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An Object-Oriented Modular Simulation Model for Integrated Gasoline Engine and Automatic Transmission Control

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ABSTRACT

In this paper a computer simulation model for control system design of gasoline engines with an automatic transmission is presented. A modular programming approach has been pursued, and MATLAB/SIMULINK has been utilized as a programming environment. Engine/transmission systems are analyzed in the object-oriented fashion. Thus, easy construction of various computer models by assembling various objects is possible. An object in this paper represents a physical part, an equation, or an algorithm. The top level in the powertrain model consists of three classes: an engine, a transmission, and a driveline. Each class is designed to perform by itself. The construction procedure of a typical powertrain model together with supplementary explanation is demonstrated. It is expected that the whole program and individual class constructed in this paper are useful for the automotive engineers who design a new engine/transmission system and/or modify an existing system.

INTRODUCTION

In the automotive industry, the enhancement of the ride quality, fuel efficiency, and exhaust emission of a newly introduced vehicle has been continuously emphasized. This endeavor can be substantially achieved by improving the performance of the powertrain, which is a primary element of the noise, vibration, harshness, and fuel efficiency of a vehicle. Fig. 1 shows the schematic diagram of a typical powertrain system. An integrated control of the powertrain system refers to a combined control of the hitherto separated engine and transmission controls as shown in Fig. 2. The subject of powertrain control will play a central role among the many issues related to the control of the vehicle[1,2]. One challenging engineering problem is to build a software model with a computer which imitates or reproduces the behavior of a real plant. If this goal is achieved, then the cost and time needed to develop a better performing machine can be greatly reduced. It would be very effective if the dynamic performance of a newly configured powertrain system could be evaluated from the early stages of its design with a computer model before making its prototype.

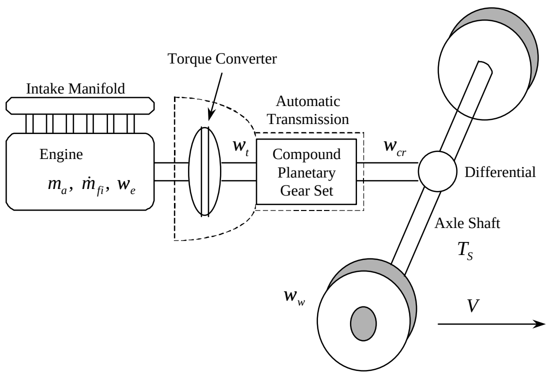


Fig. 1 Schematic diagram of a typical powertrain system

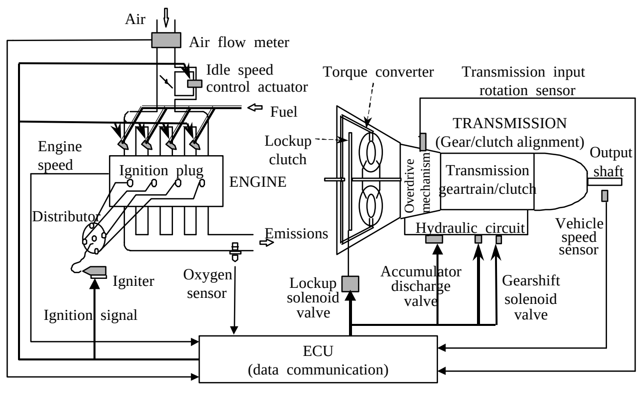


Fig. 2 Integrated control of an engine/transmission system

It would also be very useful if the model could predict the performance of a system in which a part is modified or newly added.

Wagner and Furry[3] have developed a real time hardware-in-the-loop simulation facility for the verification of automotive electronic controller software. Sayers and Mink[4] have described the architecture and usage of a simulation graphical user interface using object-oriented graphical data-base programs for vehicle dynamic models. Ciesla and Jennings[5] have presented a library of micro modules to evaluate the dynamic loadings on powertrain components and shift quality, and to develop control systems. Weeks and Moskwa[6] have presented an engine model for real-time control by using MATLAB/SIMULINK[7] block libraries.

In this paper, a modular programming technique based on the concept of object-oriented programming is adopted for the construction of a powertrain model. By setting proper boundaries between the components, the powertrain system is divided into several subsystems. Each subsystem, once its software module is constructed, is registered as an object or a class. With the object -oriented approach, a part of or the entire simulation model can be easily modified and reused by connecting the input/output signals of the objects selected. Once various objects for various products are made, the comparison among various products can be easily accomplished.

The object-oriented modular programming approach offers a powerful system modeling methodology: the encapsulation of data and functions, the reusability of the objects constructed, and the low cost for software development. The two key features of the modular programming method are: The complex entities of the entire system can be structured in a hierarchical order. Therefore, users can easily understand, describe, and modify the entire model, or a part of the model, by focusing a specific module in the hierarchical structure. The other is the reusability of the objects constructed. Since a group of objects can be registered as a single object, the programming effort can be greatly reduced because necessary objects can be replicated from other problems.

This paper proposes a computer software model for the powertrain system. The model provides various simulation environments for developing powertrain components and control systems. The program is constructed in such a way that the whole powertrain module is easily attached to a bigger class simulation module and its submodules can be independently used by themselves. The module in this paper is actually a part of a vehicle simulation model under construction by the authors. The vehicle model includes a powertrain module, a vehicle dynamics module, a suspension module, and steering and brake modules. Characterizing the attributes of the powertrain components, various modules for the engine, transmission and driveline are constructed in SIMULINK block library form. First, an object which represents a component, an equation or an algorithm is defined. Then, a bigger class is formulated by combining objects. The programming itself is accomplished in a bottom-up fashion, i.e. from a smaller object to a larger class, however the construction procedure in this paper is explained in a top-down fashion. The availability of various objects for physical components or for various algorithms allows the quick construction of various types of simulation models. The dynamic interaction among components and control systems can also be easily understood from the hierarchical structure of the model. Once a model is built, it facilitates both understanding of the entire configuration of the system, and the rapid evaluation of the performance of a new product by substituting the new object for the old.

SIMULATION ENVIRONMENT

An object is defined as a concept, an abstraction, or a thing with crisp boundaries and meaning for the given problem. It provides a programming basis for computer implementation. Examples of objects in this paper are components, equations, and algorithms.

An (object) class is defined as a group of objects with similar properties, common behavior, and common relationships to other objects. Examples of classes are engine, transmission, suspension system, brake system, fuel injector, etc. Construction of a class is completed once all objects in the class are concretized. Once all the contents in the class are fixed, it can also be treated as an object. A module is a logical construct for an object, a class, or a group of classes when it represents a portion of a bigger object. Whether a specific module indicates an object or a class will be apparent from the context.

A computer model means a software implementation of the physical entity for the purpose of evaluating its performance before building it. A computer model, once it is established, can also be used for visualizing the physical entity, reducing the complexity of the system, and communicating with customers.

In this paper, MATLAB and SIMULINK are used as a programming environment. SIMULINK provides a graphical user interface for building modules in block diagrams, and includes diverse on-line block libraries that run under MATLAB. Models are hierarchically constructed in a block diagram form using the methods of masking and grouping. How a model is organized and interacts with other parts can be retrieved by a top-down or a bottom-up approach based on the hierarchical structure. Furthermore, the blocks which represent particular objects or classes can be databased by registering them as new objects.

Fig. 3 shows a simulation model for a typical powertrain system in block diagram form. (Note that controllers are included in the model.) The three classes at the top level are an engine module, a transmission module, and a driveline module which includes rubber tires.

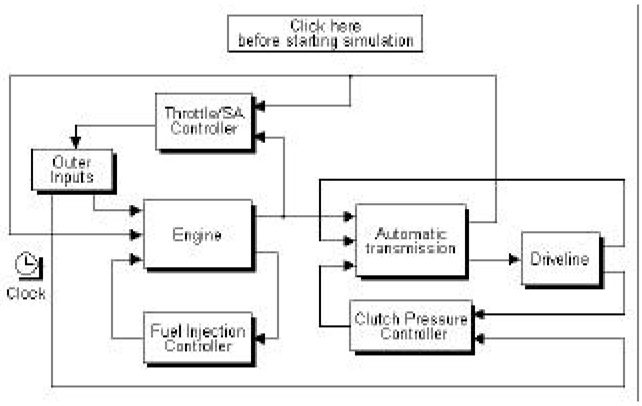


Fig. 3 A typical powertrain simulation block diagram

POWERTRAIN MODULE

ENGINE MODULE

Fig. 4 shows the signal flow diagram of a typical gasoline engine. It suggests that an engine module can be constructed with four systems. They are throttle body, intake manifold, fuel injection, and torque production. Various objects, for a particular component and incorporating different features, are made and databased in individual toolboxes. Various engine modules can then be built by selecting objects from their toolboxes and connecting them as in Fig. 4.

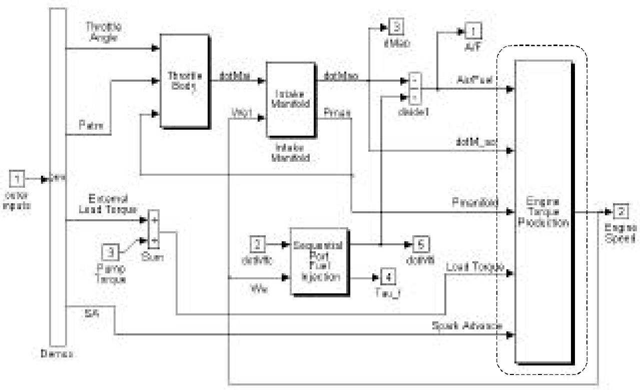


Fig. 4 An example of engine module

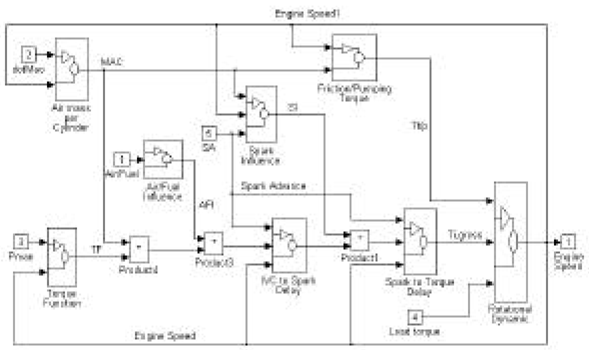


Fig. 5 Details of the engine torque production block in Fig. 4

In this section, an example of engine model by combining two researchers' results[8,9] is demonstrated. Both researchers dealt with the same V6-3800cc DOHC engine. Since it is claimed by Moskwa[8] that the engine torque generated in the Cho's model[9] is lower than the actual engine torque, the torque production part is adopted from Moskwa's work. All other parts including throttle body, intake manifold, and fuel injection are quoted from Cho's work(refer to Appendix A.1). Considering the length of this paper, only the internal structure of the engine torque production block in Fig. 4 is demonstrated in detail in Fig. 5. It is noted, however, that all other blocks in Fig. 4 can be constructed in the same way. The functions of the four blocks in Fig. 4 are summarized as:

The throttle body block calculates the total mass flow rate of air entering the intake manifold. The fuel injection block calculates equation (1) next under the assumption that the fuel spraying is completed before the opening of intake valve for better fuel atomization.

$$t_f \cdot \ddot{m}_{fi} + \dot{m}_{fi} = \dot{m}_{fc} \ . \qquad (1)$$

The effect fueling time constant t_f is the delay of the command fuel flow rate signal, $\dot{m}_{fc}$, from the ECU to the combustion chamber. The intake manifold block calculates the mass flow rate of air entering the combustion chamber. The torque production block calculates the engine indicated torque and rotational speed.

AUTOMATIC TRANSMISSION MODULE

Based on the works of Runde[10] and Cho[9], two automatic transmission (AT) modules are briefly discussed in this paper. One is a sprag type transmission equipped with a over-running sprag between the 1st clutch and the input sun gear, and the other is a clutch-to-clutch type transmission without the sprag (see Appendix A.2 for their dynamic equations.).

In the sprag type model, 1¡ 2 upshift is accomplished by transferring torque from the overrunning sprag clutch to the oncoming multi-plate clutch. The control input in this shift is the hydraulic pressure applied to the multi-plate clutch piston. While the control method for shifting is simple, the cost to manufacture the overrunning sprag clutch is quite expensive. On the other hand, for the clutch-to-clutch type transmissions, both the first (off-going) and the second (on-coming) clutches are controlled by the hydraulic pressure supplied to the clutch pistons. In the clutch-to-clutch type transmissions, the control problem is to obtain the desired clutch-pressures needed to achieve a good shifting performance.

Fig. 6 depicts a transmission model of the sprag type transmission. In the case of the clutch-to-clutch type AT, the reaction torque block does not appear and the system dynamic equations are different from those of the sprag type. The schematic diagram of the clutch-to-clutch type AT is omitted, however, the overall architecture is similar to that of the sprag type. The AT module in Fig. 6 can perform the lock-up and slip modes. The roles of six submodules in Fig. 6 are summarized as follows.

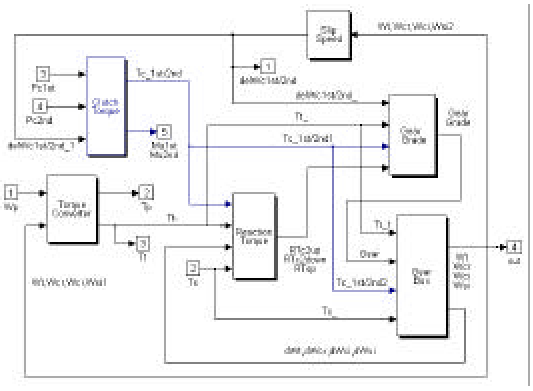


Fig. 6 Automatic transmission module of sprag type

A torque converter generally consists of a pump, a turbine, and a stator. The torque converter block calculates the pump and turbine torques. The steady-state torque converter operation in two modes are defined as:

converter mode (i.e., $w_t / w_p < 0.9$):

$$T_P = c_1 \cdot w_p^2 + c_2 \cdot w_p \cdot w_t + c_3 \cdot w_t^2 \ ,$$
$$T_t = c_4 \cdot w_p^2 + c_5 \cdot w_p \cdot w_t + c_6 \cdot w_t^2 \ . \qquad (2a)$$

fluid coupling mode (i.e., $w_t / w_p \geq 0.9$):

$$T_P = T_t = c_7 \cdot w_p^2 + c_8 \cdot w_p \cdot w_t + c_9 \cdot w_t^2 \ . \qquad (2b)$$

where T_P and T_t denote the torques of the pump and turbine respectively, and w_p and w_t denote their speeds. The converter pump is connected to the engine shaft and rotates at the same speed as the engine. The output torque from the turbine transmits directly into the transmission gear box. The slip speed block calculates the slip speeds of the 1st and 2nd clutches. The reaction torque block calculates the reaction torques acting on both sides of the clutches. The clutch torque block calculates the torques of 1st and 2nd clutches. The gear grade block determines the gear position according to the dynamic constraints. The gear box block calculates the converter turbine speed and the planetary gear speeds of reaction carrier, reaction sun, and input sun.

OUTER INPUTS AND DRIVELINE MODULES

For the outer inputs module in Fig. 3, the throttle angle, accessory load torque, spark advance, and atmospheric pressure are determined by the user or a controller module. A driveline is the power transfer device between the transmission and the vehicle body. A driveline model includes the stiffness of axle shaft, the wheel inertias, the slip of rubber tires, and the longitudinal inertia of the vehicle. The suspension dynamic is not included in the powertrain model. However, a separate module for the suspension system is under investigation. The details of the driveline module are not included in this paper. For related equations, refer to Appendix A.3.

CONTROLLER MODULE

AT CONTROLLER MODULE

The AT controller has both a shift schedule and a clutch pressure controller. The shift schedule determines the gear shifting point, while the clutch pressure controller regulates the hydraulic pressure acting on the clutches and bands. Since the hydraulic logics and circuits are very complicated, the dynamics of hydraulic circuits are not explained in detail in this paper. Instead, the following pressure profile (3a,b,c) is used in simulations[9,11].

1st gear: $P_{c1} = 1000$ kPa, $P_{c2} = 0$ kPa, (3a)

1→2 upshift: $\dot{P}_{c1} = \dfrac{-P_{c1}}{t_1}$, $\dot{P}_{c2} = \dfrac{1000 - P_{c2}}{t_2}$, (3b)

2nd gear: $P_{c1} = 0$ kPa, $P_{c2} = 1000$ kPa . (3c)

where t_1 and t_2 denote the first order time delay constants in the hydraulic circuits.

The shift schedule determines the starting point of the gear shift according to the vehicle speed and throttle angle. Fig. 7 depicts the shift schedule developed in [12].

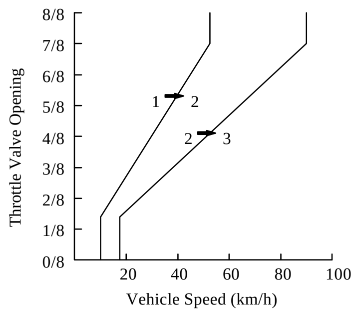


Fig. 7 A typical shift schedule

INTEGRATED ENGINE/AT CONTROLLER MODULE

It is not simple to design an integrated controller for the whole powertrain system due to the complexity, nonlinearity, and uncertainty of the system. The purposes of AT control are primarily to reduce the shock occurring while gear shifting, and to prevent excessive slip which may cause excessive wear of frictional parts. If the engine speed is not controlled during the gear shifting, the transmission controller will try to overcome the torque produced from the engine side during the inertia phase. This causes a decrease in fuel efficiency and results in a mismatch of clutch speeds. Thus, it is concluded that there are two objectives of powertrain control during the gear shift phase. One is the smooth acceleration (and jerk) transition to enhance the ride comfort. The other is the minimization of clutch energy dissipation to enhance the durability of friction elements. To achieve the first objective, the jerk level should be maintained at zero during gear shift. Concerning the second objective, the energy dissipation (E_c) during the shift is computed as:

$$E_{ci} = \int_{t1}^{t2} \left| T_{ci} \cdot \Delta w_{ci} \right| dt \; ; \quad i = 1, 2 \tag{4}$$

It is noted that the torque levels and slip speeds of the 1st and 2nd clutches can directly influence the energy dissipation quantity. Since the clutch torques are input control variables, the slip speeds of both the off-going clutch (1st clutch) and the on-coming clutch (2nd clutch) are controlled to achieve the second objective. With a closed loop control during upshift, the slip speed is reduced to zero.

In this paper, an integrated engine/AT controller is developed by using the sliding mode control method as indicated next in (5a)-(5d). The biggest change in gear ratio occurs in 1¡ 2 gear shift. Thus, 1¡ 2 gear shift deserves the primary operating point for testing the control performance. Since the reaction carrier speed (or the transmission output speed) w_{cr} is fairly easy to measure and is directly related to the smooth acceleration of the vehicle, the sliding surface can be defined as:

$$S_2 = w_{cr} - w_{cr,des}$$

where $w_{cr,des}$ represents a desired reaction carrier speed.

To improve the life of frictional components, the clutch energy requirement and shift duration must also be considered. Since the shift duration and slip speed directly influence the energy dissipation, control of the slip speeds of the on-coming and off-going clutches must be considered. Therefore, the other sliding surfaces can be defined as:

$$S_1 = \Delta w_{c1} - \Delta w_{c1,des} \;,$$

$$S_3 = \Delta w_{c2} - \Delta w_{c2,des} = \left(w_t - \frac{w_{cr}}{R_2} \right) - \left(w_t - \frac{w_{cr}}{R_2} \right)_{des}$$

where $\Delta w_{c1,des}$ and $\Delta w_{c2,des}$ denote desired slip speeds.

Since the reaction carrier speed is controlled by the sliding surface S_2, S_3 can be modified to:

$$S_3 = w_t - w_{t,des}$$

where $w_{t,des}$ represents a desired turbine speed. Engine torque control during the gear shift phase can substantially improve the shift quality and relieves the load on mechanical components. Engine torque is normally controlled by adjusting the throttle angle(a) or spark advance(*SA*). Therefore, the control variables for integrated control are throttle angle(a), spark advance(*SA*), and 2nd clutch torque(T_{c2}). From the torque converter model (2a,b), the relationship between the speed ratio, w_t / w_p, and the torque multiplication factor, T_t / T_p, can be expressed as:

$$\frac{T_t}{T_p} = R_{TC}\left(\frac{w_t}{w_p} \right)$$

Hence, by substituting the engine rotational dynamics(see Appendix A.1) into the pump torque T_p, the engine control variables can be solved as (5b) and (5c).

° Torque phase ($S_1 = \Delta w_{c1} - \Delta w_{c1,des}$):

$$T_{c2} = \left(\frac{1}{I_t} + \frac{1}{R_1 R_2 I_{cr12}} \right)^{-1} \left\{ \frac{T_t}{I_t} + \frac{R_d T_s}{R_1 I_{cr12}} - \left(\frac{1}{I_t} + \frac{1}{R_1^2 I_{cr12}} \right) T_{c1} + K_1 \mathrm{sat}\left(\frac{S_1}{f_1} \right) \right\} \tag{5a}$$

° Inertia phase ($S_2 = w_{cr} - w_{cr,des}$, $S_3 = w_t - w_{t,des}$):

$$a = \left\{ \frac{180}{p} \cos^{-1}\left(1 - \frac{\Omega_1 \Omega_2 + (\dot{m}_{ai} - \dot{m}_{ao})}{MAX \cdot PRI} \right) + 1.06 \right\} \Big/ 1.14456 \tag{5b}$$

$$SA = MBT + \sqrt{\frac{10^4}{3.801414}\left(1 - \Omega_3 \Omega_4 \right)} \tag{5c}$$

$$T_{c2} = R_2 \left\{ R_d T_s - \frac{T_{c1}}{R_1} + I_{cr12}\left[\dot{w}_{cr,des} - K_4 \mathrm{sat}\left(\frac{S_2}{f_4} \right) \right] \right\} \tag{5d}$$

where,

$$\Omega_1 = \frac{\# cyl \cdot w_e}{4p \cdot TF \cdot AFI \cdot SI_{(t - \Delta tST + \Delta tIS)}} \;,$$

$$\Omega_2 = R_{TC}^{-1}\left[I_t \dot{w}_t + T_{c1} + T_{c2} - K_2 I_t \mathrm{sat}\left(\frac{S_3}{f_2} \right) \right]_{(t+\Delta tIS)} + T_{f/p(t+\Delta tIS)} + I_e \dot{w}_{e(t+\Delta tIS)}$$

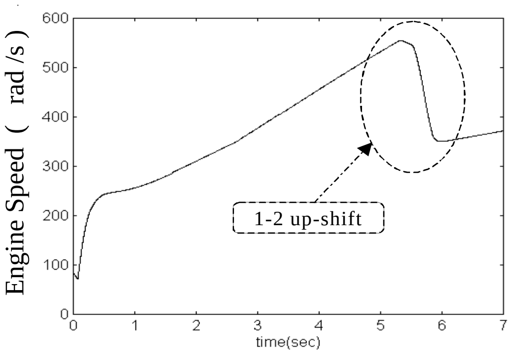
$$\Omega_3 = \left(\frac{\#cyl \cdot w_e}{4p \cdot TF \cdot AFI \cdot \dot{m}_{ao}} \right)_{(t-\Delta tIS+\Delta tST)}$$

$$\Omega_4 = R_{TC}^{-1}\left[I_t\dot{w}_t + T_{c1} + T_{c2} - K_3 I_t \mathrm{sat}\left(\frac{S_3}{f_3}\right)\right]_{(t+\Delta tST)}$$

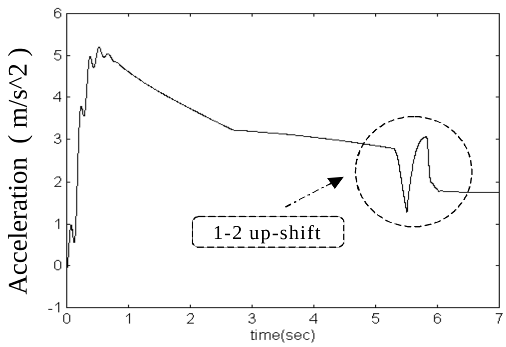
$$+ T_{f/p(t+\Delta tST)} + I_e\dot{w}_{e(t+\Delta tST)}$$

SIMULATION

The open loop characteristics of the powertrain model, which consists of a typical gasoline engine and a typical clutch-to-clutch type transmission, are depicted in Figs. 8-10. The air-fuel ratio and the spark advance are assumed fixed at 14.64 and 30 degrees, respectively. The engine speed and the vehicle acceleration, at fully open throttle angle, are shown in Fig. 8. The abrupt change of engine speed and acceleration during the 1¡ æ2 upshift is shown as between 5 and 6 seconds.



(a) Engine speed



(b) Vehicle acceleration

Fig. 8 Open loop start-up characteristics of the powertrain model with a typical gasoline engine and a clutch-to-clutch type A/T

Fig. 9 shows the details of the acceleration changes in the torque and inertia phases during the 1¡ æ2 upshift. The dynamics of the powertrain are relatively fast during the 1¡ æ2 upshift. The open loop characteristics agree well with the experimental results ([9],[13]) in terms of acceleration time and acceleration changes in the torque and inertia phases.

The profile of the output shaft torque during the upshift is not shown in this paper, however it is the same as Fig. 9. The upshifting begins with the decrease of output shaft torque (torque phase) followed by the torque bounce in the inertia phase. The decrease of output shaft torque in the torque phase is due to the second gear ratio being lower than the first. During the inertia phase, if the torque increase is too large, the passenger could experience an undesirable shock. On the other hand, if the torque increase during the inertia phase is not sufficient, the total shift time increases. If the shifting time is too long, the life of the clutch components could deteriorate.

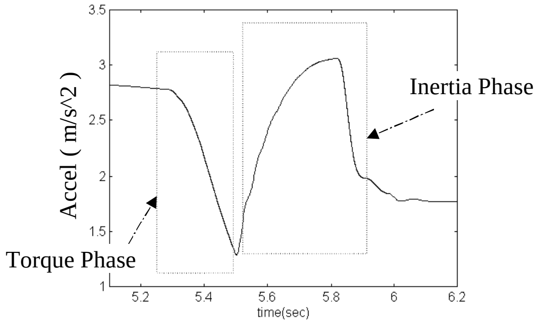
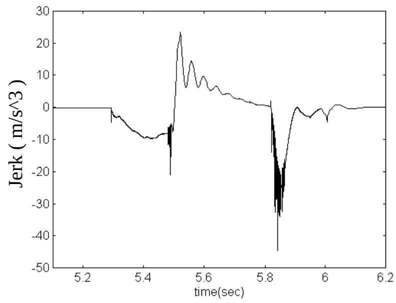
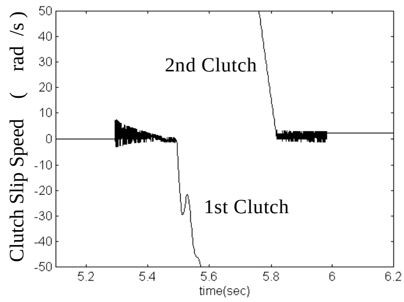


Fig. 9 Magnified profile of the vehicle acceleration during 1¡ æ2 upshift

Fig. 10(a) shows the open loop characteristics of the jerk motion during 1¡ æ2 upshift. It is remarked that its variation is too large at both the beginning and end of the inertia phase. The slip speed also oscillates during the torque phase and near the end of the inertia phase.



(a) Jerk motion during 1¡ æ2 upshift



(b) Clutch slip speed during 1¡ æ2 upshift

Fig. 10 Open loop characteristics of the jerk and clutch slip speedduring 1¡ æ2 upshift

These sharp jerk motions and oscillations are undesirable. Compared with the open loop characteristics, Figs. 11-12 demonstrate the improved characteristics of the closed loop system with variable structure control law (4a)-(4d). Fig. 11 and Fig. 12(a) shows a significant improvement in the acceleration and the jerk variations, which are directly related to the ride quality. Fig. 12(b) shows the disappearance of clutch slip variations. It is also shown that during the inertia phase, the 2nd clutch slip speed, Δw_{c2} , has been maintained at zero not increasing the output shaft torque. This was actually achieved by decreasing the engine torque via the integrated control of the engine and AT systems. However, the price for improving the shift quality and having less slip may cause a longer shift time. The increase in the shift duration can be seen by comparing Fig. 9 and Fig. 11.

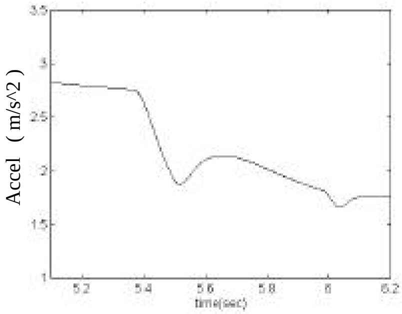
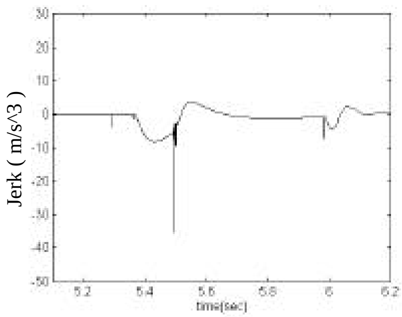
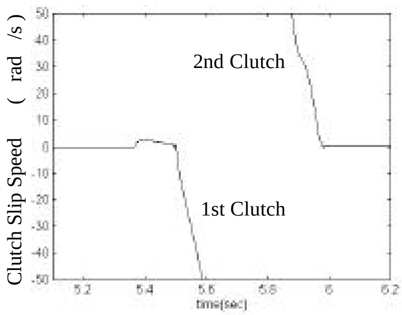


Fig. 11 Closed loop characteristics of the vehicle acceleration during 1¡ æ2 upshift



(a) Controlled jerk motion during 1¡ æ2 upshift



(b) Controlled clutch slip speed during 1¡ æ2 upshift

Fig. 12 Closed loop characteristics of the jerk and clutch slip speed during 1¡ æ2 upshift

CONCLUSIONS

In this paper, a simulation model for evaluating the dynamic behavior of powertrain systems and the control performance for various control laws has been constructed in the MATLAB/SIMULINK environment. Concretizing various objects associated with the powertrain components, equations, and control algorithms, the entire simulation model was completed by combining related objects in the hierarchical structure. The contents and construction procedure of the individual engine and transmission modules were explained in a top-down fashion, while the programming itself was done in a bottom-up fashion. The open loop start-up characteristics of the powertrain model constructed in this paper were demonstrated, and close agreement with experiment results in terms of acceleration time and pattern, and torque profiles during gear shifts has been confirmed.

A variable structure control for the whole engine and transmission system was investigated. The two control objectives, which are smooth acceleration transition and minimal energy dissipation during the gear shift, have been achieved by exercising control action on both engine torque and transmission clutch pressures.

The powertrain simulation model constructed in this paper can be usefully used for refining the control strategies of the integrated control of engine and AT systems. Furthermore, it can easily be extended to a larger simulation model for a whole vehicle simulation. Another possible application of the model lies in the area of real-time control and hardware-in-loop simulation in the dSPACE environment[14]. Last, it was shown that the engine module itself can be independently used for simulation by setting the torque from the converter pump to zero. Hence, various control strategies for the engine subsystems such as fuel injection system, idle speed control, etc., can be developed with the model.

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APPENDIX

A.1 ENGINE DYNAMICS

$$\dot{m}_a = \dot{m}_{ai} - \dot{m}_{ao} \ , \quad \dot{m}_{ai} = MAX \cdot TC \cdot PRI \ ,$$

$$\dot{m}_{ao} = c_T \cdot h_{vol} \cdot m_a \cdot w_e \ , \quad I_e \dot{w}_e = T_i - T_{f/p} - T_p - T_{load}$$

$$T_i = \left\{ \left[\frac{4p \cdot \dot{m}_{ao}}{\#cyl \cdot w_e} \cdot TF \cdot AFI \right]_{(t-\Delta tIS)} \cdot SI \right\}_{(t-\Delta tST)}$$

where,

$\dot{m}_a$	mass flow rate of air in the intake manifold (§ /sec)
$\dot{m}_{ai}$	mass flow rate of air entering the manifold (§ /sec)
$\dot{m}_{ao}$	mass flow rate of air entering the combustion chamber (§ /sec)
MAX	maximum flow rate through the throttle plate (§ /sec)
TC	normalized throttle influence
PRI	normalized pressure ratio influence
AFI	normalized air fuel influence
SI	normalized spark influence
$T_{f/p}$	friction and pumping torque (Nm)
T_p	torque converter pump torque (Nm)
T_{load}	exogenous load torque (Nm)
c_T	engine induction system physical constant
h_{vol}	cylinder volumetric efficiency
w_e	engine speed (rpm)
I_e	engine inertia (§ § ³)
$\#cyl$	number of cylinder
TF	torque function (Nm/g)
ΔtIS	time delay, intake to spark (sec)
ΔtST	time delay, spark to torque (sec)

A.2 AUTOMATIC TRANSMISSION DYNAMICS

¡ ã State equation at first gear: $I_{t1}\dot{w}_t = T_t - R_1 R_d T_s$

where,

$$I_{t1} = I_t + I_{si} + R_1^2 I_{cr} + \frac{R_1^2}{R_2^2} I_{ci}$$

I_t	converter turbine and chain inertia
I_{si}	input sun inertia
I_{cr}	reaction carrier inertia
I_{ci}	input carrier inertia
R_1	first gear speed reduction ratio
R_2	second gear speed reduction ratio
R_d	final drive speed reduction ratio
T_s	axle torque (both sides combined) (Nm)

¡ ã State equation while 1 ¡ æ 2 upshift:

¡ SPRAG Type ¡ µ

Torque phase: $I_{t1}\dot{w}_t = T_t - R_1 R_d T_s - \left(1 - \frac{R_1}{R_2}\right) T_{c2}$

Inertia phase: $I_t \dot{w}_t = T_t - T_{c2}$, $I_{cr12}\dot{w}_{cr} = \frac{T_{c1}}{R_1} - R_d T_s$

¡ Clutch-to-Clutch Type ¡ µ

$$I_t \dot{w}_t = T_t - T_{c1} - T_{c2} \ , \quad I_{cr12}\dot{w}_{cr} = \frac{T_{c1}}{R_1} + \frac{T_{c2}}{R_2} - R_d T_s$$

where, $I_{cr12} = I_{cr} + \frac{I_{si}}{R_1^2} + \frac{I_{ci}}{R_2^2}$, w_{cr} reaction carrier speed

¡ ã State equation at second gear:

¡ SPRAG Type ¡ µ $I_{t2}\dot{w}_t = T_t - R_2 R_d T_s$

¡ Clutch-to-Clutch Type ¡ µ $I_{t2}\dot{w}_t = T_t - \frac{R_1}{R_2} T_{c1} - R_2 R_d T_s$

where, $I_{t2} = I_t + I_{ci} + R_2^2 I_{cr} + \frac{R_2^2}{R_1^2} I_{si}$

¡ ã Clutch torque:

$T_{ci} = AR_i \cdot m_i \cdot P_{ci} \cdot \mathrm{sgn}(\Delta w_{ci})$, *i* =1,2

where,

AR_i total clutch area of *i* gear ¡ ¯ effective radius

$$m_i = 0.1316 + 0.0001748 \left|\Delta w_{ci}\right| \ , \quad \Delta w_{ci} = w_t - \frac{w_{cr}}{R_i}$$

A.3 DRIVELINE DYNAMICS

$$\dot{T}_s = K_s \left(R_d w_{cr} - w_{wf} \right) \ , \quad I_{wf}\dot{w}_{wf} = T_s - h_f F_{tf} - T_{rf}$$

$$F_{tf} = K_i \cdot i_d \ , \qquad M \cdot \dot{V} = F_{tf} - F_{tr} - F_a$$

where,

K_s	axle shaft stiffness (both sides combined)
w_{wf}	front wheel speed (rpm)
h_f	static ground-to-axle height of front wheel
I_{wf}	front wheel inertia (both sides combined)
K_i	tire slip proportionality (both sides combined)
M	vehicle mass
F_{tf}	tractive/braking force of front tire
F_{tr}	tractive/braking force of rear tire
T_{rf}	front tire rolling resistance (both sides combined)
F_a	aerodynamic drag
i_d	tire slip (£ ¥) for $h_f w_{wf}$ > 0.5 § ½ $i_d = 1 - \frac{V}{h_f w_{wf}}$

for $h_f w_{wf}$ ¡ Â 0.5 § ½ $i_d = 1 - \frac{h_f w_{wf} - V}{0.5\mathrm{sgn}(h_f w_{wf})}$