appropriate to the AT powertrain operation. These result in the efficiency of the control algorithm and the automatic controller with a fixed gear ratio. That is also the basic foundation of optimizing the control algorithm for the AT drive system with more gear ratios. Future directions, the automatic transmission shifting control can be continually developed to enhance the efficiency in performance and the fuel economy of the vehicle.

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