International Journal of Automotive Technology, Vol. 14, No. 2, pp. 189−194 (2013)
DOI 10.1007/s12239−013−0021−3

pISSN 1229−9138/ eISSN 1976−3832

TORQUE CONTROL OF A VEHICLE WITH ELECTRONIC THROTTLE CONTROL USING AN INPUT SHAPING METHOD

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(Received 15 November 2011; Revised 10 July 2012; Accepted 28 September 2012)

ABSTRACT−The shock-jerk phenomenon is usually observed in vehicles with manual transmission systems that are rapidly accelerating, and this phenomenon makes the passenger feel uncomfortable. This phenomenon can be minimized using torque control of the vehicle with throttle-by-wire or an ETC (Electronic Throttle Control) system. In this paper, the drivetrain of the vehicle is modeled to simulate the vehicle behavior, and the control strategy of ETC is studied to reduce shock and jerk characteristics using an input shaping method. The control logic is verified by using vehicle modeling and simulations.

KEY WORDS : Drivability, Torque control, Throttle-by-wire, Shock-jerk, Input shaper

1. INTRODUCTION

The most prominent factors that determine the marketability of today's automobiles are power, fuel economy, safety, and drivability; among these factors, drivability or quality of driving plays a larger role in marketability in the luxury car segment. The drivability of an automobile can be divided into the following subcriteria: cornering, acceleration, noise level, and vibration. Among these features, acceleration and vibration characteristics often appear to be closely related. This phenomenon is commonly seen in vehicles with manual transmissions when the torque increases quickly as the result of the acceleration pedal being pressed rapidly.

Automotive manufacturers evaluate drivability using criteria such as shock−jerk. Shock refers to impact felt by the driver when pressing down on the accelerator, and jerk refers to the duration of the vibration during this period of impact (Lee, 2010). This shock-jerk phenomenon occurs at tip-in when an ETC (Electronic Throttle Control) system is in place, as shown in Figure 1. The APS (Accelerator Position Sensor) detects the signal from the accelerator pedal, and the TPS (Throttle Position Sensor) is used as a feedback signal from the throttle valve, which is controlled by a microprocessor that responds to the accelerator pedal. The acceleration is measured using signals from the acceleration sensor placed on the driver seat, which can be an indication of the vibration felt by the driver. As shock−jerk increases, the vibration of a vehicle from front to back increases, which also causes discomfort to the driver. Therefore, reducing this unwanted phenomenon is an important area of vehicle research and development.

During tip in/out, the vibration of a vehicle is affected by several key factors that can be classified into three areas. The first is the change in engine torque (Song *et al*., 2001; Choi *et al*., 2002); the second involves the properties of the powertrain, including the clutch and axle; and the third involves the characteristics of body structures that support vehicle parts (Ko, 2002). Vibration typically occurs in a mass-spring system. In a vehicle, when the engine torque is transferred to the wheels, the driving shafts function as a spring, which results in vibration.

This study proposes that engine torque can be controlled by using an ETC system to minimize shock-jerk; this reduction in acceleration-induced vibration enhances overall drivability. Each part of the powertrain is modeled to simulate vehicle vibration, and the engine torque is

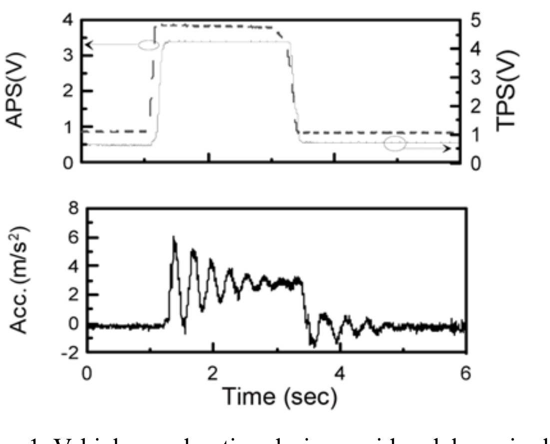

Figure 1. Vehicle acceleration during rapid pedal manipulation.

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controlled using the proposed input shaping method to minimize shock-jerk.

2. INPUT SHAPING METHOD

When there is a sudden transfer of torque from the engine to the wheels, the twisting of the shaft functions as a spring and induces vibration that negatively affects the drivability of a vehicle. If this shock-jerk could be reduced, it would be very helpful in generating smoother acceleration of the vehicle. An input shaping method can be applied during sudden acceleration to control the engine torque with an ETC system to enhance drivability.

The basic principles of the input shaping method are as follows (Shinghose, 2009). First, the system is considered a second-order damped vibration system, where the frequency of the system is assigned as ω and the damping as ζ. When n impulses were made in the time interval, the amplitude of the residual vibration is shown below (Vaughan *et al*., 2008).

$$V(\omega,\zeta) = e^{-\zeta\omega t_n}\sqrt{c(\omega,\zeta)^2 + S(\omega,\zeta)^2} \tag{1}$$

where

$$C(\omega,\zeta) = \sum_{i=1}^{n} A_i e^{\zeta\omega t_i}\cos(\omega\sqrt{1-\zeta^2}t_i)$$

$$S(\omega,\zeta) = \sum_{i=1}^{n} A_i e^{\zeta\omega t_i}\sin(\omega\sqrt{1-\zeta^2}t_i)$$

A_i is the amplitude of the impulse, and t_i is the time at which the i-th impulse is applied. To remove residual vibration, equation (1) should equal 0. The ZV (Zero Vibration) input shaper shown in equation (2) is used to accomplish this.

$$I(t) = A_1\delta(t) + A_2\delta(t - 0.5T_d) \tag{2}$$

where

$$A_1 = \frac{1}{1+K},\ A_2 = \frac{K}{1+K},\ K = exp\left(\frac{\zeta\pi}{\sqrt{1-\zeta^2}}\right)$$

and T_d is the period of vibration.

In the ZV input shaper, the time gap between the two input impulses is half of the period of the vibration (Vaughan *et al*., 2008). The application of the ZV input shaper in the system is shown in Figure 2. The initial system command is established as the step input; the step input and ZV input shaper go through the convolution process and create a modified command. The input shaping method can modify the original command into two step inputs.

Figure 3 shows the simulation results that demonstrate the effect of the step input on residual vibration before and after applying the ZV input shaper (Vaughan *et al*., 2008). The results with the input shaping method are shown with a dotted line where the residual vibration did not appear. The basic principle of the ZV input shaping method is to cancel the vibration due to the first input using the second input. Because the phase of vibration changes 180 degrees in a half cycle, the vibration from the first input and the vibration from the second input cancel each other out.

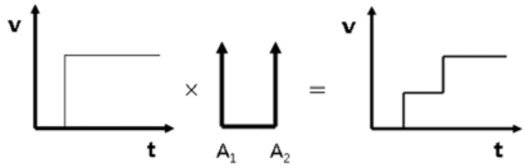

Figure 2. ZV shaping for a step input.

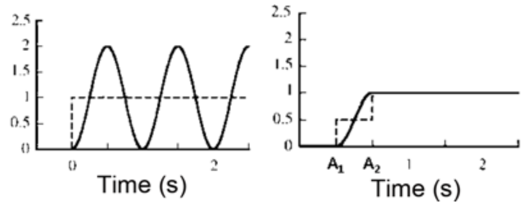

Figure 3. Simulation result of the system response to (a) step input and (b) ZV-shaped input.

However, when the natural frequency of this system is unknown, the timing of the second input is not exact; thus, the first vibration cannot be eliminated correctly. To counter this situation, input shaping is developed by factoring in errors from the natural frequencies; this is called ZVD (Zero-Vibration-Derivative) (Shinghose, 2009) and ZVDD (Zero-Vibration-Derivative-Derivative). ZV consists of two impulses, while ZVD has three, and ZVDD has four. Various input shaping techniques are currently under continuous development.

Currently, if the natural frequencies are known, the residual vibration can be eliminated with any type of input shaping method; even if there is an error in the natural frequencies of 30%, ZVDD can effectively minimize the residual vibrations (Vaughan *et al*., 2008). In this study, vehicle drivability improvement was investigated using the most basic ZV input shaping method.

3. VEHICLE MODELING

3.1. Rotational Speed Measurement

Vehicle modeling to simulate vehicle behavior with and without torque control using the input shaping method can be explained as follows. Generally, power delivery in an automobile begins in the engine and then is transferred to the clutch, transmission, driving shaft, and wheel. This flow of power can be expressed by simple modeling using a combination of the vehicle mass, springs, and dampers, as shown in Figure 4 (Park *et al*., 2005). The figure also displays the position of the attached sensors.

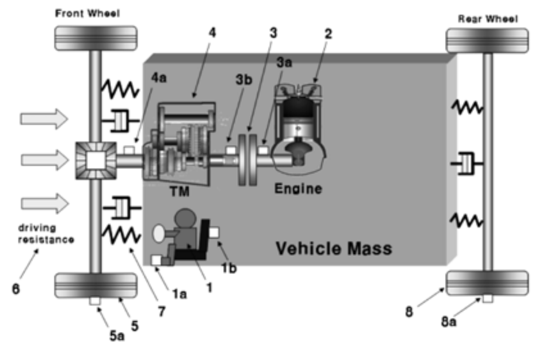

Figure 4. Schematic of the vehicle model and measuring points.

Numeral 1 refers to the driver's side, which is the most important position for the enhancement of drivability. Numeral 1a is the sensor located on the accelerator pedal. Numeral 1b is the acceleration sensor located on the driver's seat. Numeral 2 is the engine. Numeral 3a measures the speed of the clutch on the engine side, and numeral 3b measures the speed after applying the clutch. Numeral 4a measures the speed of the transmission. Numerals 5 and 8 indicate the front and rear wheels, respectively, and each has a 1-degree resolution encoder that measures wheel speed precisely. The wheels, when driven, slip due to relative motion, which may cause the front and rear wheels to have different rotational speeds. Numeral 6 is the driving resistance, and numeral 7 indicates the structures that support and connect the body.

The results of the speed measurement from the sensors are displayed in Figure 5. The top picture shows the detailed behavior right after tip-in, while the lower picture shows that the engine speed fluctuates several times. The high-frequency variation of the engine speed in the enlarged picture comes from the power stroke after combustion in each cylinder. During the initial stage of acceleration, a partial slip of the clutch plate leads to a slight difference in the speed on the engine side and transmission, but this difference soon disappears. The output speed from the transmission, or the input speed of the differential gear shown in the figure, is multiplied by the gear ratio, and the front and rear wheel speed is multiplied by the gear ratio and the final gear ratio. The front wheel speed has a large difference in amplitude and phase. That is, the driving shaft located between the differential gear and wheel is not rigid enough and ends up functioning as a spring.

When comparing the front wheel speed and the rear wheel speed, a phase delay can also be seen. Thus, the body and suspension supported by the front and rear wheels behave like a spring and damper, even though the vibration is small. In summary, the vibration during acceleration can be modeled with a spring-mass system.

Based on these results, dynamic vehicle models were constructed and are shown in Figure 6 (Park *et al*., 2006). The equation of motion based on the above model can be determined as follows. For the engine and transmission, the dynamic equation is shown in equations (3) ~ (6). Although the clutch exists in the model, clutch slip is ignored during the simulation.

I denotes the mass moment of inertia, T denotes the torque applied to engine or transmission, M denotes the mass of vehicle, and F denotes the traction force applied to the wheel. K is the spring constant, and C is the damping constant. x, $\dot{x}$, $\ddot{x}$ are the longitudinal position, speed, and acceleration, respectively. θ, $\dot{\theta}$, $\ddot{\theta}$ are the angular position, speed, and acceleration, respectively. R_{tire} is the radius of the tire, R_{GR} is the gear ratio, and R_{FGR} is the final gear ratio. Subscript E denotes the engine, EF denotes engine friction, CL denotes clutch, T denotes transmission, TF denotes transmission friction, D denotes drivetrain, FW denotes front wheel, RW denotes rear wheel, V denotes vehicle, and RL denotes road load.

$$I_E\ddot{\theta}_E = T_E - T_{EF} - T_{CL} \tag{3}$$

$$T_{CL} = K_{CL}(\theta_{CL} - \theta_T) + C_{CL}(\dot{\theta}_{CL} - \dot{\theta}_T) \tag{4}$$

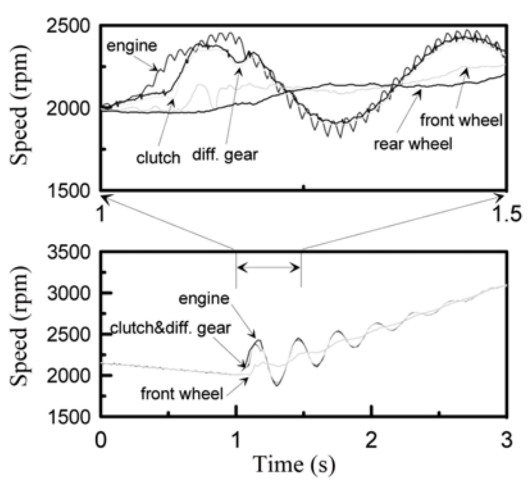

Figure 5. Measured speed during the tip-in test.

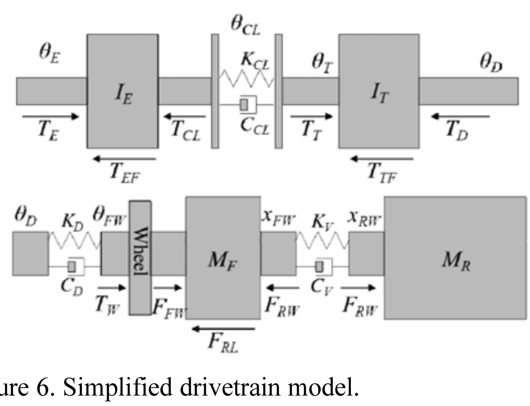

Figure 6. Simplified drivetrain model.

$$I_T\ddot{\theta}_T = T_{CL} - T_{TF} - \frac{T_D}{R_{CR}R_{FCR}} \tag{5}$$

$$\theta_D = \frac{\theta_T}{R_{CR}R_{FCR}} \tag{6}$$

$$T_D = K_D(\theta_D - \theta_{FW}) + C_D(\theta_D - \dot{\theta}_{FW}) \tag{7}$$

$$F_{FW} = \frac{T_D}{R_{tire}} \tag{8}$$

$$M_F\ddot{x}_{FW} = F_{FW} - F_{RW} \tag{9}$$

$$M_R\ddot{x}_{RW} = F_{RW} - F_{RL} \tag{10}$$

$$x_{FW} = R_{tire}\theta_{FW} \tag{11}$$

$$x_{RW} = R_{tire}\theta_{RW} \tag{12}$$

$$F_{RW} = K_V(x_{FW} - x_{RW}) + C_V(\dot{x}_{FW} - \dot{x}_{RW}) \tag{13}$$

$$F_{RL} = f(\dot{x}_{FW}) \tag{14}$$

Equations (7) ~ (14) represent the remaining components, such as the wheel and body. The constants required for these equations can be obtained experimentally and are shown in Table 1. Engine friction torque and transmission friction torque can be measured by placing the vehicle on a lift and maintaining a constant speed (Park *et al*., 2005). Engine inertia can be derived by manipulating the accelerator pedal and measuring the angular acceleration of the engine. The vehicle road load can be obtained by cost-down testing. Spring constants and damping coefficients were obtained from the acceleration test data. These steps were accomplished using the SIMULINK program, which is shown in Figure 7.

3.2. Model Validation

The vehicle model can be validated by comparing the results of the experiment and the simulation (Park *et al.,* 2006). In this study, the engine torque was calculated from the measured cylinder pressure, which is the input of the simulation. The outputs of the simulation are vehicle speed and engine speed, which are compared with the experimental results in Figure 8. The experimental condition is sudden acceleration in 2nd gear. The simulation results are demonstrated to be in good agreement with the experimental results. The vibration of the engine is larger than the vehicle vibration because the vehicle inertia is much larger than the engine inertia. The driver may feel shock-jerk during acceleration.

Table 1. Coefficients of Vehicle model.

Parameter	Physical object	Value
I_E	Engine inertia	0.27 [kg·m^2]
I_T	Transmission inertia	0.03 [kg·m^2]
K	Spring constant	200~700 [N/rad]
C	Damping constant	60~70 [N·s/rad]
R_{GR}	2nd gear ratio	8.39
R_{tire}	Tire radius	0.318 [m]
M_F / M_R	Front/rear vehicle mass	1500/300 [kg]
F_{road_load}	Road load	170~250 [N]

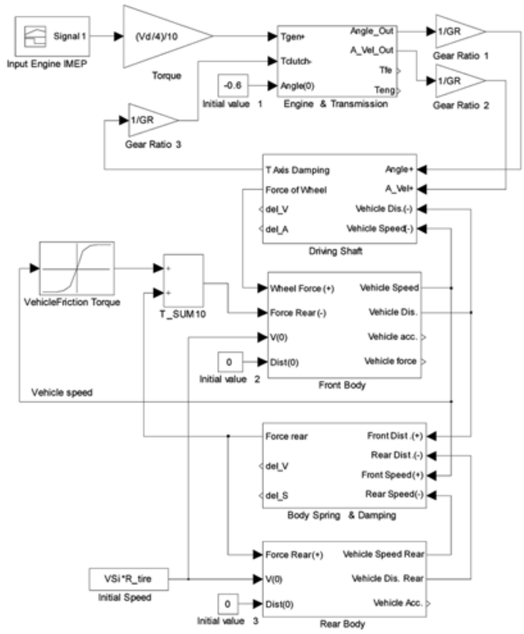

Figure 7. Simulation model of the vehicle using SIMULINK.

4. TORQUE CONTROL

4.1. Input Filtering Method

The simplest way to reduce shock-jerk during acceleration is to filter the accelerator command so that the engine torque increases smoothly (Lee, 2010). Figure 9 shows the effect of filtering on the input engine torque during acceleration in various cases, and this filter may lessen the acceleration-induced vibration. The shock-jerk is approximately ±3 m/s^2 with a step increase of the engine torque, and it is reduced as the filtering slope decreases. However, the shock-jerk remains at ±2 m/s^2 with 0.2 s delay (the 1st case), and the acceleration delay time increases as the slope decreases; thus, if the torque slope decreases, then the vehicle has a poor response during the initial stage of acceleration.

4.2. Input Shaping Method

To compensate for these shortcomings, the ZV input shaping method was applied. Figure 10 shows the engine torque that was modified using input shaping and the

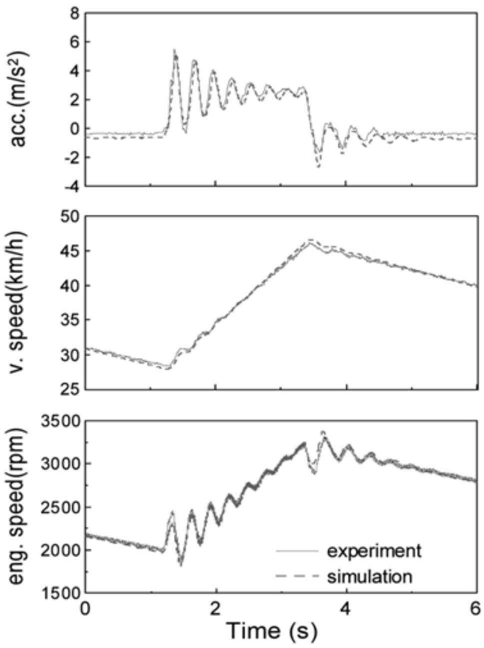

Figure 8. Simulation model validation.

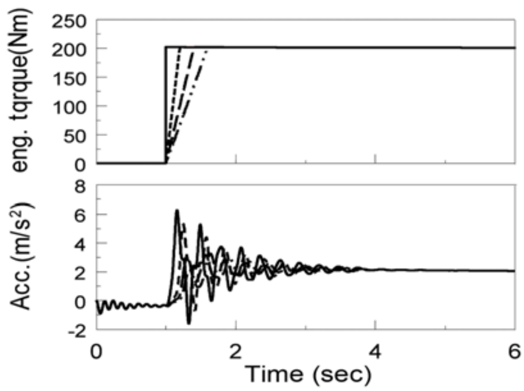

Figure 9. Simulation results of vehicle acceleration with input filtering.

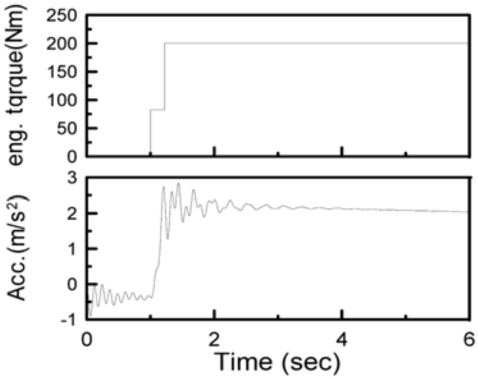

Figure 10. Simulation results of vehicle acceleration using the input shaping method.

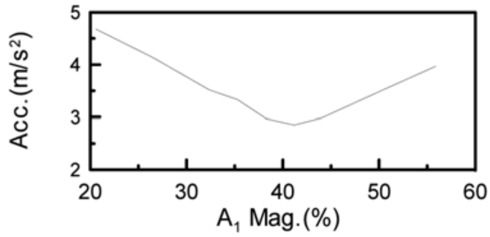

Figure 11. Effects of an A_1 impulse mismatch on vehicle acceleration.

subsequent acceleration of the vehicle. In figure 9, the vibration of the vehicle acceleration with the step response showed that the damping is approximately 20%, and oscillation period is approximately 0.4 s. Using these values, the input command in equation (2) can be made; A1 and A2 is set to 40% and 60%, respectively, from the size of the entire impulse, and the timing of the second impulse A2 is set to 0.2 s. The result of this simulation is given in Figure 10. The initial strong shock-jerk vibration has been reduced to ±0.5 m/s^2 with the delay time of 0.2 s, and only a few high-frequency vibrations remain. The engine, in general, has nonlinear characteristics; thus, there is a greater chance that the magnitude of A1 in the torque control may not be precise. The effects of the magnitude of A1 on peak acceleration are shown in Figure 11. The peak acceleration increases gradually near 40% of A1 over a wide range; thus, the size of the A1 impulse has little effect on the outcome.

The input shaping method has a shorter time delay development during initial acceleration than the filtering method; thus, the sluggish reaction of the automobile is minimal for the driver.

5. CONCLUSION

In this study, manual transmission vehicles that use an electronically controlled throttle system were studied to improve vibration at tip-in, which effects drivability. It has been determined through experimental research that the driving shaft acts like a spring in a vehicle's main powertrain and is responsible for the strong shock-jerk phenomenon. However, rather than using the standard filtering method, applying input shaping methods may be a more effective way to control engine torque, which reduces the shock-jerk from ±2 m/s^2 to ±0.5 m/s^2 with the same delay time.

Furthermore, rather than improving shock-jerk behavior by changing the characteristics of the vehicle hardware, controlling the engine torque via ETC would be more desirable.

ACKNOWLEDGMENT−This research was supported by the sabbatical program of Kumoh National Institute of Techology in 2010.

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