FUEL CONSUMPTION IMPROVEMENT OF VEHICLE
WITH CVT BY BOND GRAPHS

Katsuya Suzuki[a], Kazuhiro Tanaka[b], Sho Yamakawa[c], Fumio Shimizu[d], Masaki Fuchiwaki[e]

[a]- [e]Kyushu Institute of Technology

[a]kachandesu2002@yahoo.co.jp, [b]kazuhiro@mse.kyutech.ac.jp, [c]yamakawa@vortex.mse.kyutech.ac.jp, [d]shimizu@mse.kyutech.ac.jp, [e]futiwaki@mse.kyutech.ac.jp,

ABSTRACT

Dynamic system simulation is important for automotive development and improvement as well as for analyzing mechanism of the system. A vehicle power train with a continuously variable transmission (CVT) was modeled into a simple and a detailed Bond Graph model and the fuel economy and engine power performance was simulated by use of the models according to the Japanese 10 Mode cycle. As a result, it is verified that the proposed Bond Graph models represent the basic dynamic characteristics of vehicle power train system with CVT and it is useful to calculate the optimum gear change pattern for low fuel consumption.

Keywords: Bond Graphs, Vehicle, CVT, Power train

1. INTRODUCTION

Recently environmental problems such as global warming and air pollution become more serious year by year. Car exhaust gas emission of vehicle is one of the causes of the environmental problems, so that in order to solve the problems vehicle industries have been making activities, one of which is to improve fuel economy of car. A simulation method can play an important role to predict and determine parameters necessary to develop and improve a fuel economy, especially. Though simulations on fuel economy were studied in the past (Suzuki and Hosoi 2007; Yokoi 1979), there are few studies of the inverse problem such as running resistance is an input parameter for a vehicle power train system (Togai and Koso 2006).

The objectives of our study are to establish a system Bond Graph model of a power train system with the continuously variable transmission (CVT) (Carbone, Mangialardi, Bonsen, Tursi, and Veenhuizen 2007; Cho and Hedrick 1989; Kong and Parker 2008; Mantriota 2002; Srivastava and Haque 2008; Srivastava and Haque 2009; Srivastava and Haque 2009) as a study of the inverse problem in which running resistance is an input parameter for a vehicle power train system, and a simulation method calculating an optimal gear ratio of CVT suitable for running pattern of a vehicle.

In this study, a one-dimensional system Bond Graph of a vehicle power train including the CVT was established and an engine output power was simulated in the Bond Graph model according to The Japanese 10 Mode cycle. Then it is also clarified that it is possible to calculate the optimal gear ratio suitable for running pattern of a vehicle with CVT.

2. MODELING OF VEHICLE WITH THE CVT

Generally driving power performance is shown by vehicle maximum speed and arrival time for some distance as output variables against engine torque as an input variable. However, fuel consumption can be measured after running according to a legal running mode. In this case, an input variable is power for running resistance connected with vehicle acceleration and an output variable is engine power, from which the fuel consumption can be calculated. Namely, power-flow is quite reverse against the case of calculating driving power performance in a vehicle system model.

2.1. Analysis Object

In this paper, an analysis target is a power train system of vehicle including a belt type CVT with a gasoline engine as a driving source. Figure 1 shows a schematic illustration of the power train system, which consists of an engine, a clutch, a belt-type CVT, a final gear, an axle, and tires. This system can be expressed by Bond
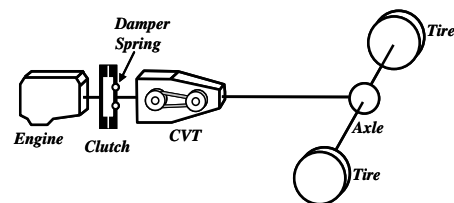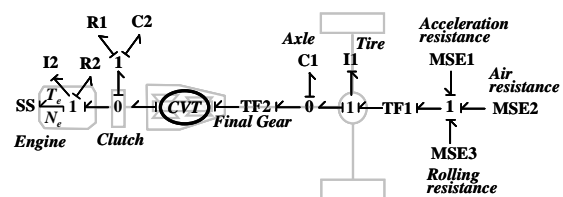

Figure 1: Schematic Illustration of Power Train

Figure 2: Bond Graph for Power Train

Graphs (Karnopp and Rosenberg 1970; Karnopp, Margolis, and Rosenberg 2006; Thoma 1990), shown in Figure 2 (Hrovat and Tobler 1991).

Each Bond Graph symbol indicates the respective following;

SE elements (effort source elements),
MSE1; the acceleration resistance R_s,
MSE2; the air resistance R_a,
MSE3; the rolling resistance R_r,
R elements (resistance),
R1; the frictional loss in the clutch,
R2; the frictional loss in the engine,
C elements (capacitor),
C1; the stiffness of the axle,
C2; the damper spring of the clutch,
I elements (inertia),
I1; the inertia of tires,
I2; the inertia of the engine,
TF elements (transformer),
TF1; transformer of the tire radius,
TF2; transformer of the final gear.
SS elements (flow and effort sensor in Bicausal Bond Graphs) (Gawthrop 1995)

Using this model, the engine revolution as well as engine torque are calculated as the output variables through the running resistance proportional to vehicle speed as the input variable. A way for calculating fuel consumption is described in Section 2.3.

2.2. Input Variable

The input variable in the simulation is the running resistance which is the sum of the following four resistances, the acceleration resistance, the air resistance, the rolling resistance and the gradient resistance, that are such function of the vehicle speed as expressed by Equation (1), (2), (3), and (4), respectively. Here, as described later, a running mode is supposed running on a level ground, the hill climbing resistance is always zero, 0 [N].

$$R_s = Ma \qquad (1)$$

$$R_a = C_D \frac{\rho}{2} A V^2 \qquad (2)$$

$$R_r = \mu M g \ (V > 0) \qquad (3)$$

$$R_c = Mg \sin \theta \qquad (4)$$

Here, A, a, C_D, g, M, V, μ, θ, and ρ indicate vehicle frontal projected area [m²], vehicle acceleration [m/s²], drag coefficient [-], acceleration of gravity [m/s²], vehicle mass [kg], vehicle speed [m/s], coefficient of rolling resistance [-], up-hill road gradient [deg], and air density [kg/m³], respectively.

In a simulation of forward power-flow direction, the engine power is delivered to the running resistance (the acceleration resistance, the air resistance and the rolling resistance), the frictional loss dissipated in
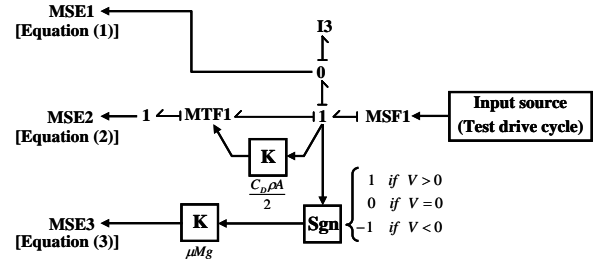

Figure 3: Input Variables

components and the inertial power. However, in a simulation of reverse power-flow direction, the running resistance (the acceleration resistance, the air resistance and the rolling resistance) calculated in proportional to each running mode is given to the system as an input variable and reverse calculation through the power train system including a CVT component causes the output power transmitted to the engine finally.

Figure 3 shows the input variables for the system. Each input variable shown by MSE-element is defined as the respectively corresponding Equation (1), (2) and (3). MSF1-element shows flow source (the 10 Mode cycle). In SE3-element, the rolling resistance is active only at $V > 0$. In this model the friction resistance in the clutch and the power lost in the engine are taken into consideration, so that the engine output power and fuel economy can be calculated more precisely.

2.3. Method of Calculating Fuel Economy

Figure 4 shows experimental data for the representative performance curves of vehicle engine. Fuel consumption is calculated using specific fuel consumption (*SFC* [g/(PS*h)]) which is expressed by broken lines in Figure 4. The chain line, the solid line with an arrow and WOT in the Figure indicate the optimum operating line for engine, the constant horse power line and the torque line on condition that the engine throttle valve is fully opened. In order to calculate easily the digital value of *SFC* from Figure 4, Figure 5 as a new diagram of *SFC* is made by the least-square method.

Equations (5), (6), (7), and (8) are used to calculate the sum of fuel consumption (*TFC* [*l*]) and the average fuel consumption (*AFC* [km/*l*]). In Equation (5), an engine output power (*P* [PS]) is calculated as the product of engine torque (T_e [Nm]) and engine revolution (N_e [rpm]). Fuel consumption (*FC* [*l*/s]) is calculated by Equation (6), in which unit of *SFC* is changed from [g/(PS*h)] to [*l*/(PS*s)] using density of gasoline, 0.783 [g/cm³] at 15 [°C]. To integrate *FC* for vehicle running time leads *TFC* in Equation (7). Finally *AFC* is given in Equation (8), where *SM* ([km]) indicates sum of millage for vehicle running time.

$$P = T_e \times N_e \times 1.42 \times 10^{-4} \qquad (5)$$

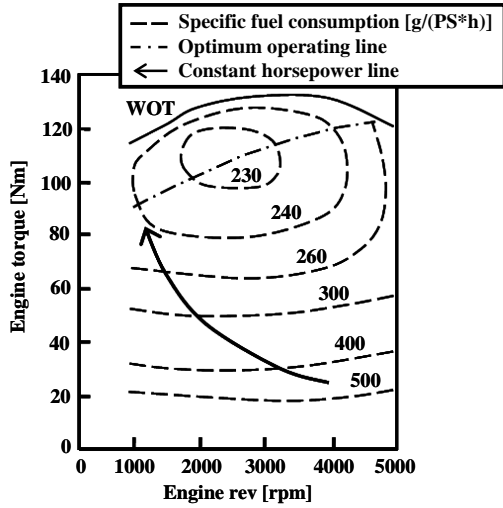$$FC = [((SFC \div 3600) \div 0.783) \times 10^{-3}] \times P \qquad (6)$$

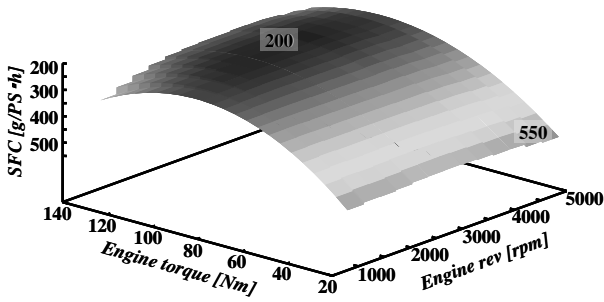Figure 4: Representative Engine Performance Curves (Experimental Data)

Figure 5: Specific Fuel Consumption (Approximate Value)

$$TFC = \int FC\, dt \tag{7}$$

$$AFC = SM \div TFC \tag{8}$$

The 20-Sim software is used for Bond graph modeling and simulation of vehicle fuel consumption with a CVT component in the present study.

2.4. How to Search for Optimum Gear Ratio in the CVT

A continuously variable transmission (CVT) is a new component expected to improve fuel economy. Especially in vehicle transmissions, the CVT is well suited for more fuel-efficient driving in any driving mode compared with traditional gear transmissions with fixed gear ratio, because it is possible to change the gear ratio continuously and smoothly, to reduce the shock of gear change and to operate at high efficiency domain of engine revolution.

A CVT component, shown in Figure 6, is modeled into two kind systems, CVT-1 in Figure 7 as a simple model and CVT-2 in Figure 9 as a detailed model. The difference between the two models is also studied after simulation of dynamic characteristics.

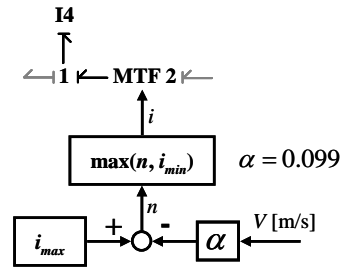Figure 6: CVT (Daihatsu Motor Co. 2006)
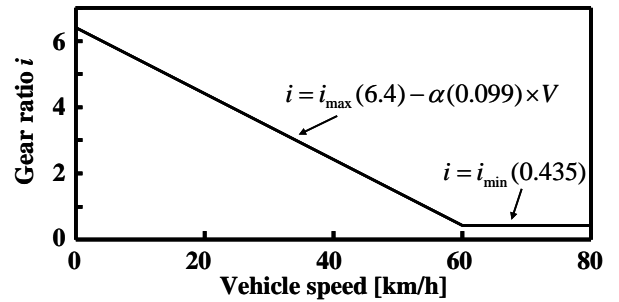

Figure 7: Bond Graph of CVT-1

Figure 8: Assumed Gear Ratio of CVT-1

Figure 7 shows the Bond Graph model of CVT-1, where one inertial mass consisting of two pulleys and a belt is expressed by one I-element, I4, and gear ratio by a modulated transformer, MTF2-element, and its variable coefficient is given as CVT gear ratio i shown in Figure 8.

In the present case, the coefficient function of MTF2-element is defined to be linearly proportional to vehicle speed. In Figure 8, α indicates the proportional coefficient and is defined as 0.099. The gear ratio i is defined as a function $\max(n, i_{\min})$ and is calculated by Equations (9) and (10), where i, n, V, $i_{\max}$, and $i_{\min}$ indicate gear ratio, variable, vehicle speed [km/h], maximum and minimum gear ratio, respectively.

$$n = i_{\max} - \alpha \times V \tag{9}$$
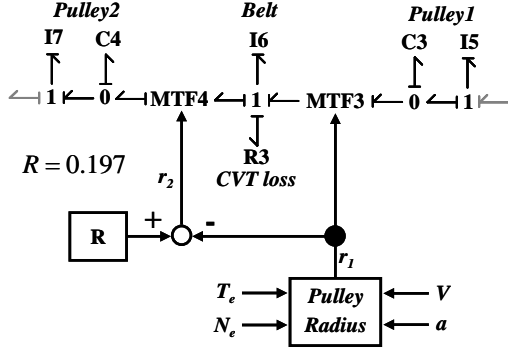
$$i = \max(n, i_{\min}) \tag{10}$$

Figure 9: Bond Graph of CVT-2

Figure 9 shows Bond Graph of CVT-2, where the pulley and the belt are expressed independent in the model. Inertial effects of Pulley1 and Pulley2 are expressed by I5- and I7-element, respectively. Stiffness between Pulley1 and the belt and stiffness between Pulley2 and the belt are expressed by C3- and C4-elements, respectively. Belt transforms power between rectilinear motion and rotational motion, and its variable coefficient is given by r, which is a pulley radius from the center of pulley shaft to the contact point at the belt and the pulley. Because r is variable according to driving condition in CVT, the belt is represented by MTF-element. The inertial effect of the mass of belt is represented by I6-element and the frictional power loss between the belt and the pulleys by R3-element.

Gear ratio is decided as to reduce the torque difference between a point A on the constant horse power line and a point B on the optimum operating line (Figure 10) (Pfiffner, Guzzella, and Onder 2003; Takiyama and Morita 1993). Radius r_1 and r_2, the coefficients of MTF3- and MTF4-elements, are decided from the function of engine torque, engine revolution, vehicle speed and vehicle acceleration force. Radius r_1 is changeable on the arrow direction of solid line. The optimum operating line shown by chain line in Figure 10 is represented as Equation (11) obtained with the least-square method. Radius r_2 is given by Equation (12), where R is constant and is the sum of r_1 and r_2. Then, gear ratio i of CVT-2 is represented by Equation (13).

$$T_e = -1.0 \times 10^{-6} \times N_e^2 + 1.45 \times 10^{-2} \times N_e + 79.29 \quad (11)$$

$$r_2 = R - r_1 \quad (12)$$

$$i = r_2 / r_1 \quad (13)$$

3. JAPANESE AUTOMOTIVE TEST DRIVE CYCLES

In the present study, the test driving mode is Japanese test drive cycle, called as the Japanese 10 Mode cycle and shown in Figure 11, which approximates a situation of driving in the city by combining ten kinds of driving patterns such as idling, acceleration, constant speed, and deceleration.
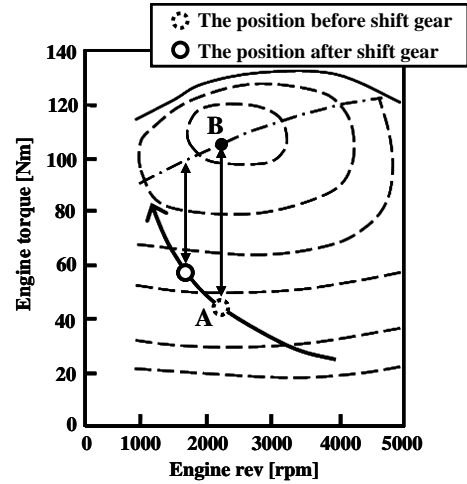

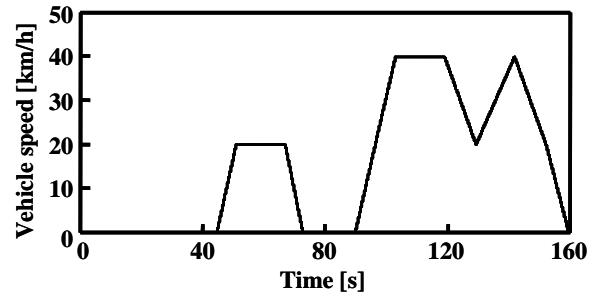Figure 10: Active Control of CVT-2 (Yokoi 1979)

Figure 11: The Japanese 10 Mode cycle

Table 1: Automotive Specifications

CVT gear ratio	0.435 – 6.4
Final gear ratio	5.69
Vehicle weight	1305 [kg]
Frontal projected area	1.8 [m^2]
Coefficient of drag	0.36
Tire radius	0.273 [m]
Coefficient of rolling resistance	0.018
Air density	1.166 [kg/m^3]

4. SIMULATION RESULTS

Table 1 shows automotive specifications of the analysis target in the present study. Time history of the gear ratio of the CVT is shown in Figure 12 which is obtained by Bond Graph simulation. In the Figure a broken line and a solid line indicate the calculated results of CVT-1 and CVT-2, respectively. The different results are caused by the difference of gear change pattern between CVT-1 and CVT-2. Hereafter in all figures the chain line, the broken line and the solid line indicate the 10 Mode cycle, the results of CVT-1 and CVT-2, respectively.

Figures 13, 14, 15, 16, 17, 18, and 19 show time histories of engine revolution, engine torque, engine output power of CVT-1 and CVT-2, sum of mileage according to the Japanese 10 Mode cycle, fuel consumption per unit time, sum of fuel consumption, and average fuel consumption calculated by Equation (7) from Figure 18, respectively.

In Figure 13, engine revolution of CVT-2 is always lower than that of CVT-1. Though engine revolution of CVT-1 is kept constant for constant speed driving mode, engine revolution of CVT-2 always varies even for constant speed driving mode. It is because the gear ratio is continuously-varied to determine engine revolution with high efficiency in CVT-2, as shown in Figure 12.

On the other hand, in Figure 14 the engine torque of CVT-2 is always larger than that of CVT-1, contrary to features of engine revolution shown in Figure 13. As a result, the engine output power is almost same between CVT-1 and CVT-2, as shown in Figure 15, though the gear ratio is different. This is because parameters except for the driving resistance as an input variable and CVT are the same between the simple model of CVT-1 and the detailed model of CVT-2. This result shows the validity of CVT-1 and CVT-2 models.

Both Figures16 and 18 have upward-sloping curves, which are similar tendency in sum of mileage and sum of fuel consumption. It is because vehicle speed late in the Japanese 10 Mode cycle is twice as fast as that early in the 10 Mode cycle and fuel consumption increases rapidly in proportion to vehicle speed especially late in the 10 Mode cycle. At acceleration period, engine torque and revolution increase and fuel consumption increase similarly, though average fuel consumption deteriorates inversely.
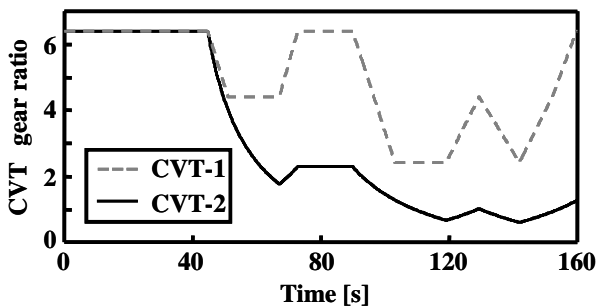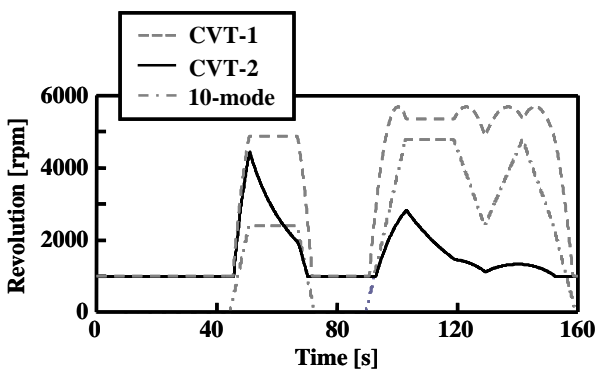

Figure 12: CVT Gear Ratio
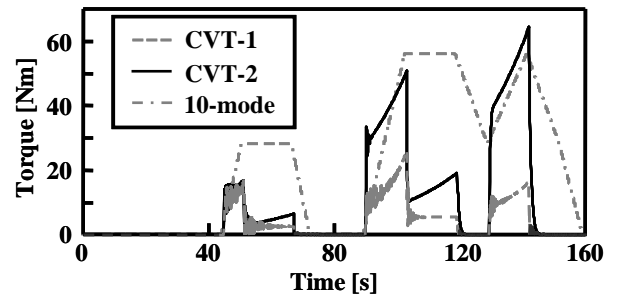

Figure 13: Engine Revolution
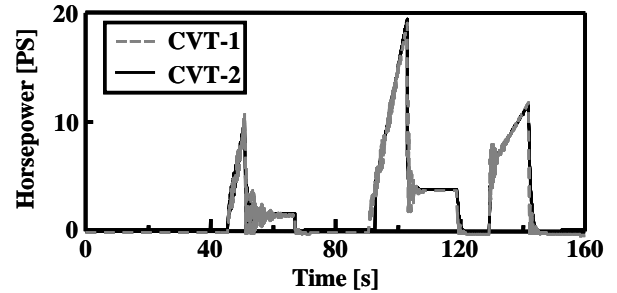

Figure 14: Engine Torque
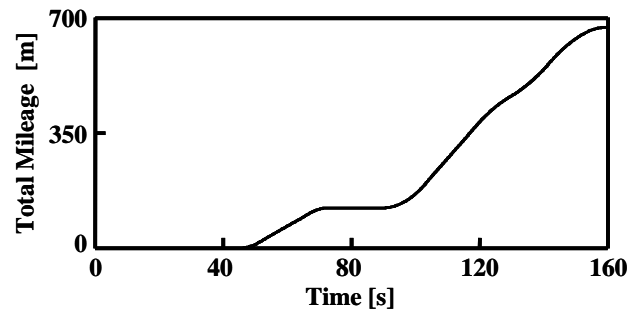

Figure 15: Horse Power
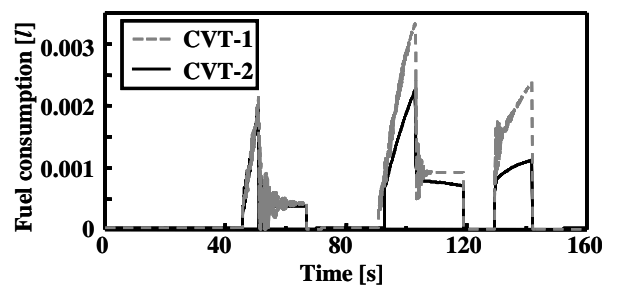

Figure 16: Sum of Mileage (The 10 Mode cycle)

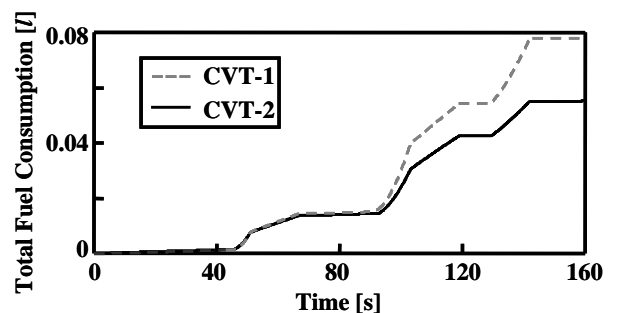

Figure 17: Fuel Consumption
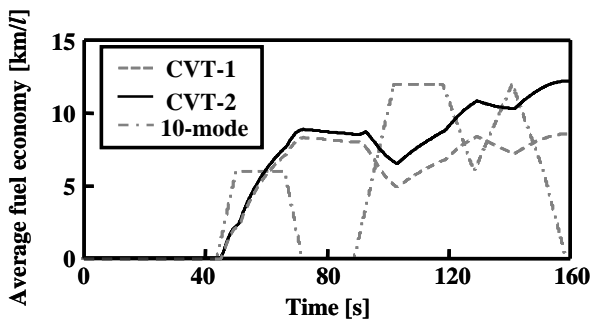

Figure 18: Sum of Fuel Consumption

Figure 19: Average Fuel Consumption

In Figure 19 the average fuel consumption for 160 [s] according to the Japanese 10 Mode cycle is finally calculated as 12 [km/*l*] and 8.6 [km/*l*] in CVT-2 and CVT-1 model, respectively. It is caused by controlling the gear change minutely that the average fuel consumption in CVT-2 is better than that in CVT-1.

From the results mentioned above, it is verified that the Bond Graph models represent the basic dynamic characteristics of power train system with CVT and it is possible to calculate the optimum gear change pattern for high fuel-efficiency by combining the above Bond Graph models with some control programs for gear change pattern in CVT.

PC calculating time is 72 [s] for 160 [s] the Japanese 10 Mode cycle and is almost same between the models of CVT-1 and CVT-2.

5. CONCLUSION
The proposed Bond Graph models represent the basic dynamic characteristics of vehicle power train system with CVT and it is useful to calculate the optimum gear change pattern for low fuel consumption.

REFERENCES
Carbone, G., Mangialardi, L., Bonsen, B., Tursi, C., Veenhuizen, P.A., 2007. CVT dynamics: Theory and experiments. *Mechanism and Machine Theory* 42: pp.409-428

Cho, D., Hedrick, J.K., 1989. Automotive Powertrain Modeling for Control. *Transaction of the ASME Journal of Dynamic Systems, Measurement, and Control* Vol. 111: pp568-573.

Daihatsu Motor Co., Ltd.,2006. Continuously Variable Transmission. *Challenge Next Press Information* Vol.4.

Gawthrop, P., 1995. Bicausal Bond Graphs. *International Conference on Bond Graph Modeling and Simulation*, pp. 83-88. January 15-18, Las Vegas (Nevada, USA).

Karnopp, D., Rosenberg, R.C., 1970. Application of Bond Graph Techniques to the Study of Vehicle Drive Line Dynamics. *Transactions of the ASME Journal of Basic Engineering.* June: pp355-359

Karnopp, D.C., Margolis, D. L., Rosenberg, R. C., 2006. *System Dynamics.* New Jersey: John Wiley & Sons, Inc.

Kong, L., Parker, R.G., 2008. Steady mechanics of layered, multi-band belt drives used in continuously variables transmission (CVT). *Mechanism and Machine Theory* 43: pp.171-185

Hrovat, D., Tobler, W.E., 1991. Bond Graph Modeling of Automotive Power Trains. *Journal of the Franklin Institute Pergamon Press plc*: pp. 623-637.

Mantriota, G., 2002. Performances of a parallel infinitely variable transmissions with a type II power flow. *Mechanism and Machine Theory* 37: pp.555-578

Pfiffner, R., Guzzella, L., Onder, C.H., 2003. Fuel-optimal control of CVT powertrains. *Control Engineering Practice* 11: pp.329-336

Srivastava, N., Haque, I., 2008. Transient dynamics of metal V-belt CVT: Effects of band pack slip and friction characteristic. *Mechanism and Machine Theory* 43: pp.459-479

Srivastava, N., Haque, I., 2009. A review on belt and chain continuously variable transmissions (CVT): Dynamics and control. *Mechanism and Machine Theory* 44: pp.19-41

Srivastava, N., Haque, I., 2009. Nonlinear dynamics of a friction-limited drive: Application to a chain continuously variable transmission (CVT) system *Journal of Sound and Vibration* 321: pp.319-341

Suzuki, T., Hosoi, K., 2007. Development of Fuel Economy Simulation for Heavy-Duty Vehicles with Automatic Transmission. *Journal of Automotive Study* Vol.29, No.12: pp15-18.

Takiyama, T., Morita, S., 1993. Study of the Control Algorithm for Engine-CVT Consolidated Control. *JSME Journal Series B* Vol. 59: pp. 377-383.

Thoma, J. U., 1990. Simulation by Bondgraphs. Berlin Heidelberg New York London Paris Tokyo Hong Kong: Springer-Verlag.

Togai, K., Koso, M., 2006. Dynamic Scheduling Control for Engine and Gearshifts: Consolidation of Fuel-Economy Optimization and Reserve Power. *Mitsubishi Motors Technical Review* No.18: pp. 25-32.

Yokoi, T., 1979. Automatic Transmission Optimization for Better Fuel Economy. *Toyota Technology* Vol.100: pp. 24-29.