Co-Simulation Study of Coordinated Engine Control Focusing on Tracked Vehicle Shift Quality

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***Abstract*—Coordinated engine control is an essential method to improve automotive shift quality, which is also an important aspect of performance for modern tracked vehicle. In order to study the possibility of using coordinated engine control in tracked vehicle, a co-simulation study was conducted by coupling highly detailed and accurate sub-models of the various powertrain components, each created using the most appropriate simulation package. The simulation results show that using coordinated engine control by reducing engine fuel flow during shift can effectively improve tracked vehicle shift quality without sacrificing overall vehicle performance.**

***Index Terms*—coordination engine control, co-simulation, tracked vehicle, shift quality**

I. Introduction

The introduction of hydraulic automatic transmissions into tracked vehicle has made driving easier in the last few decades. [1] However, this type of transmission has also caused some problems. Additional control problems involved in automating shifts are a few of the problems encountered. [2] The objectives of automatic transmission control are to perform various transmission related control functions in an optimal way, so as to achieve satisfactory gear shifts with reduced shift shock, and to improve the efficiency of automatic transmissions efficiency and the overall fuel economy. [3] - [6] Among these goals, better shift quality is a hot issue associated with shifting automatic transmissions. Due to the discrete gear ratios of automatic transmissions, engine speed, engine and driveshaft torques undergo large changes during shifts, and hence ensuring smooth transitions is a demanding task.[7][8]

Electronically controlled transmissions which have become more common in recent years enable integration of the control of the engine and the transmissions to achieve optimal performance of the power train [9]. From a systems perspective, the engine and transmission are not stand-alone units. They have to be better coordinated to make the vehicle perform well. Coordinated engine control, specifically, engine torque control, is one of the key ways to shift smoothly and acquire a good shift quality.[10][11] The objective of engine torque control when shifting up under load is to reduce the energy dissipated in the friction elements during shifting. This can increase the lift of the friction elements by shortening the clutch slip time and has been widely used in wheeled vehicle with gasoline engine.[12] - [14] But, for turbocharged engine normally equipped in tracked vehicle, engine torque control is still a novel technology and needs to research. One apparent problem in implementing engine torque control in tracked vehicle is using what to control the engine torque, when there is no throttle and injection timing could also be uncontrollable. Fuel flow control may be the most available way to control engine torque, since no additional actuators are required. However, it is a difficult task to coordinate shift control together with air-to-fuel ratio control which is a complex control problem in itself.

Integrated powertrain modeling is important in the design of powertrain control strategies. For one thing, isolated component models may not yield adequate information to deal with system-level interactive issues, especially when it comes to transient behavior. In addition, massive amounts of expensive experimental work will be required for developing control strategy. There are three types of integrated powertrain models in the literature in terms of the details of engine dynamics included: 1)the first type focused on automatic transmission modeling, and assumed engine torque as input without including details of engine modeling except the engine crankshaft inertial effect [15][16]; 2) the second type of powertrain model incorporated engine torque-engine speed maps with transmission models [17]; 3) the third type of powertrain model integrated engine dynamics with transmission dynamics [18]. Most of them were modeled by using Matlab/Simulink with high accuracy highly relied on experiment data. Moreover, even the most complicated engine model mentioned in these papers were not completely competent to the task of coordinated engine control, since the in-cylinder phenomenon which is a vital monitoring parameter during the process could not be predicted. Recent computing speed improvements are beginning to enable the use of co-simulation [19][20] to couple highly

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detailed and accurate submodels, each created using the most appropriate available simulation package, which provides a practical way to overcome the problem encountered by traditional Simulink-based integrated powertrain modeling. This paper describes such a system model using GT-Power to model the engine, AMESim to model torque converter, transmission and vehicle dynamics, and Matlab/Simulink to model the engine controllers and transfer data between GT-Power and AMESim. The objective of this study is to improve the shift quality in tracked vehicle. To meet this objective, coordinated engine control is proposed in this study.

II. Model Description

Detailed models of the various Powertrain sub-systems were created and coupled in co-simulation. Fig. 1 shows the co-simulation architecture, focusing on the key parameters of each module. The engine modeled in GT-Power transmitted torque to the torque converter, transmission and vehicle dynamics built in AMEsim, receiving speed responded by AMEsim as an initial condition for next step's calculation. Matlab/Simulink acted as a bridge between those two kinds of models and provided the central control of information passed between the submodels. In order to proceed, a control system model was created in Matlab/Simulink, and connected to the GT-Power engine model and the AMEsim driveline model. By acquiring signals as engine speed and vehicle speed, controllers calculate the required engine fuel flow and make shift decisions. For brevity, a detailed discussion of model features will not be given, but schematic diagrams of the engine and other powertrain systems are shown in Fig. 2 and Fig. 3 to give the reader some appreciation for the level of modeling detail.

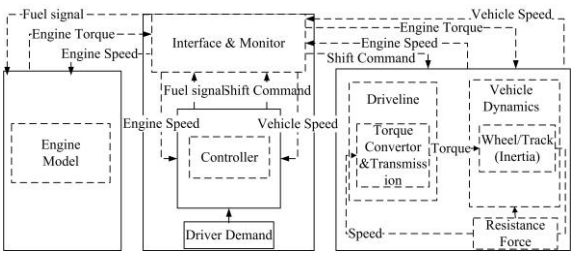

Figure 1. System architecture for co-simulation.

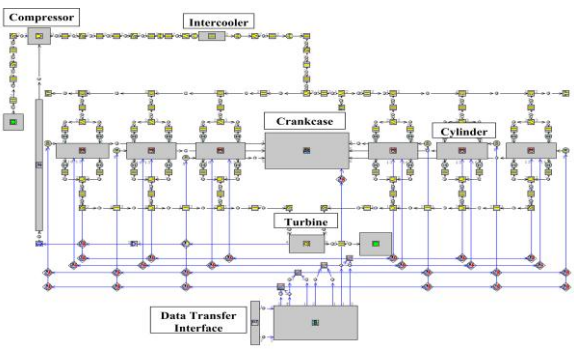

Figure 2. GT-power engine model.

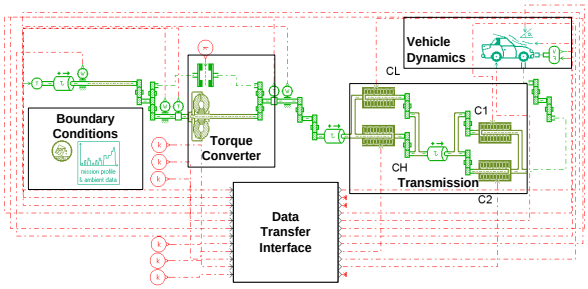

Figure 3. Driveline and vehicle dynamics model.

A. Engine Model

The engine model represents the DEUTZ turbocharged diesel engine in a tracked vehicle. The model was calibrated against dynamometer test data for both part-load and full-load accuracy. A summary of engine features is given in Table I.

TABLE I. Papers Basic Engine Characteristics

DEUTZ Turbocharged Diesel Engine	
Number of cylinders	6 90 °Vee-angle
Bore	132mm
Stroke	145mm
Compression ratio	17:1
Maximum engine speed	2100 r/min
Number of valves	4 per cylinder
Injection system	In-line pump
Charge cooling	Liquid-cooled

B. Driveline and Vehicle Dynamics Model

The tracked vehicle driveline modeling is mainly comprised of torque converter, transmission, and other shafts and gears. Simplified vehicle dynamics is included to describe the motion of vehicle in the longitudinal and heave direction. A summary of driveline and vehicle features is given in Table Ⅱ.

TABLE II. Basic Driveline and Vehicle Characteristics

Vehicle		Transmission	
Parameters	Value	Gear	Ratio
Mass of vehicle	16.8 tonne	1st	0.5
Tire radius	0.345m	2nd	0.8
Vehicle active area in aerodynamic drag	10 m2	3rd	1.33
Final driving transmission	5.303	4th	2.13
coefficient of rolling friction	0.01		

Torque converter is located between the engine and transmission and utilizes the kinetic energy of the fluid in the converter to transfer power. The torque converter model takes into account following aspects: impeller, turbine, stator and fluid inertias; fluid frictions; shocks losses; influence of system geometry on fluid speeds and torques.

One kind of hydraulic automatic transmission with three fixed axes is established by considering the detailed clutch dynamic. A friction model, as shown in Fig. 4, is used to simulate multi-disc clutch encountered typically in automatic gearboxes. It models a global friction which is the sum of a dry part (Coulomb and stiction friction)

and a viscous part (drag torque). Dry friction is modeled with a realistic Reset-Integrator model that considers a static friction coefficient and a dynamic one that can vary with velocity (Stribeck effect). Moreover an equivalent viscous damping is considered to take into account the oil film effects by adjusting it versus the mean oil thickness and a delay in the friction applied torque due to this oil effect (oil should be removed to have a dry friction) has been introduced.

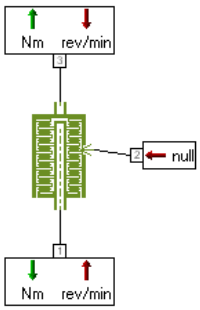

Figure 4. Multi-disc clutch friction model

Vehicle dynamics model is a simplified model in which the suspension or vibration is not considered and vehicle mass is assumed lumped at the center of gravity. The input to the system is the road profile. The toque from the driveline is applied to the wheel hub, and the available traction force to accelerate the vehicle is calculated by equation.

$$F_j = \frac{T_t \eta}{r_z} \tag{1}$$

where T_t is the driver wheel axle torque and r_z is the working radius, η represents track efficiency, which is difficult to calculate and normally described by empirical equation as follows:

$$\eta = 0.95 - 0.0017v \tag{2}$$

where v is the vehicle's line speed.

The summation of the rolling resistance, gravity effects along the slope and aerodynamic drag constitutes the propulsion load for the vehicle.

C. Controller

A control system model including governor control and shift control was created. All-speed governor is built in this vehicle, which means the governor regulates speed over the whole engine operating range. The desired engine speed (as dictated by the position of the accelerator pedal in vehicles) and the actual engine speed are the two basic inputs to the governor control loop. Any difference in the values of these two signals results in displacement of the governor sensing element; the latter alters the fuel pump rack position, therefore the injected fuel quantity and engine torque in order to establish a new equilibrium between engine and resistance at a different or even the same rotational speed. The vehicle shift schedule is based on vehicle speed and accelerator pedal position, as shown in Fig. 5.

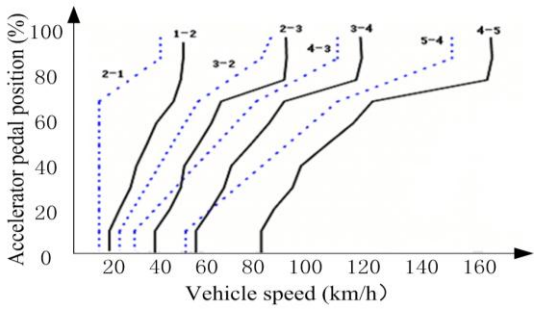

Figure 5. Shift schedule diagram.

III. Model Verification

To validate the predictive capability of the simulation, vehicle simulation tests were performed using prescribed acceleration pedal transient maneuvers. The control parameters actually used in the vehicle were inserted into the simulation. Data logged during drive cycle testing of the vehicle was examined to identify a severe transient condition for the study.

The simulation closely matched measured transmission output torque and fuel consumption, as shown in Fig. 6. This verifies that the engine, vehicle and transmission faithfully predict overall vehicle behavior.

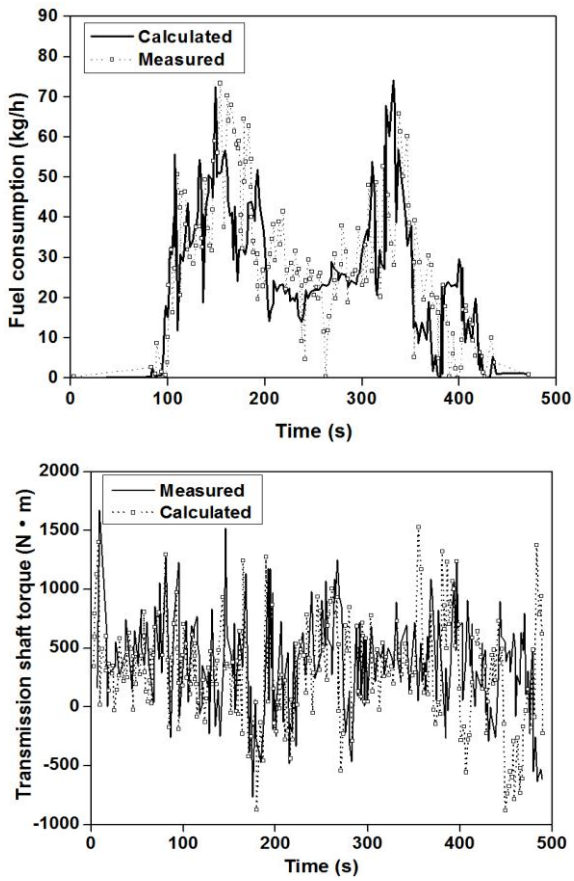

Figure 6. Integrated vehicle model verification

IV. Coordinated Engine Control

A. 1st to 2nd Gear Upshift

In this simulation, the vehicle was driven to accelerate on a level road. The 1st to 2nd gear upshift is firstly analyzed, when the A clutch is on-coming due to the control pressure increase and the B clutch is off-going, as shown in Fig. 7. The clutch capacity, through which the maximum possible torque is transmitted, is fixed, because this is related to the size and number of clutch plates, clutch plate size, maximum working hydraulic pressure, minimum friction coefficient and other physical properties and determined when it was designed. In addition, ideal oil pressure characteristic is imposed and engaging and disengaging timings for both oncoming clutch and offgoing clutch are fixed.

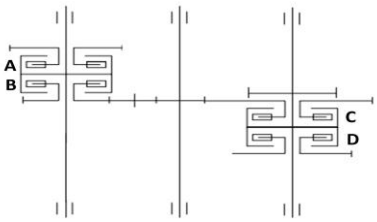

Figure 7. Schematic diagram of a transmission

In order to analyze the influence of engine's fuel flow on shift quality, a set of case studies are designed as follows:

- *Case1:* baseline without coordinated engine control during shift process;
- *Case2:* reduce the fuel supply to 20mg per cycle per cylinder(normally 60 mg) during shift process;
- *Case3:* reduce the fuel supply to 10mg per cycle per cylinder(normally 60 mg) during shift process;
- *Case4:* cut off engine's fuel supply during shift process;
- *Case5:* increase fuel flow in torque phase and reduce fuel supply in inertia phase;
- *Case6:* cut off fuel supply merely in inertia phase.

Two phenomena during vehicle shift process should be focused: 1) torque hole and 2) harshness of the lockup. The torque hole magnitude is the drop of transmission output torque during the torque phase. It causes a noticeable shift disturbance in the transmission output torque and affects shift quality although it is an integral result of the transmission design. The harshness of the lockup depends on the change in the torque transmitted by the oncoming friction element. The larger this change, the more abrupt the transition.

The simulation results display in Table Ⅲ and Fig. 8. show that shift duration decrease along with less dissipated power through implementing fuel flow reduction control during shift process, shift duration decrease along with less dissipated power. This is a result of smaller engine output torque and the transmission input shaft torque compared with those in the baseline. Meanwhile, engine speed and clutch active side speed began to decline in torque phase and declined faster than those in baseline, which results in faster engagement and shorter sliding friction stage duration. When the engine speed has been decelerated until achieving synchronization to the level of target gear, the sliding friction stage was ended and shift shock happened, then the friction torque declined rapidly to the torque transmitted. The more fuel the engine reduced, the faster the clutch active side decreased, thus, the earlier the engagement happened, and the smaller the shift shock was.

According to the governing characteristic of the diesel engine with all-speed governor, when the fuel reduced too significantly, the fuel flow would recover rapidly after shifting, according to the governing characteristic of the diesel engine with all-speed governor, which would result in the emergence of another torque hole, except the one caused by torque handover, as shown in Fig. 9. In order to avoid experiencing a power interruption, some methods should be used to limit the slope of fuel recovery in practical use.

Two phases can be identified in each shift-the torque phase and the inertia phase. The last two cases were designed to discuss individual-phase coordinated engine control. As shown in Fig. 9 and Table Ⅲ, the simulation results of case 5 and 6 resemble the results of case 1 and 3 respectively. The results also indicate that increasing fuel flow in torque phase which meant to reduce the torque hole didn't work well, since the shift duration is so short (about 1 s) that the transmission inlet torque cannot be increased timely and decreased arbitrarily. This is also partly due to the low sensitivity of turbocharged diesel engine during transient situation, namely, the poor dynamic response against sudden fuel increments caused by "turbocharger lag".

TABLE III. Main Shift Quality Results of Six Cases

Case number	Shift duration (s)	Inertia phase duration (s)	Dissipated power (J)	Minimum transmission shaft torque (Nm)	Maximum transmission shaft torque (Nm)
Case 1	0.928	0.706	11140	185	502
Case 2	0.820	0.598	7062	186	452
Case 3	0.753	0.531	5332	186	430
Case 4	0.699	0.477	4831	186	429
Case 5	1.038	0.816	8024	196	565
Case 6	0.767	0.545	6700	186	424

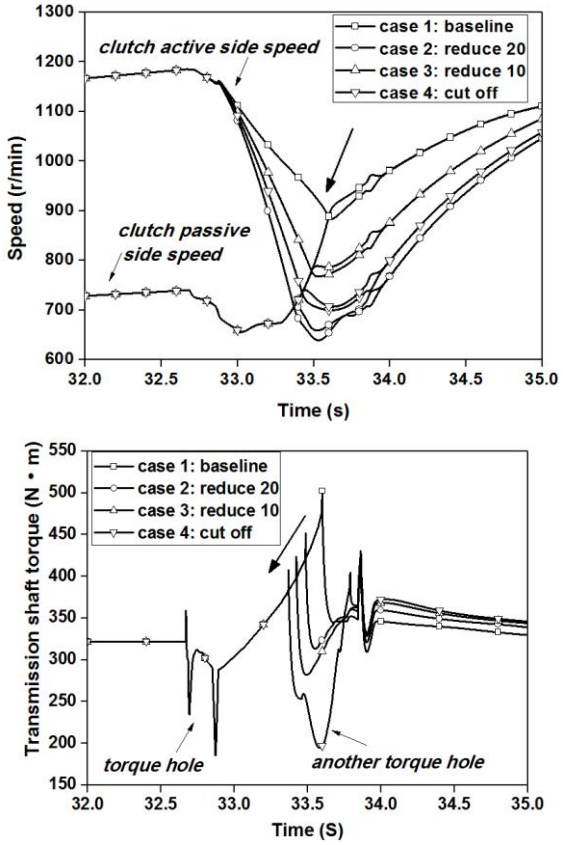

Figure 8. Simulation results during shift processes for case1-case4.

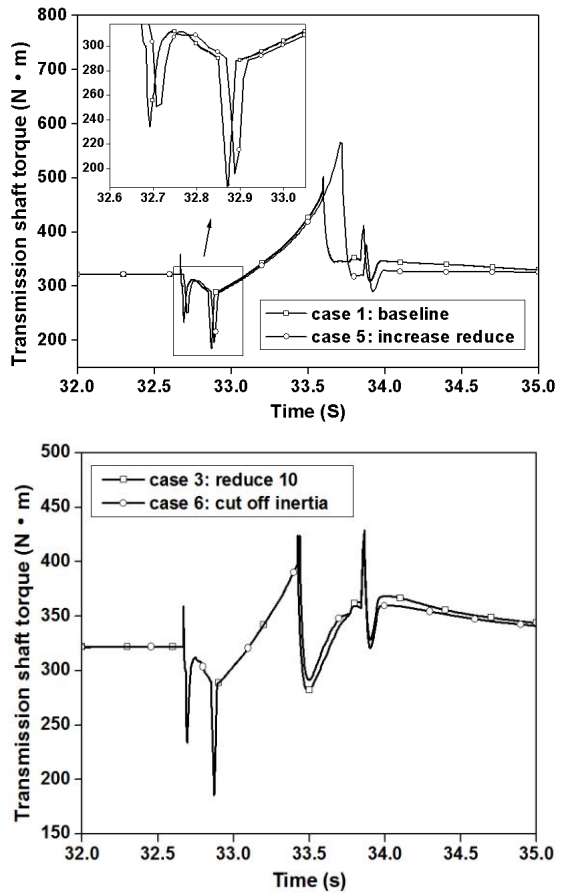

Figure 9. Simulation results of case5 and case6.

B. Vehicle Acceleration Process

Finally, in order to see the influence of coordinated engine control during shift process on vehicle's overall drivability, vehicle acceleration processes involved with 3 shift processes (0-1st upshift, 1st-2nd upshift, 2nd-3rd upshift) were simulated respectively by applying control strategy described in case 1 and case 6. As shown in Fig. 10, after shifting, the vehicle speed was interrupted, with coordinated control (case 6), the interruption was more serious, but the discrepancy between two cases was insignificant. On the other hand, the shift shocks during these three shifts were reduced through coordinated control respectively by about 10%, 20%, 30%. In sum, the fuel reduction during shifting affects the whole vehicle performance insignificantly but ameliorates the harshness of lockup.

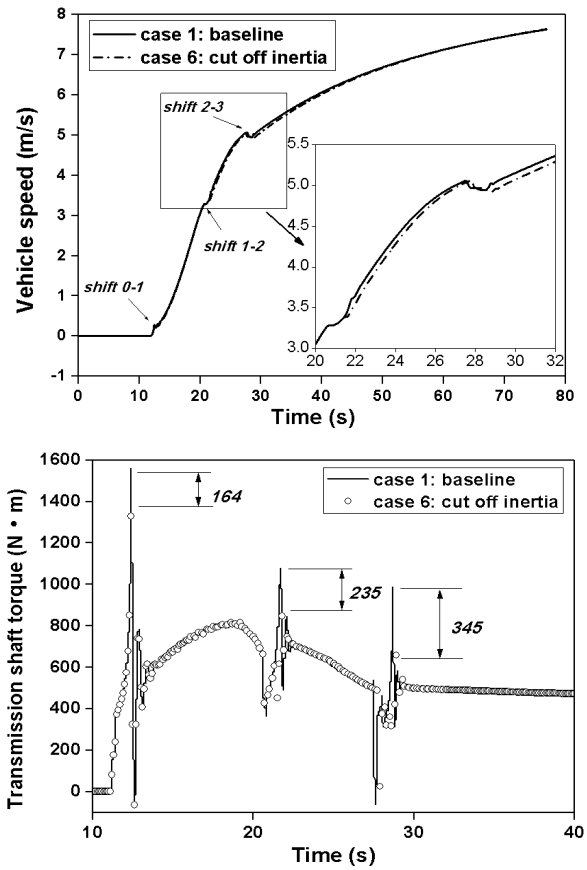

Figure 10. Effects of coordinated engine control during vehicle acceleration.

V. Conclusions

In this study, powertrain in tracked vehicle has been modeled by using co-simulation technology. Detailed models of the various powertrain sub-systems including the engine, the torque converter, the transmission with multidisc clutch, and the vehicle dynamics have been taken into account in order to understand the overall system. The torque converter dynamics and detailed clutch friction model have been brought in to analyze the powertrain performance of the vehicle and shift transient characteristics for the automatic transmission. To improve the shift quality, coordinated engine control,

namely, engine torque control by reducing fuel flow is suggested. Simulation results show that:

1) Using coordinated engine control during shift can effectively improve shift quality without sacrificing overall vehicle performance.

2) When the fuel reduced too significantly, another torque hole would emerge after shift process; the slope of fuel flow recovery should be limited.

3) Increasing fuel flow in torque phase has little impact on the magnitude of torque hole.

REFERENCES

[1] M. F. Brejcha, *Automatic Transmissions and Transaxles*, Third edition, Regnts Prentice Hall,1993.

[2] W. B. Wang, "*Dynamic powertrain system modeling and simulation with applications for diagnostics design and control*," Ph.D. dissertation, Dept. Mech. Eng, University of Wisconsin-Madison,Wisconsin, 2000.

[3] D. C. Kim, K. U. Kim, Y. J. Park, and J. Y. Huh, "Analysis of shifting performance of power shuttle transmission," *Journal of Terramechanics*, vol. 44, no. 1, pp. 111-122, 2007.

[4] K. Kusaka and Y. Ohkura, "A transmission control system for construction machinery," *SAE* 901557, 1990.

[5] A. Yoon, P. Khargonekar, and K. Hebbale, "Randomized algorithms for open-loop control of clutch-to-clutch transmissions," *Journal of Dynamic Systems, Measurement, and Contro*l, vol. 121, Sep. 1999.

[6] A. Haj-Fraj and F. Pfeiffer, "*A model based approach for the optimization of gear shifting in automatic transmissions*," *Int. J. of Vehicle Design*, vol. 28, pp. 171–188, 2002.

[7] A. Gibson and I. Kolmanovsky, "Modeling and analysis of engine torque modulation for shift quality improvement," *SAE 2006-01-1073*, 2006.

[8] T. D. Anna, K. Govindswamy, and F. Wolter, "Aspects of shift quality with emphasis on powertrain integration and vehicle sensitivity," *SAE 2005-01-2303*, 2005.

[9] A. G. Ulsoy, H. Peng, and M. Cakmakci, *Automotive Control Systems*, Cambridge University Press, 2012.

[10] G. L.Ohl, M. J. Prucka, F. S.Kanafani, *et al.*, "Development of powertrain coordination at DaimlerChrysler corporation," SAE 2004-01-0893, 2004.

[11] M. Ibmoto, H. Kuroiwa, T.i Minowa, *et al.*, "Development of smooth shift control system with output torque estimation," *SAE 950900*, 1995.

[12] D. Kim, H. Peng, S. Bai, and J. M. Maguire, "Control of integrated powertrain with electronic throttle and automatic transmission," *IEEE*. vol. 15, no. 3, May 2007.

[13] B. Mencher, H. Jessen, and L. Kaiser, *et al*., "Preparing for CARTRONIC-interface and new strategie for torque coordination and conversion in a spark ignition engine management system," *SAE 2001-01-0268*, 2001.

[14] Z. H. Wu and R. Pei, "Torque based spark ignition engine and powertrain modeling," in *Proc.7th World Congress on Intelligent and Automation*, Chongqing, China, 2008.

[15] C. H. Pan and J. J. Moskawa, "Dynamic modeling and simulation of the Ford AOD automobile transmission," *SAE 950899*, 1995.

[16] H. S. Jeong and K. I. Lee, "Shift transient torque control of a vehicle power transmission system," *FISITA T 945031*, 1994.

[17] K. Kono, H. Itoh, and S. Nakamura, "Torque converter clutch slip control system," *SAE 950672*, 1995.

[18] D. Assanis, W. Bryzik, and N. Chalhoub, "Integration and use of diesel engine, driveline and vehicle dynamics models for heavy duty truck simulation," *SAE 1999-01-0970*, 1999.

[19] Y. S. He, C. C. Lin, and A. Gangopadhyay, "Integrated simulation of the engine and control system of a turbocharged diesel engine," *SAE 2006-01-0439*, 2006.

[20] J. F. Sinnamon, "Co-simulation analysis of transient response and control for engines with variable valvetrains," *SAE 2007-01-128*, 2007.

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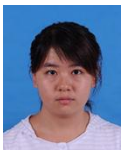

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