Object-Oriented Modeling for Gasoline Engine and Automatic Transmission Systems

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ABSTRACT: In this article a computer 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 used as a programming environment. Engine/transmission systems are analyzed in the object-oriented fashion that provides and ensures easy construction of various computer models by assembling various objects. An object in this article 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 here are useful for the automotive engineers who design a new engine/transmission system and/or modify an existing system. © 1999 John Wiley & Sons, Inc. Comput Appl Eng Educ 7: 107–119, 1999

Keywords: object-oriented model; MATLAB/SIMULINK; gasoline engine; automatic transmission

INTRODUCTION

In the automotive industry, the improvement 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 powertrain, which is a primary element of the noise, vibration, harshness, and fuel efficiency of a vehicle. Figure 1 shows the schematic diagram of a typical powertrain system. An integrated control of powertrain system refers to a combined control of the hitherto separated engine and

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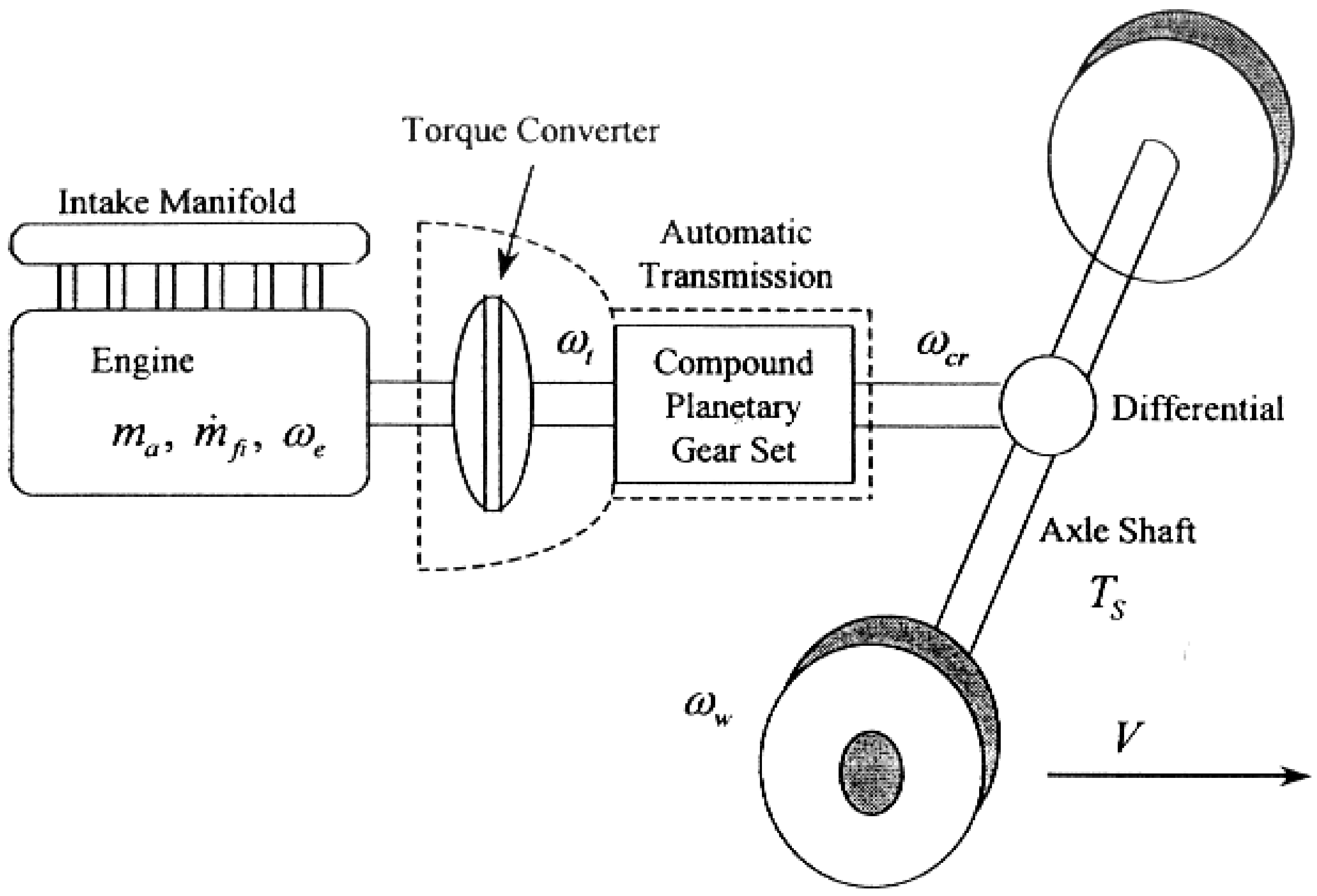


Figure 1 Schematic diagram of a typical powertrain system.

transmission controls, as shown in Figure 2. The issue of powertrain control will play a central role among many issues related to the vehicle control [1,2].

In the problems of engineering, to build a software simulator which imitates or reproduces the behavior of a real plant with a computer is very challenging. If this goal is achieved, then the cost and time to develop a better performing machine can be much reduced. In the early stage of powertrain design, it would be very effective if the dynamic performance of a system newly configured could be evaluated from a computer model before making its prototype. It is also very useful if the simulator can predict the performance of the system when a part is modified or newly introduced.

Wagner and Furry [3] developed a real-time hardware-in-the-loop simulation facility for the verification of automotive electronic controller software. Sayers and Mink [4] described the architecture and use of a simulation graphical user interface using object-oriented graphical database programs for vehicle dynamic models. Ciesla and Jennings [5] presented a library of micromodules to evaluate the dynamic loadings on powertrain components and shift quality, and

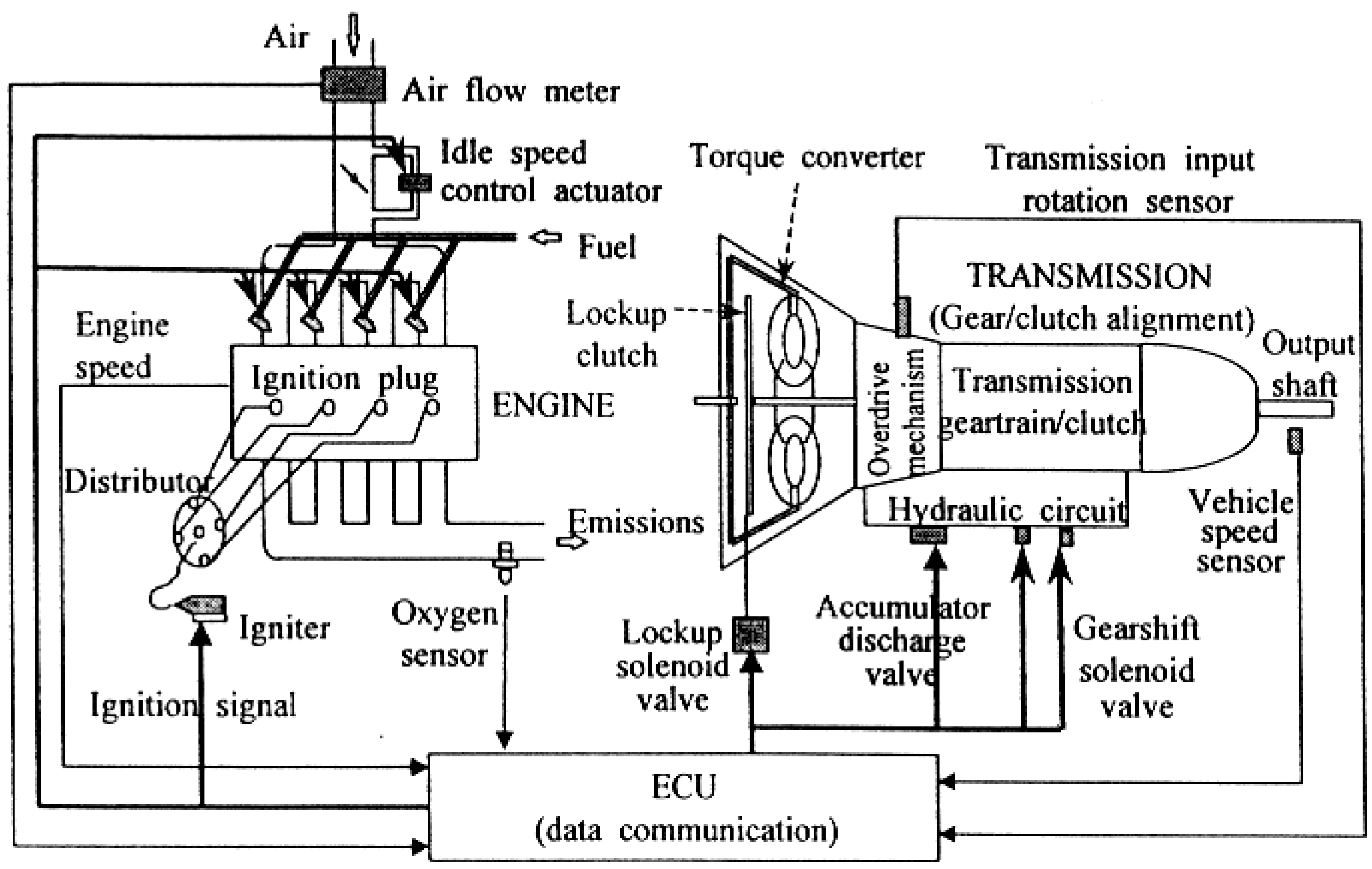


Figure 2 Integrated control of an engine/transmission system.

to develop control systems. Weeks and Moskwa [6] presented an engine model for real-time control by using SIMULINK block libraries.

In this article, a modular programming technique based on the concept of object-oriented programming is adopted for the construction of a powertrain model. The powertrain system is divided into several subsystems by setting proper boundaries between the components. Each subsystem, once its software module is constructed, is registered as an object or a class. With the object-oriented approach, part of or an 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: encapsulation of data and functions, reusability of the objects constructed, and low cost of software development. Two features of the modular programming method are as follows. The complex entities of the entire system can be structured in a hierarchical order. Therefore, users are easy to understand, describe, and modify the entire model or a part of it 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 much reduced when necessary objects can be replicated from other problems.

The contributions of this article are as follows. This report proposes a computer software simulator for the powertrain system. The simulator 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 larger simulation module (for example, a vehicle simulator) and its submodules are independently used by themselves. Characterizing the attributes of the powertrain components, various modules for the engine, transmission, and driveline are constructed in SIMULINK block library form. An object which represents a component, equation, or algorithm is first 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 article 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 simulation environment proposed here is to analyze various powertrain systems and design proper controllers using MATLAB/SIMULINK [7], which provides a graphical user interface for building modules as block diagrams. The powertrain model in this article is part of a bigger model, which is the vehicle simulator under construction by the authors. The vehicle simulator includes a powertrain module, a vehicle dynamics module, a suspension module, and steering and brake modules. Once a simulator is built, it allows easy understanding of the configuration of the system and rapid evaluation of the performance of a new product by substituting the object corresponding to the new one. The dynamic interaction among components and control systems can be also easily understood from the hierarchical structure of the simulator.

SIMULATION ENVIRONMENT

Object-Oriented Approach

An object is defined as a concept, an abstraction, or a thing with crisp boundaries and meaning for the problem given. It provides a programming basis for computer implementation. The examples of objects in this article 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. The examples of classes are engine, transmission, suspension system, brake system, fuel injector, etc. A class is concretized once all objects in the class are concretized. A class can be treated as an object, too, once all the contents in the class are fixed. A module is a logical construct for an object, a class, or a group of classes when it represents as a portion of bigger one. Whether a specific module indicates an object or a class will be apparent from the context.

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

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

Overview of the Powertrain Model

Figure 3(a) shows the structure of a vehicle simulation model. The model includes five modules: powertrain, suspension system, steering, braking, and body dynamics. Each module is designed to perform by itself as a simulation model.

Figure 3(b) shows the details of the powertrain module described in the object modeling technique [8]. The powertrain module consists of three classes: engine, transmission, and driveline. Appropriate attributes are attached to each class so that interchangeability can be ensured in the same class. The contents of each class are then concretized by concretizing all the objects underneath. The objects correspond to components such that throttle, fuel injector, spark control, EGR, clutches, bands, brakes, etc., are defined, grouped, and stored in the names of tool boxes.

Once objects are defined as SIMULINK blocks, the simulation model is constructed in a hierarchical structure. Figure 4 shows a simulation model for a typical powertrain 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.

POWERTRAIN MODULE

Engine Module

Figure 5 shows the signal flow diagram of a typical gasoline engine. It suggests that an engine module can be constructed with four submodules: throttle body, intake manifold, fuel injection, and torque production. Noting that several objects incorporating different features are databased for each submodule, an engine module is built by connecting the four objects corresponding to the four subsystems, as in Figure 6.

In this section, an example of engine model by combining two researchers' results [9,10] is demonstrated. Both researchers dealt with the same V6-3800cc DOHC engine. Since it is claimed by Moskwa [9] that the engine torque generated in the Cho's model [10] is lower than the actual engine torque, the torque production part is quoted from Moskwa's work. All other parts, including throttle body, intake manifold, and fuel injection, are quoted from Cho's work (see Appendix). Figure 7 shows the details of the engine torque production block which is adopted in Figure 6. All other modules in Figure 6 can be constructed in the same way. The functions of the four submodules in Figure 6 are summarized as follows.

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.

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

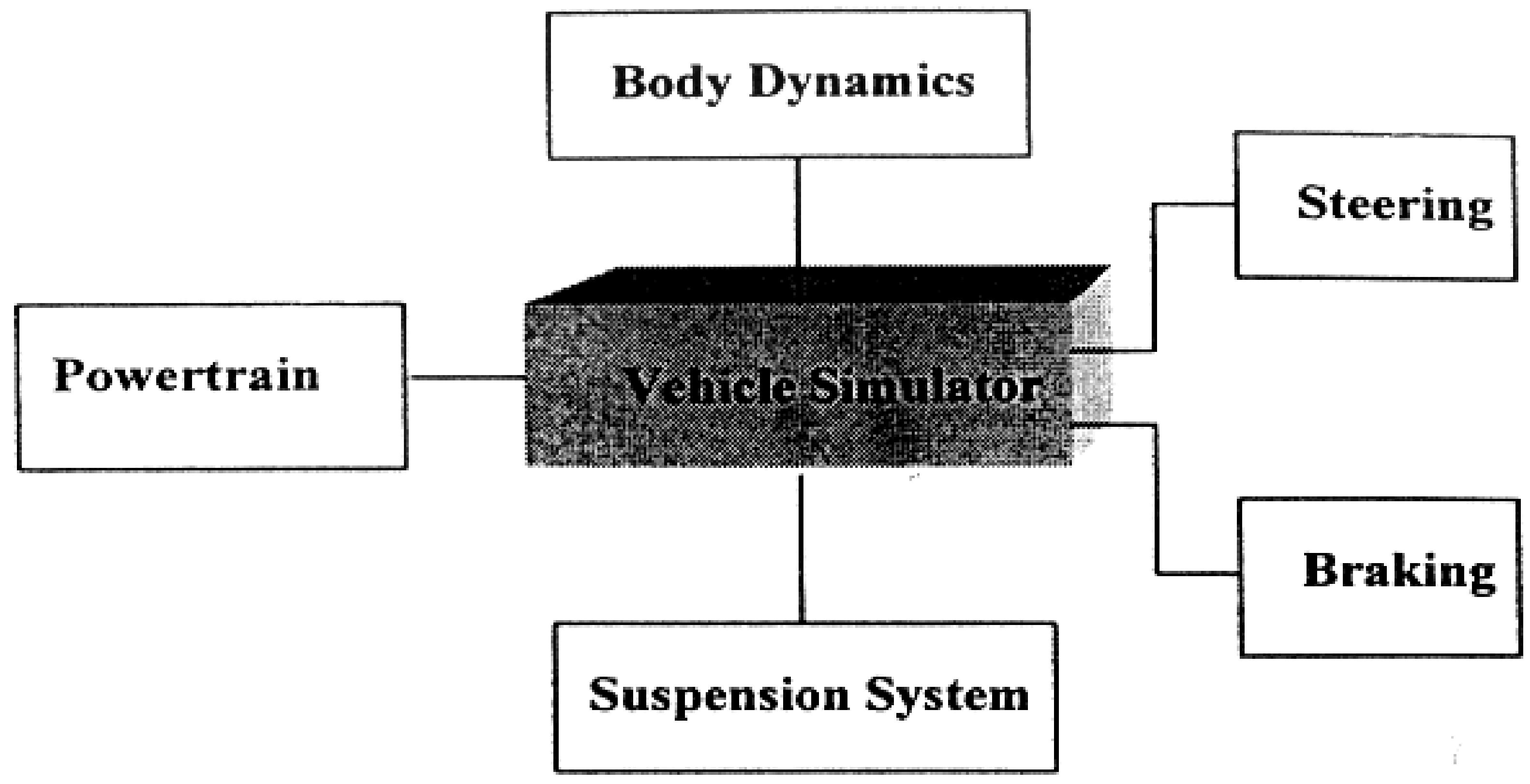
The effective fueling time constant τ_f is the delay of the command fuel flow rate signal, $\dot{m}_{fc}$, from the ECU to the combustion chamber. The intake manifold block calculates the mass flow rate of air entering the combustion chamber. The torque production block calculates the engine indicated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

Automatic Transmission Module

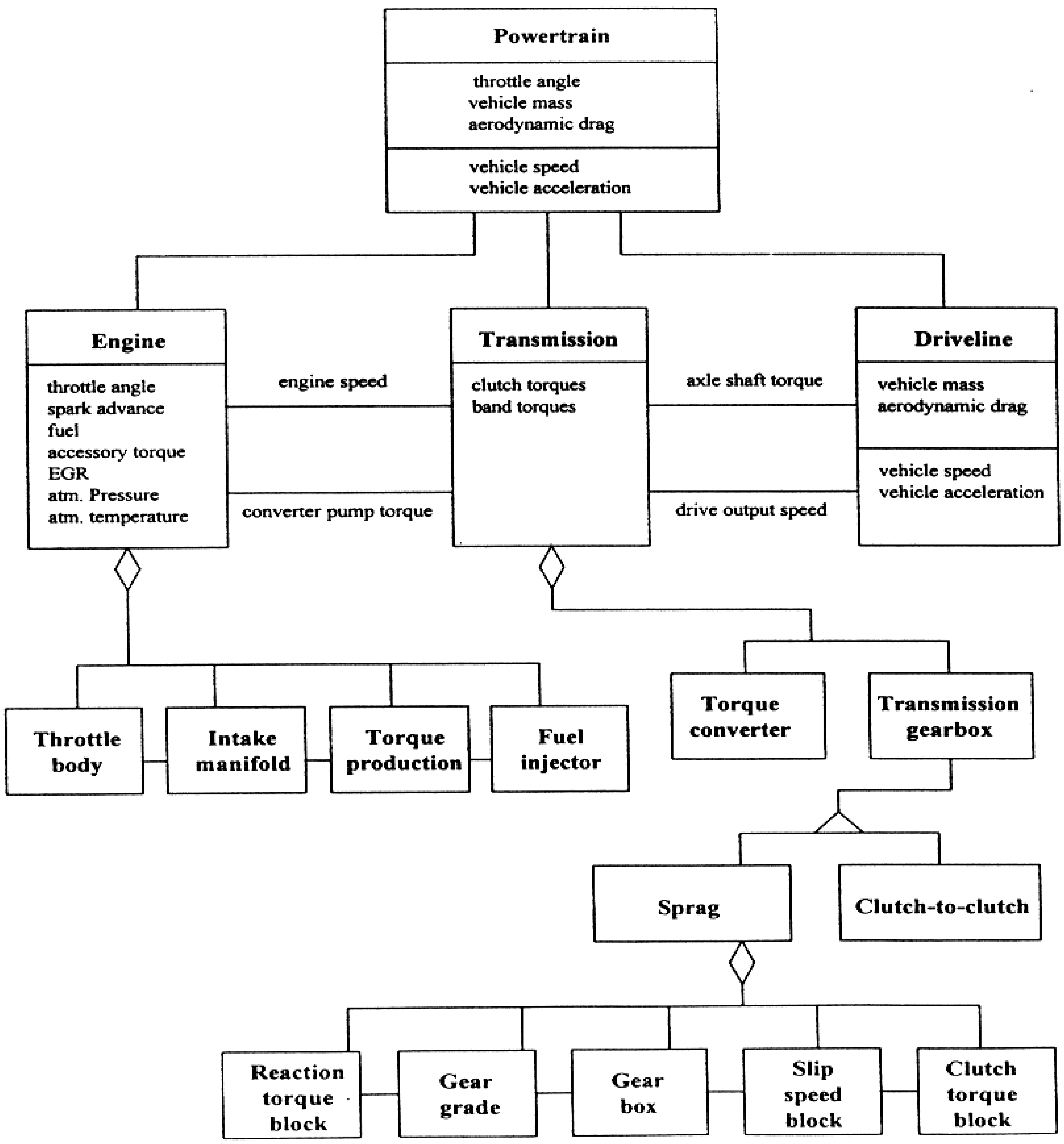
Based on the work of Runde [11] and Cho [10], two automatic transmission (AT) modules are illustrated in this article. One is a sprag-type transmission equipped with a overrunning sprag between the first clutch and the input sun gear; the other is a clutch-to-clutch type transmission without a sprag (see Appendix).

In the sprag-type model, $1 \rightarrow 2$ upshift is accomplished by transferring torque from the overrunning sprag clutch to the oncoming multiplate clutch. The control input supplied in this shift is the hydraulic pressure to the multiplate 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 transmission, both the first (off-going) and the second (oncoming) clutches are controlled by the hydraulic pressure to the clutch pistons. Therefore, in the clutch-to-clutch–type case, the control problem for these two clutch pressures should be solved to achieve a good shifting performance, such as minimum jerk or minimum energy dissipation.

Figure 8 shows the signal flow diagram of a typical sprag type AT. Figure 9 depicts a transmission model of the sprag-type transmission based on Figure 8. In the case of clutch-to-clutch–type AT, the reaction torque block is omitted in Figure 9, and the system dynamic equations are different from those of the sprag type. However, the overall architecture is sim-



(a)



(b)

Figure 3 Powertrain simulation model structure: (a) a vehicle simulator; (b) abstraction architecture of the powertrain module in object-oriented technique.

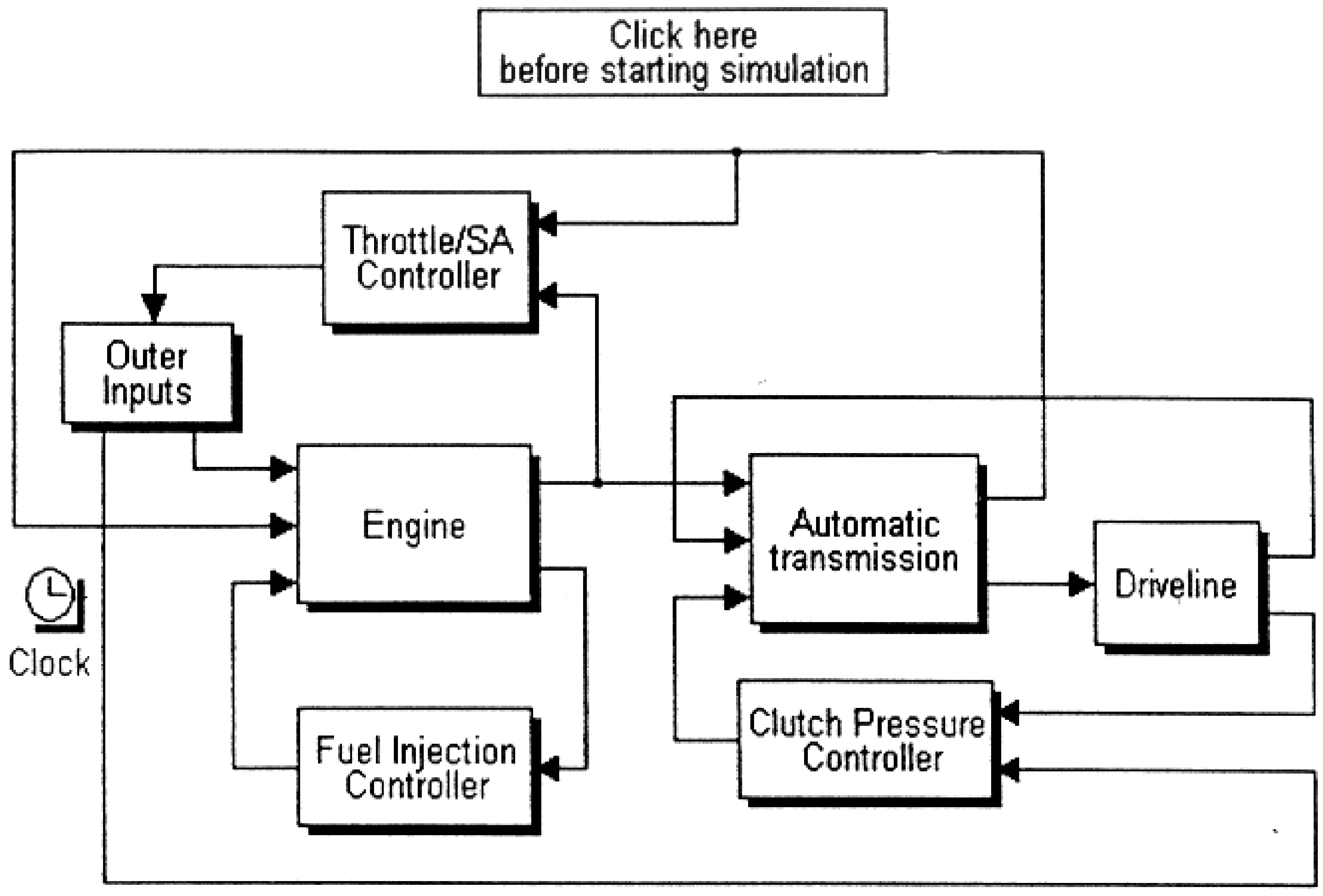


Figure 4 A typical powertrain simulation block diagram.

ilar to that of sprag type. In Figure 9, the two state variables which are used for 1 → 2 upshifting are the turbine speed and reaction carrier speed. The AT modules can perform the lockup and slip modes. The roles of six submodules in Figure 9 are summarized as follows.

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

a. Converter mode (i.e., $\omega_t/\omega_p < 0.9$):

$$T_p = c_1 \cdot \omega_p^2 + c_2 \cdot \omega_p \cdot \omega_t + c_3 \cdot \omega_t^2$$

$$T_t = c_4 \cdot \omega_p^2 + c_5 \cdot \omega_p \cdot \omega_t + c_6 \cdot \omega_t^2 \quad \text{(2a)}$$

b. Fluid coupling mode (i.e., $\omega_t/\omega_p \geq 0.9$):

$$T_p = T_t = c_7 \cdot \omega_p^2 + c_8 \cdot \omega_p \cdot \omega_t + c_9 \cdot \omega_t^2 \quad \text{(2b)}$$

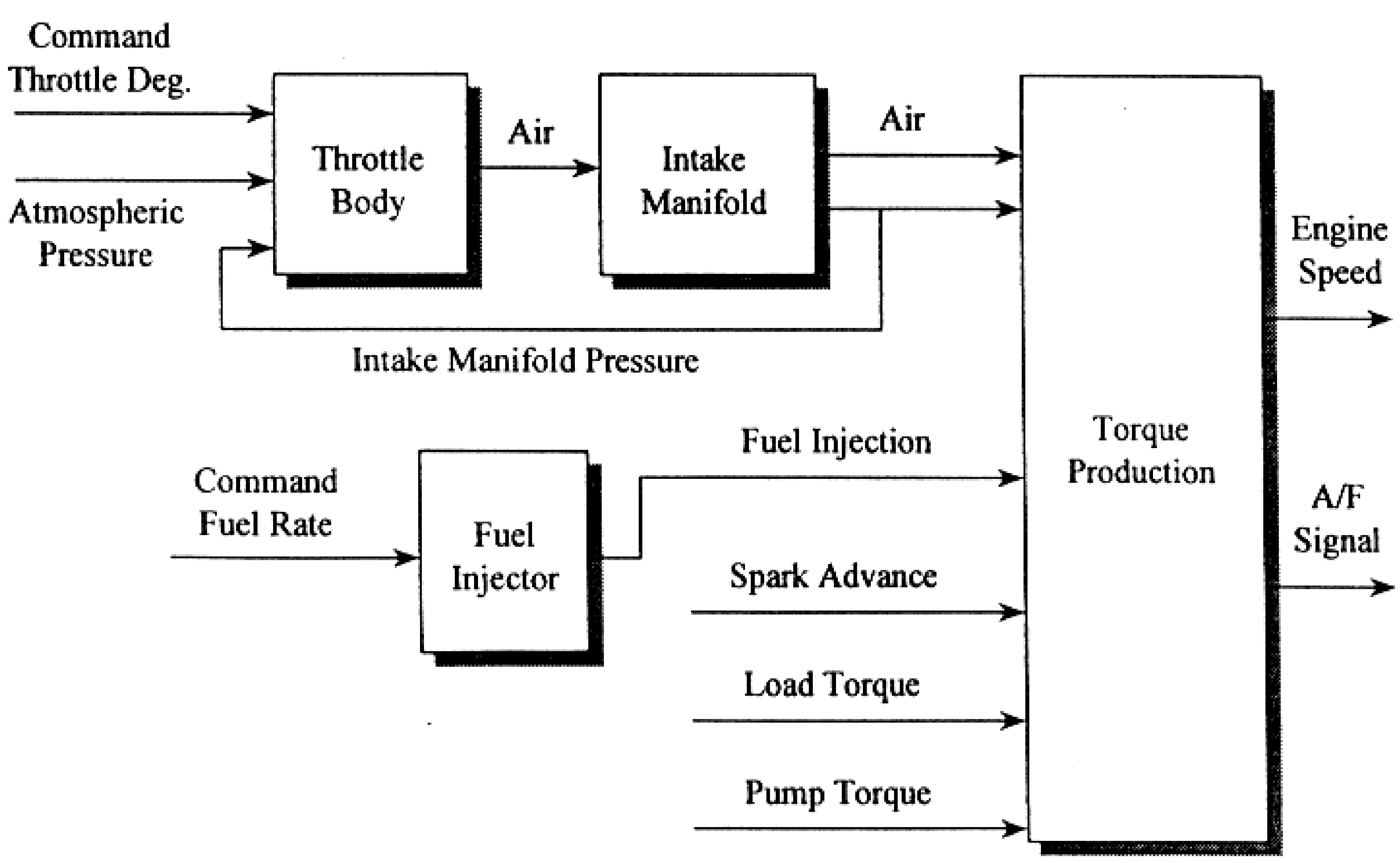


Figure 5 Signal flow diagram of a typical gasoline engine.

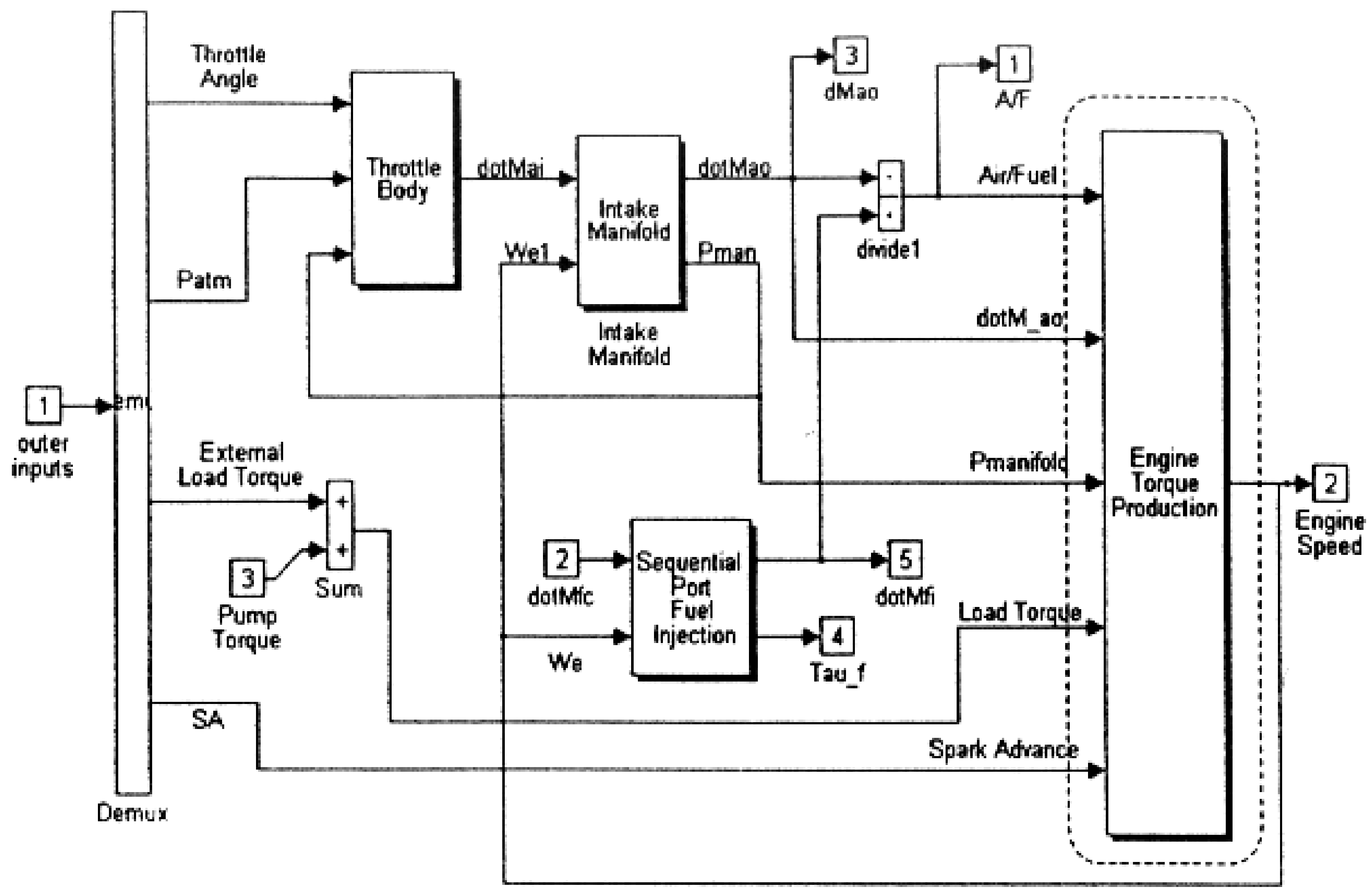


Figure 6 An example of engine module based on Figure 5.

where T_p and T_t denote pump and turbine torques, respectively, and ω_p and ω_t denote the speeds of those. The converter pump is connected to the engine shaft and rotates at engine speed. The output torque from the turbine transmits directly into the transmission gear box. The slip speed block calculates the slip speeds of the first and second clutches. The reaction torque block calculates the reaction torques acting on both sides of the clutches. The clutch torque block calculates the torques of first and second 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 Figure 4, the throttle angle, accessory load torque, spark advance, and atmospheric pressure are determined by users. A driveline is the power transfer device between transmission and vehicle body. A driveline model includes the stiffness of axle shaft, the wheel inertias, slip of rubber tires, and longitudinal inertia of a vehicle. The geometry changes of suspensions are not included, to keep the model simple. The module for the driveline is omitted here. Related equations are referred to in the Appendix.

CONTROLLER MODULE

AT Controller Module

The AT controller includes a clutch pressure controller and a shift schedule. The clutch pressure controller regulates the hydraulic pressure acting on the clutch and 1–2 band. The shift schedule determines the gear-

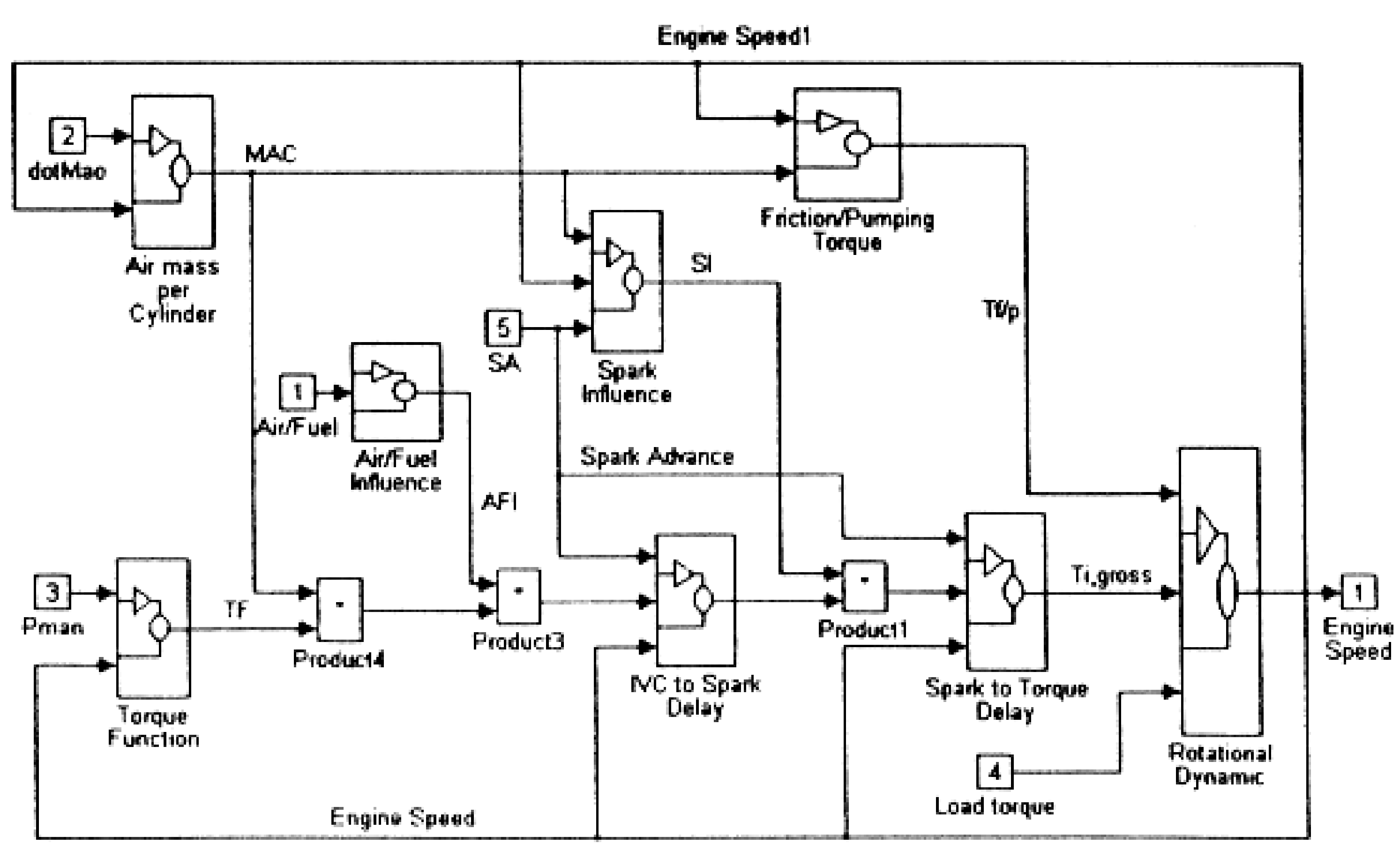


Figure 7 Details of the engine torque production block in Figure 6.

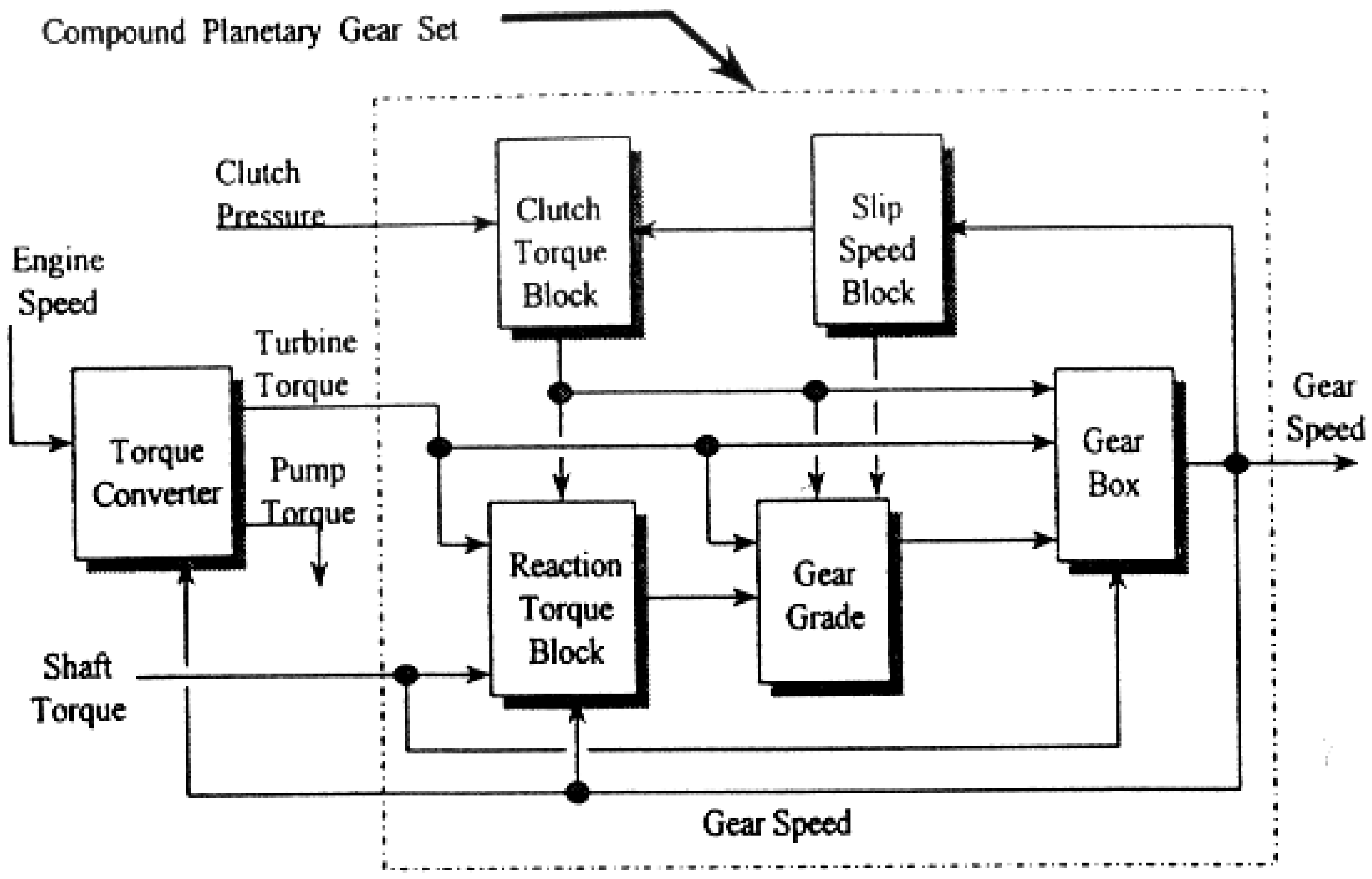


Figure 8 Signal flow diagram of a typical sprag-type automatic transmission.

shifting point. Since the hydraulic logics and circuits are very complicated, the dynamics of the hydraulic circuits are not included in detail here. Instead, the following pressure profile [Eqs. (3a)–(3c)] is used in simulations [10,12].

$$\text{First gear: } P_{c1} = 1000 \text{ kPa}, \quad P_{c2} = 0 \text{ kPa} \tag{3a}$$

$$1 \to 2 \text{ shift: } \dot{P}_{c1} = \frac{-P_{c1}}{\tau_1}, \quad \dot{P}_{c2} = \frac{1000 - P_{c2}}{\tau_2} \tag{3b}$$

$$\text{Second gear: } P_{c1} = 0 \text{ kPa}, \quad P_{c2} = 1000 \text{ kPa} \tag{3c}$$

where τ_1 and τ_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. Figure 10 depicts the shift schedule developed in Yamaguchi et al. [13].

Integrated Engine/AT Controller Module

It is not simple to design an integrated controller for powertrain system because of the high complexity, nonlinearity, and uncertainty of the system. The purpose of AT control is primarily to reduce the shock during gear shift. If the engine speed is not controlled during gear shift, the transmission controller will try to overcome the torque produced from the engine side during the speed phase. This causes a decrease of the fuel efficiency and brings about harsh matching of clutch speeds.

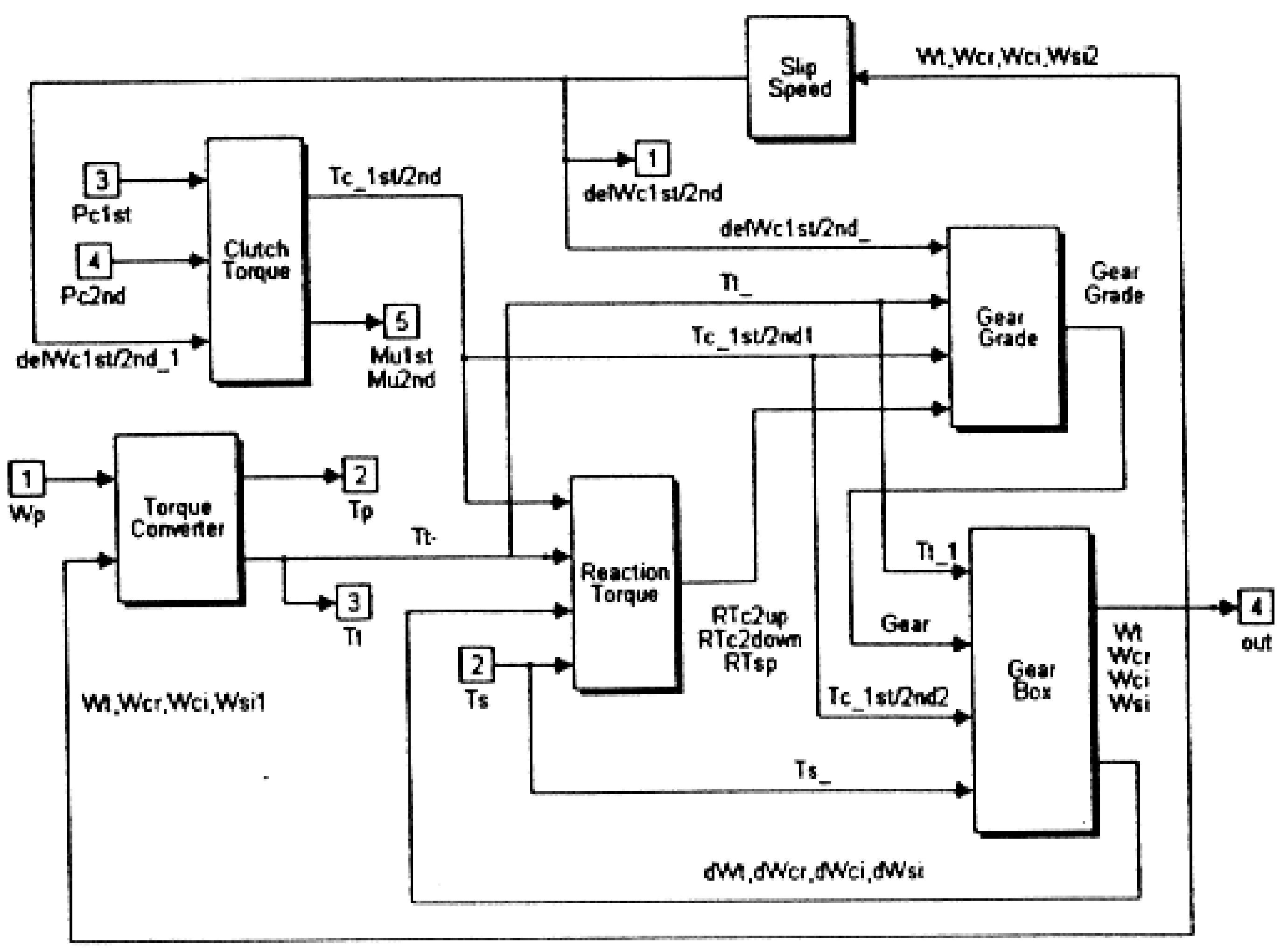


Figure 9 Automatic transmission module of sprag type.

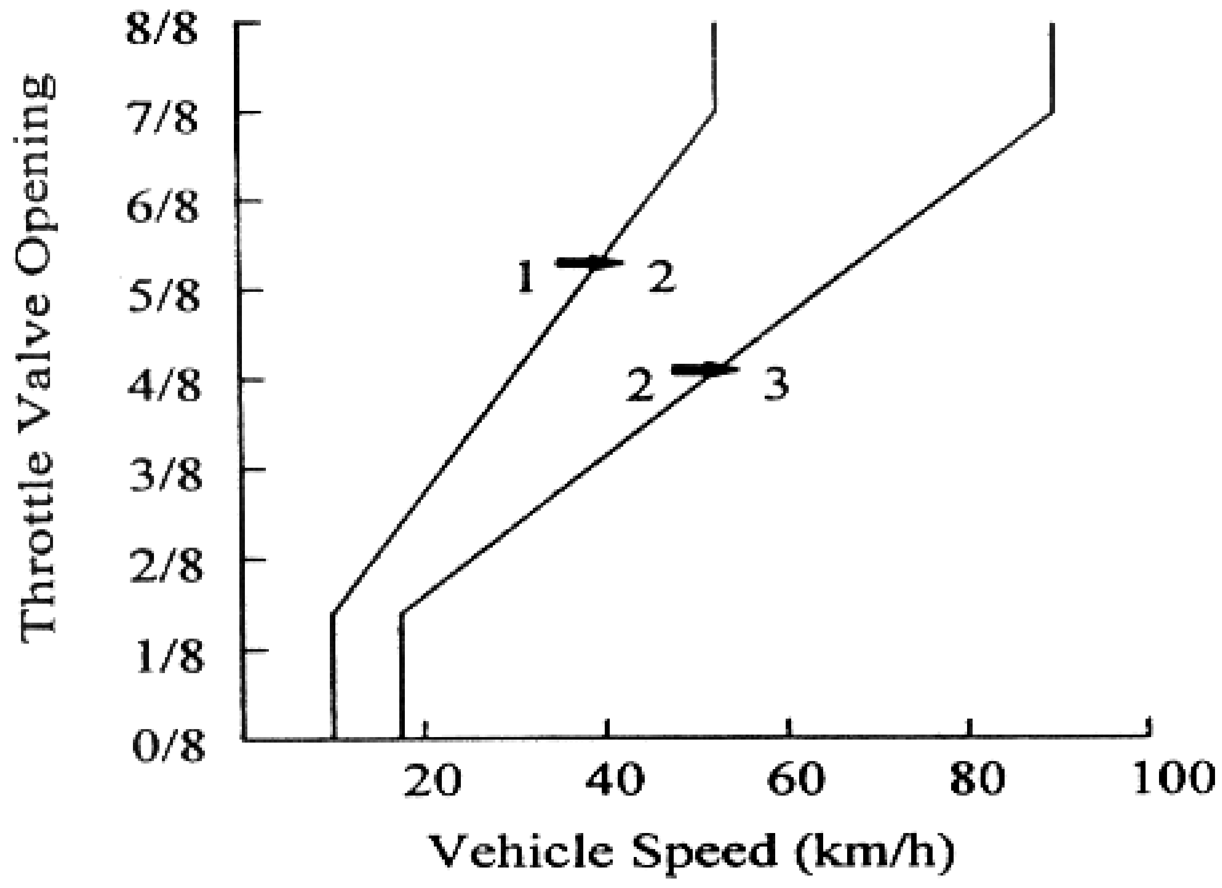


Figure 10 Shift schedule.

In this article, an integrated engine/AT controller is developed by using the sliding mode control method as indicated in Equations (4a)–(4d). Throttle angle (α), spark advance (SA), and second clutch torque (T_{c2}) are used as control inputs.

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

$$T_{c2} = \left(\frac{1}{I_t} + \frac{1}{R_1 R_2 I_{cr12}}\right)^{-1} \left\{\frac{T_t}{I_t} + \frac{R_d T_s}{R_1 I_{cr12}} - \left(\frac{1}{I_t} + \frac{1}{R_1^2 I_{cr12}}\right) T_{c1} + K_1 \operatorname{sat}\left(\frac{S_1}{\phi_1}\right)\right\} \quad (4a)$$

- Inertia phase ($S_2 = \omega_{cr} - \omega_{cr,des}$, $S_3 = \omega_t - \omega_{t,des}$):

$$\alpha = \left\{\frac{180}{\pi} \cos^{-1}\left(1 - \frac{\Omega_1 \Omega_2 + (\dot{m}_{ai} - \dot{m}_{ao})}{MAX \cdot PRI}\right) + 1.06\right\} / 1.14456 \quad (4b)$$

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

$$T_{c2} = R_2 \left\{R_d T_s - \frac{T_{c1}}{R_1} + I_{cr12}\left[\dot{\omega}_{cr,des} - K_4 \operatorname{sat}\left(\frac{S_2}{\phi_4}\right)\right]\right\} \quad (4d)$$

where

$$\Omega_1 = \frac{\#cyl \cdot \omega_e}{4\pi \cdot TF \cdot AFI \cdot SI_{(t-\Delta t_{ST}+\Delta t_{IS})}},$$

$$\Omega_2 = R_{TC}^{-1}\left[I_t \dot{\omega}_t + T_{c1} + T_{c2} - K_2 I_t \operatorname{sat}\left(\frac{S_3}{\phi_2}\right)\right]_{(t+\Delta t_{IS})} + T_{flp(t+\Delta t_{IS})} + I_e \dot{\omega}_{e(t+\Delta t_{IS})},$$

$$\Omega_3 = \left(\frac{\#\, cyl \cdot \omega_e}{4\pi \cdot TF \cdot AFI \cdot \dot{m}_{ao}}\right)_{(t-\Delta t_{IS}+\Delta t_{ST})}, \text{ and}$$

$$\Omega_4 = R_{TC}^{-1}\left[I_t \dot{\omega}_t + T_{c1} + T_{c2} - K_3 I_t \operatorname{sat}\left(\frac{S_3}{\phi_3}\right)\right]_{(t+\Delta t_{ST})} + T_{flp(t+\Delta t_{ST})} + I_e \dot{\omega}_{e(t+\Delta t_{ST})}.$$

SIMULATION

The simulation results of a powertrain model with clutch-to-clutch–type transmission are depicted in

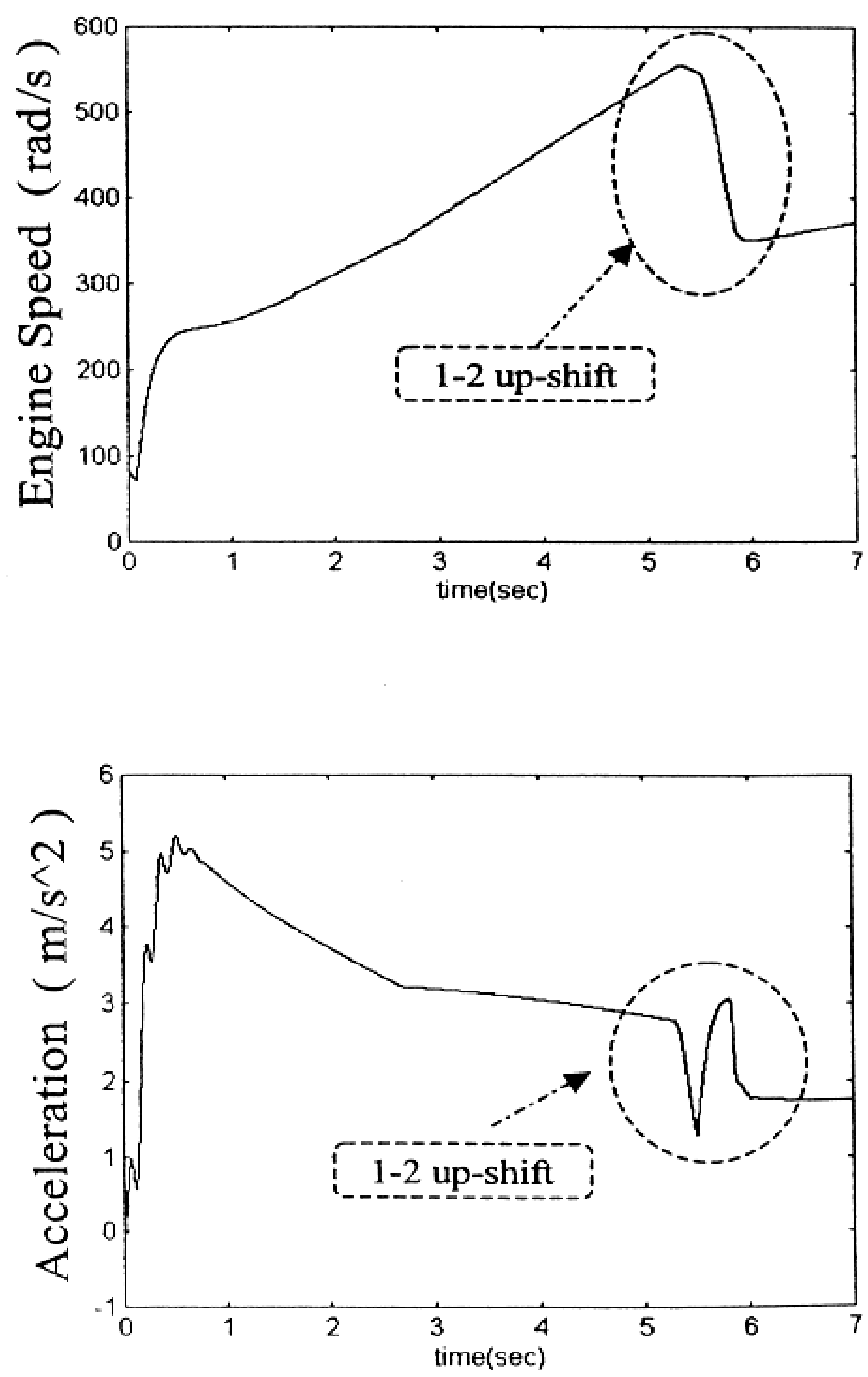


Figure 11 Start-up simulation with a clutch-to-clutch A/T.

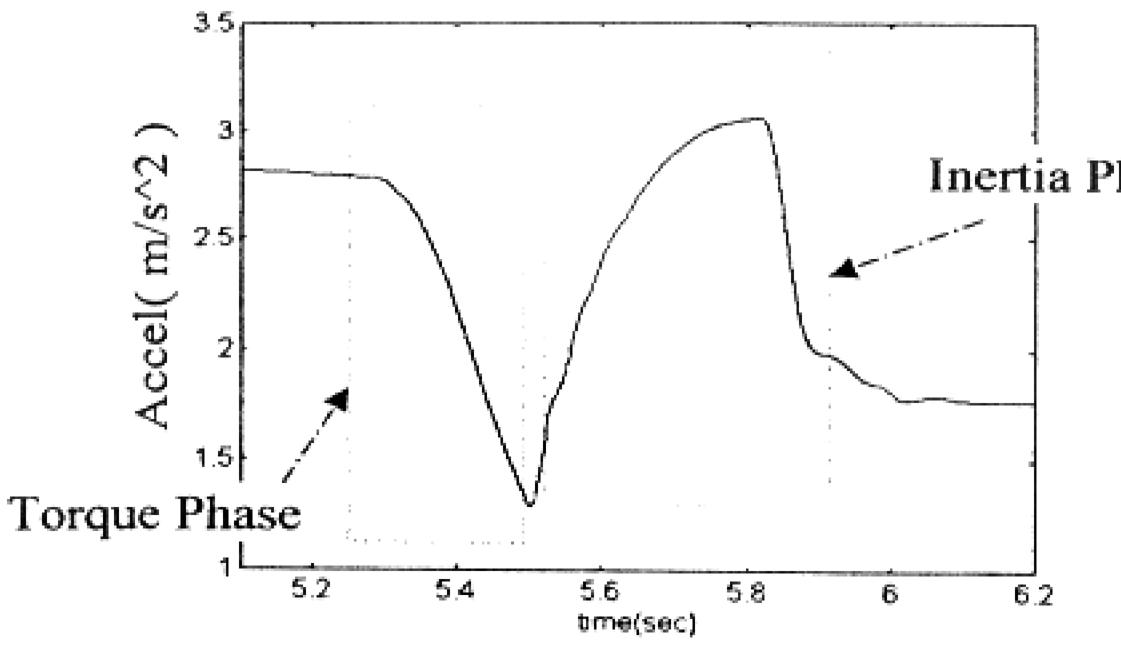


Figure 12 Acceleration during 1 → 2 upshift.

Figures 11–13. The air–fuel ratio and the spark advance are assumed to be fixed at 14.64 and 30 degrees, respectively. The engine speed and vehicle acceleration during startup operation are shown in Figure 11. The abrupt change of the engine speed and acceleration during 1 → 2 upshift is shown between 5 and 6 s. Figure 12 shows the details of the acceleration changes in torque and inertia phases during 1 → 2 upshift. The dynamics of the powertrain are relatively fast during 1 → 2 upshift. The simulation results agree well with the experimental results [10,14].

The profile of the output shaft torque during the upshift is the same as Figure 12. The upshifting begins with the decrease of output shaft torque (torque phase) as shown in Figure 12, followed by the torque bounce in inertia phase. The decrease of output shaft torque in torque phase is due to the second gear ratio being lower than the first gear ratio. During the inertia phase, if the torque increase is too large, it will cause an undesirable shock to the passenger, like a torque drop. On the other hand, if the torque increase during the inertia phase is not sufficient, then the total shift time gets longer. If the shifting time is too long, it might deteriorate the life of clutch components.

Thus, it is concluded that there are two objectives to control the powertrain during power-on upshift. One is the smooth acceleration (and jerk) transition to enhance the ride comfort. The other is the minimization of the clutch energy dissipation to increase 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 shift is computed as

$$E_{ci} = \int_{t_i}^{t_f} |T_{ci} \cdot \Delta\omega_{ci}| \, d\tau \qquad (5)$$

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

Figure 13 shows the open-loop characteristics of the jerk motion during 1 → 2 upshift. Its magnitude is big at 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. These sharp jerk motions and oscillations are undesirable. Figure 14 demonstrates the results of the closed-loop system with control law (4). Figure 15 depicts the vehicle acceleration which is related to the output shaft torque of transmission. With closed-loop control, the magnitudes of vehicle acceleration and jerk are much reduced. During the inertia phase, the second clutch slip speed, $\Delta\omega_{c2}$, is reduced to zero and does not increase the output shaft torque much. This is actually achieved by decreasing the engine torque. However, the price for improving the shift quality is the increase of energy dissipation due to the longer shift time.

As demonstrated above, the powertrain simulation model constructed in this article can be usefully used

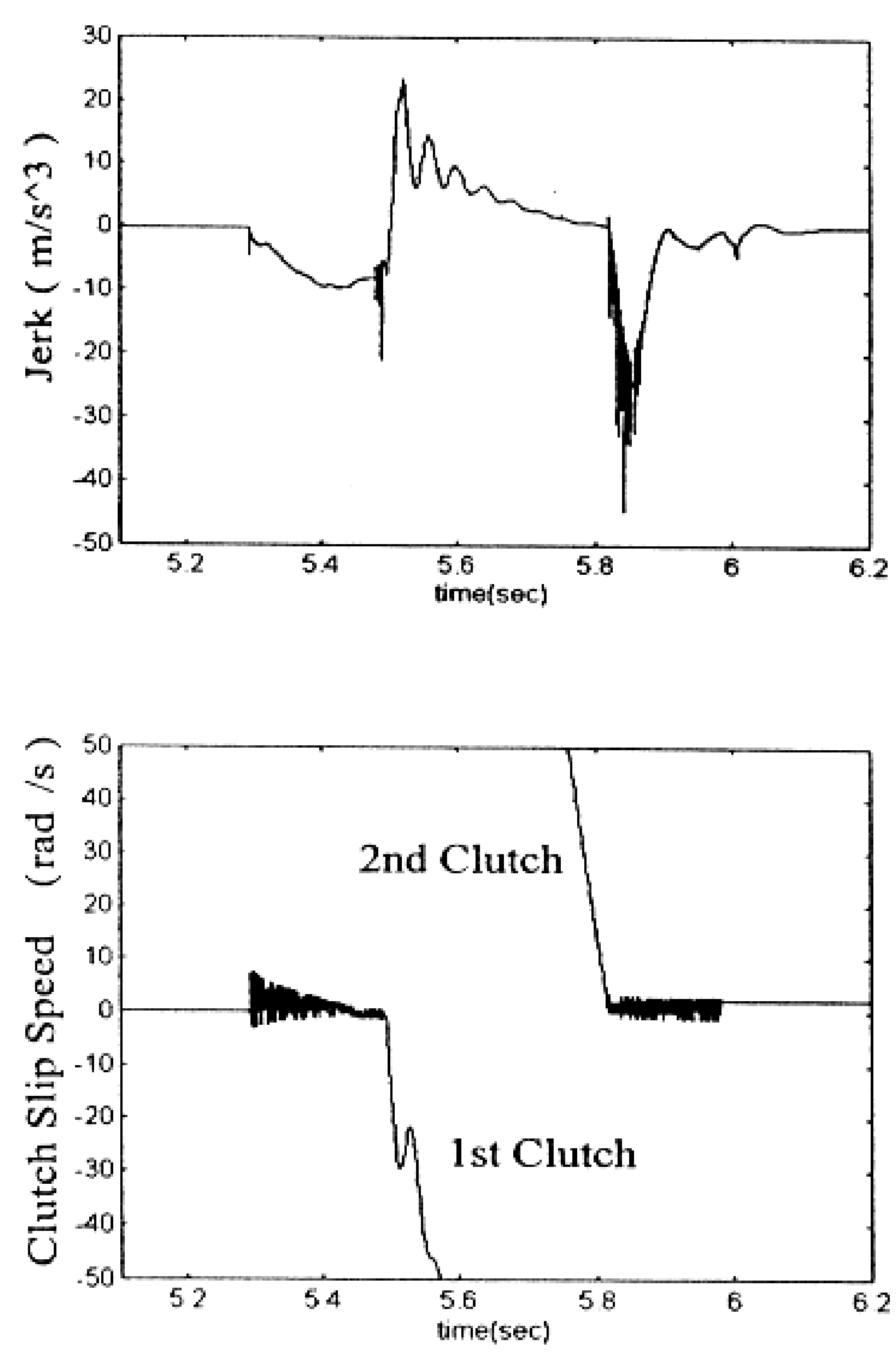


Figure 13 Jerk and clutch slip speeds during 1 → 2 upshift.

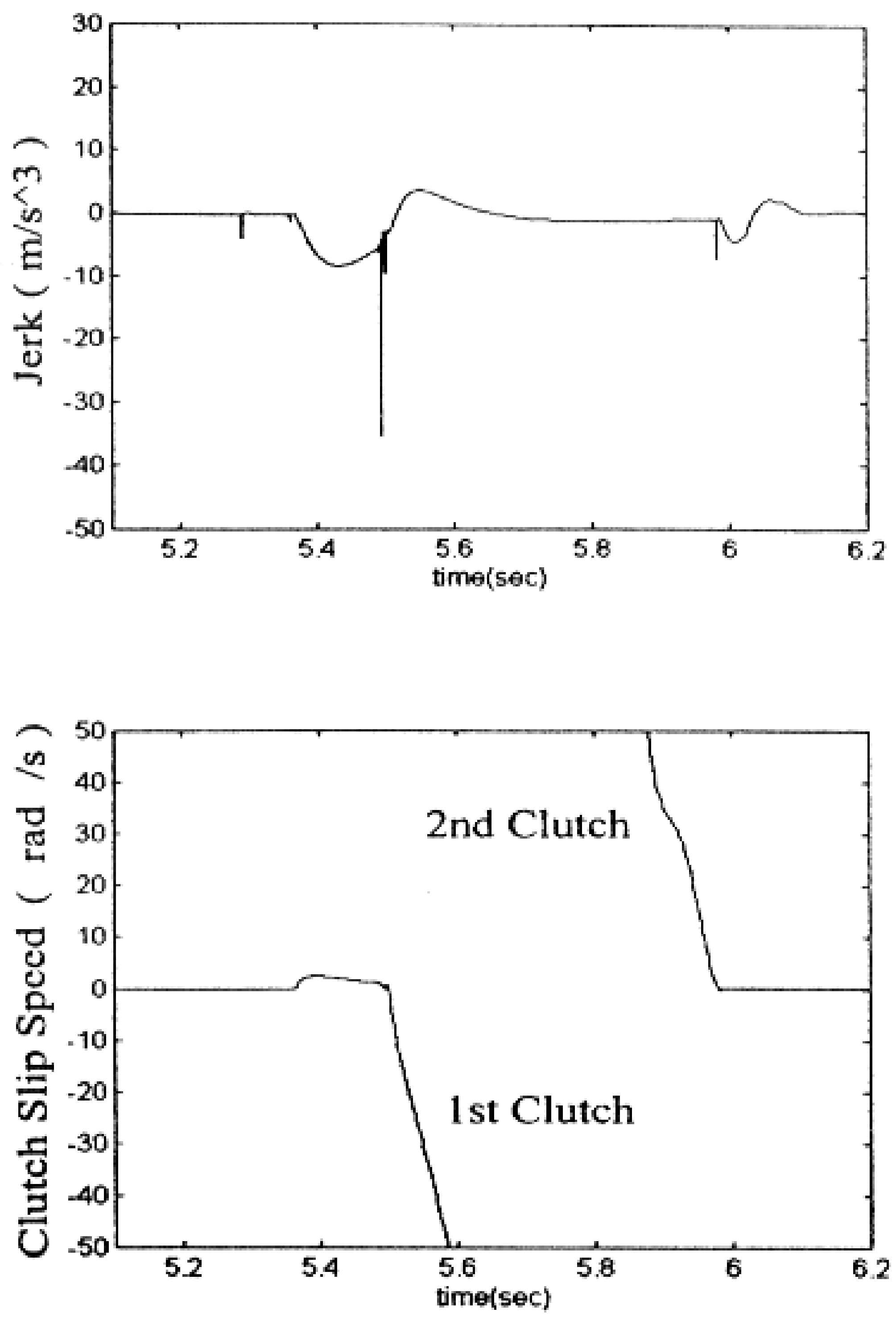


Figure 14 Jerk and clutch slip speeds during $1 \rightarrow 2$ upshift (closed-loop control).

for refining the strategies of the integrated engine and AT control. Finally, the engine module itself can be separately simulated by setting the torque of converter pump to zero. Hence, various control strategies can be developed for the fuel injection system, the idle speed control valve, etc.

CONCLUSIONS

In this article, a simulation model for evaluating the performance of powertrain and its controller has been constructed in the MATLAB/SIMULINK environment.

The approach used here reduces the model complexity by using the modular programming approach, and improves the visual effectiveness by the graphical user interface. The structure of the model is simple owing to the hierarchical order. The model provides easy testing of the performance of powertrain components. Individual subsystem modules are effectively reused once they are constructed as objects. Furthermore, the powertrain model is easily extended to a larger simulation model for the whole vehicle dynamics. The simulation model can be also used for real-time control and hardware-in-loop simulation in the dSPACE environment [15].

APPENDIX: DYNAMICS EQUATIONS

Engine Dynamics

$$\dot{m}_a = \dot{m}_{ai} - \dot{m}_{ao}$$

$$\dot{m}_{ai} = MAX \cdot TC \cdot PRI,\ \dot{m}_{ai} = c_T \cdot \eta_{vol} \cdot m_a \cdot \omega_e$$

$$I_e \cdot \dot{\omega}_e = T_i - T_{f/p} - T_p - T_{load}$$

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

where

$\dot{m}_a$ mass flow rate of air in the intake manifold (kg/s)
$\dot{m}_{ai}$ mass flow rate of air entering the manifold (kg/s)
$\dot{m}_{ao}$ mass flow rate of air entering the combustion chamber (kg/s)
MAX maximum flow rate through throttle plate (kg/s)
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
η_{vol} cylinder volumetric efficiency
ω_e engine speed (rpm)
I_e engine inertia (kg m^2)
$\#_{cyl}$ number of cylinder
TF torque function (Nm/g)
Δt_{IS} time delay, intake to spark (s)
Δt_{ST} time delay, spark to torque (s)

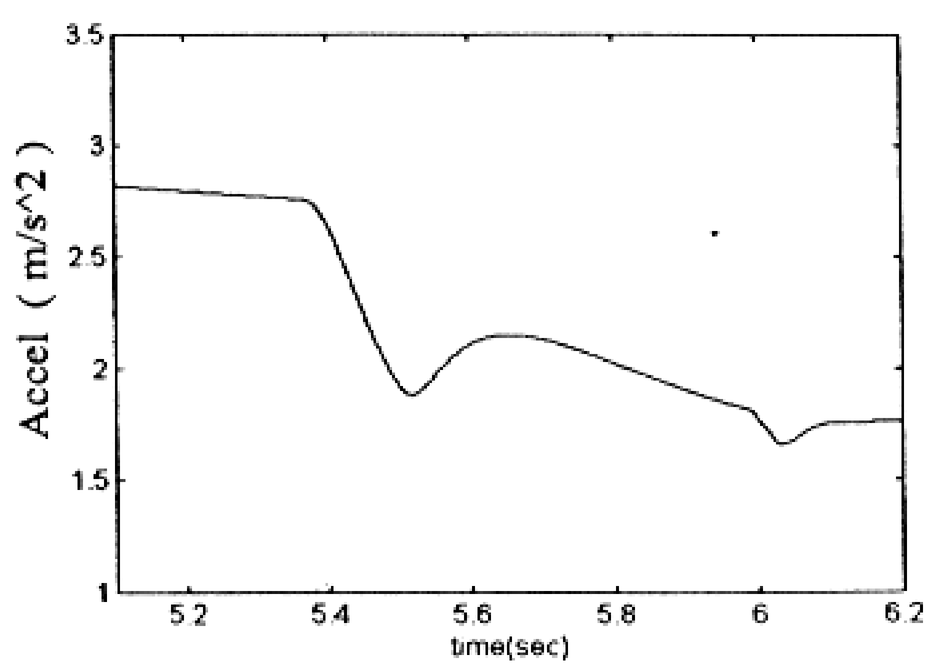


Figure 15 Vehicle acceleration during $1 \rightarrow 2$ upshift (closed-loop control).

Automatic-Transmission Dynamics

▲ State equation in first gear: $I_{f1}\dot{\omega}_t = T_t - R_1 R_d T_s$, where

$$I_{f1} = I_t + I_{si} + R_1^2 I_{cr} + \frac{R_1^2}{R_2^2} I_{ci}$$

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)

▲ State equation in 1 → 2 upshift:
<SPRAG transmission>

Torque phase: $I_{f1}\dot{\omega}_t = T_t - R_1 R_d T_s - \left(1 - \frac{R_1}{R_2}\right) T_{c2}$

Inertia phase: $I_t\dot{\omega}_t = T_t - T_{c2}$, $I_{cr12}\dot{\omega}_{cr} = \frac{T_{c2}}{R_2} - R_d T_s$

<Clutch-to-clutch transmission>

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

where $I_{cr12} = I_{cr} + \frac{I_{si}}{R_1^2} + \frac{I_{ci}}{R_2^2}$, ω_{cr} reaction carrier speed (rpm).

▲ State equation in second gear:
<SPRAG transmission> $I_{f2}\dot{\omega}_t = T_t - R_2 R_d T_s$
<Clutch-to-clutch transmission> $I_{f2}\dot{\omega}_t = T_t - \frac{R_1}{R_2} T_{c1} - R_2 R_d T_s$

where $I_{f2} = I_t + I_{ci} + R_2^2 I_{cr} + \frac{R_2^2}{R_1^2} I_{si}$

▲ Clutch torque: $T_{ci} = AR_i \cdot \mu_i \cdot P_{ci} \cdot \mathrm{sgn}\,(\Delta\omega_{ci})$, $i = 1, 2$
where AR_i = total clutch area of i gear × effective radius; $\mu_i = 0.1316 + 0.0001748|\Delta\omega_{ci}|$, $\Delta\omega_{ci} = \omega_t - \frac{\omega_{cr}}{R_i}$

Driveline Dynamics

$\dot{T}_s = K_s(R_d\omega_{cr} - \omega_{wf})$, $I_{wf}\dot{\omega}_{wf} = T_s - h_f F_{tf} - T_{rf}$; $F_{tf} = K_i \cdot i_d$; $M \cdot \dot{V} = F_{tf} - F_{tr} - F_a$, where

K_s axle shaft stiffness (both sides combined)
ω_{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 $h_f\omega_{wf} > 0.5$ m/s: $i_d = 1 - \frac{V}{h_f\omega_{wf}}$. For $h_f\omega_{wf} \le 0.5$ m/s: $i_d = \frac{h_f\omega_{wf} - V}{0.5\ \mathrm{sgn}\,(h_f\omega_{wf})}$.

REFERENCES

[1] S. Kuroyanagi, T. Ono, T. Wada, and B. Cohen, An engine and transmission control system with new 16-bit single chip microcomputer, SAE paper, No. 910082, 1991.
[2] N. Tomisawa and D. L. Davis, Trends in powertrain integrated control and development of optimum microcomputers, SAE paper, No. 900238, 1990.
[3] J. R. Wagner and J. S. Furry, A real time simulation environment for the verification of automotive electronic controller software, Int J Vehicle Design 13 (1992), 365–377.
[4] M. W. Sayers and C. Mink, A simulation graphical user interface for vehicle dynamics models, SAE paper, No. 950169, 1995, pp. 15–24.
[5] C. R. Ciesla and M. J. Jennings, A modular approach to powertrain modeling and shift quality analysis, SAE paper, No. 950419, 1995, pp. 139–147.
[6] R. W. Weeks and J. J. Moskwa, Automotive engine modelling for real-time control using MATLAB/SIMULINK, SAE paper, No. 950417, 1995, pp. 123–137.
[7] The MathWorks, Inc., Using SIMULINK, 1997.
[8] J. Rumbaugh, M. Blaha, W. Premerlani, F. Eddy, and W. Lorensem, Object-oriented modeling and design, Prentice Hall, Englewood Cliffs, NJ, 1991.
[9] J. J. Moskwa, Automotive engine modeling for real time control, Ph.D. thesis, Department of Mechanical Engineering, MIT, 1988.
[10] D. Cho, Nonlinear control methods for automotive powertrain systems, Ph.D. thesis, Department of Mechanical Engineering, MIT, 1987.
[11] J. K. Runde, Modeling and control of an automatic transmission, M.S. thesis, Department of Mechanical Engineering, MIT, 1986.
[12] C. H. Pan and J. J. Moskwa, Dynamic modeling and

simulation of the Ford AOD automobile transmission, SAE paper, No. 950899, 1995, pp. 1689–1697.

[13] H. Yamaguchi, Y. Narita, H. Takahashi, and Y. Katou, Automatic transmission shift schedule control using fuzzy logic, SAE paper, No. 930674, 1993, pp. 617–627.

[14] B. Martin and C. J. Redinger, 42LE electronic four-speed automatic transaxle, SAE paper, No. 930671, 1993, pp. 61–83.

[15] H. Hanselmann, Automotive control: From concept to experiment to product, IEEE Int Symp Computer-Aided Control System Design, Dearborn, MI, September 15–18, 1996.

BIOGRAPHIES

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