# Controller Design for Fuel Consumption Reduction of a CVT Equipped Vehicle

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**Abstract**

The primary objective of this work is to introduce a gear ratio selection strategy for a CVT equipped vehicle and show its effectiveness on the fuel consumption reduction. A Fuzzy control algorithm is designed for this purpose. A nonlinear model is developed for simulating the longitudinal vehicle dynamics with accelerator pedal applied by the driver as an input. In order that pedal input values can be used for evaluation of control strategy, a pedal cycle concept was introduced. With the help of these cycles different driving conditions were simulated and the fuel consumption results were obtained using Advisor software. Results showed that the control system was successful in reducing the fuel consumption, especially in low acceleration driving cycles.

## 1. INTRODUCTION

Modern passenger cars are expected to be comfortable, high performing and safe. Growing expectations from drivers and tougher legislations imposed by governments urge the designers and manufacturers to consider new challenging issues. Despite all efforts, the passenger car is still a fairly inefficient system. As far as the transmission is concerned, part of the mechanical power generated by the engine is lost and not used to propel the wheels. Various methods and potentials for fuel economy improvement by enhancing the efficiency of individual driveline components have been studied and employed by the manufacturers over the years. Improving the efficiency of traditional transmissions and that of the CVT has also been in focus of attention. Despite the lower efficiency of CVTs compared to manuals, they can control the engine operating points which in turn can improve the overall vehicle efficiency and fuel economy. An optimum CVT control strategy, however, is not an easy task owing to two partially opposite features that have to be satisfied: the reduction of fuel consumption and the requirement of appropriate drivability and acceleration performance [1].

A bond graph model of continuously variable transmission was established and a synthetic control algorithm was developed by Qin et al [2] for the purpose of improving the ratio change response and reducing the engine torque response lag. The simulation result showed that by this algorithm the engine torque response lag could be reduced and the driving torque response could be improved effectively.

Kim and Vachtsevanos [3] designed a fuzzy controller to control the primary pressure of the CVT. The structure and parameters of the fuzzy logic were designed via clustering techniques and through genetic algorithms. The controller is shown to be robust against parameter variations, with improved convergence time for the genetic algorithm.

Pfiffner and Guzzella [4] considered the fuel-optimal operation of a CVT-equipped powertrain in stationary and in transient conditions. For transient vehicle operation a cost function was defined and the optimal control problem was solved. Two different engines namely a conventional Spark Ignition (SI) and a Downsized Super Charged (DSC) SI engine were considered. Larger gains in fuel economy were observed for DSC engines.

Takiyama and Morita [5] applied Linear Quadratic Integral (LQI) theory to control the drivetrain with aim to satisfy the demanded speed and better fuel economy. From the results obtained it was concluded that the inclusion of engine power is essential in the weight factor modification of LQI algorithm for the desired performance, fuel economy and vehicle acceleration.

Kim et al [6] used a feed forward PID (Proportional-Integral-Derivative) controller for the line pressure control together with a Fuzzy control for the CVT ratio. The effect of CVT shift dynamics and on-off characteristics of the ratio control valve were also considered. With the use of experimental results it

was concluded that a desired speed ratio could be achieved at a steady state by the fuzzy logic in spite of the fluctuating primary pressure.

Mantriota, G. [7] studied the performance of a passenger car equipped with a Power Split CVT system. The aim was to improve the efficiency of the CVT by means of a power flow without recirculation using two separate phases of operation. They evaluated the vehicle's fuel consumption with the hypothesis of choosing the transmission ratio that minimizes the specific fuel consumption.

Bonsen, B. et al [8] studied the possibilities of improving the CVT by minimizing variator clamping forces by using a slip control system. Experimental results on two test rigs are also presented.

Cho, B. and Vaughan, N.D. [9] studied the control strategy of a mild hybrid electric vehicle (HEV) equipped with a CVT. They introduced an ideal operating surface (IOS) to increase tank-to-wheel efficiency.

Adachi, K., Ochi, Y. and Kanani, K. [10] proposed a control system for a belt-driven CVT system. A feed-forward controller was designed by a combination of the inverse system of the plant and a reference model that gave desired output response. The results showed that the control system improved fuel-efficiency by choosing proper gear ratios.

Pesgens, M. et al [11] focused on the development of a ratio controller for a hydraulically actuated metal push-belt CVT. The controller consisted of an anti-windup model based PID feedback. Vehicle experiments presented to show that adequate tracking together with good robustness against actuator saturation was obtained.

In the current work with the use of a pedal cycle concept a link is made between the results of the developed forward facing model with those of the well established Advisor model for the comparison purposes. A Fuzzy control algorithm is also developed for the gear ratio selection of a CVT system with main objective of reducing fuel consumption. The controller is found to be successful in reduction of fuel consumption especially at low longitudinal accelerations.

## 2. MODELING

In order to simulate the performance of the introduced control algorithm in response to driver's various demands and resulting effects on the fuel consumption, a comprehensive vehicle model is developed. The model input is accelerator pedal angle and output is vehicle speed.

The pedal angle applied by driver, in one hand and engine speed on the other hand, enter to engine model and engine torque is calculated. The powertrain model calculates the effective output torque from driving wheels by using gear ratio determined by the controller, lost torque from rotating parts and also friction torque. Output torque to the driving wheels is entered to vehicle dynamics model and by using longitudinal dynamics equations, quantities such as wheel angular velocity, vehicle speed and acceleration are calculated. The engine rotating speed is then determined by kinematical equations of powertrain.

### 2. 1. Internal Combustion Engine

The internal combustion engine used here is a conventional port-injected SI with following differential equations:

$$\frac{d}{dt}Lp_{act} = \frac{1}{\tau_1}[Lp(t) - Lp_{act}(t)] \tag{1}$$

$$\frac{d}{dt}T_e(t) = \frac{1}{\tau_2}(fT_e(Lp_{act}(t),\omega_e(t)) - T_e(t)) \tag{2}$$

In the first equation, $Lp_{act}$ is throttle angle and $Lp$ is pedal angle. $\tau_1$ indicates the time constant of lag between pedal and throttle. This equation shows that there is no mechanical joint between pedal and throttle.

The second differential equation is representing lag between developed engine torque and input throttle pedal angle with $\tau_2$ indicating the time constant. In this equation $T_e$ and $\omega_e$ are the engine torque and speed respectively. $fT_e$ is the quasi-static engine torque function usually obtained by dynamometers. The Fuel Consumption (FC) of engine is obtained by integration of instantaneous engine quantities:

$$FC = \int T_e(t)\omega_e(t)SFC(\omega_e,T_e)dt \tag{3}$$

In this formula $SFC$ is engine specific fuel consumption, $gr/(kw.h)$, and is a function mainly of engine speed and torque.

### 2. 2. Powertrain

The powertrain consists of a series of subsystems that carry torque from engine to wheels. This system includes gearbox, clutch, differential and axle. Torque flow is described by following equations:

$$T_c = T_e - I_e . \alpha_e \qquad (4)$$

$$T_{CVT} = n_{CVT} T_C \eta_{CVT} - I_{CVT} \alpha_{CVT} \qquad (5)$$

$$T_{AXLE} = n_{ds} . T_{CVT} - I_{ds} . \alpha_{ds} \qquad (6)$$

Where $n_{CVT}$ and $n_{ds}$ are CVT and differential gear ratios respectively. I stands for moment of inertia and subscripts $e$, $ds$ and CVT indicate engine, differential and CVT transmission respectively. $T_{CVT}$ is the output torque of CVT and $T_{Axle}$ is torque on axle and wheels. $T_c$, the transferred torque in clutch is proportional to spring force and in transient conditions depend on difference between input and output speeds. However, the transient behavior is ignored and therefore the clutch is either engaged at accelerating situations or disconnected in braking conditions.

## 2. 3. Vehicle Dynamics

The vehicle longitudinal dynamics is governed with below equations.

$$F_{xf} + F_{xr} - F_R = M \frac{dv}{dt} \qquad (7)$$

$$F_R = F_{RR} + F_g + F_A \qquad (8)$$

Longitudinal axle forces $F_{xf}$ and $F_{xr}$ are functions mainly of longitudinal tire slips and normal force on driving wheels. The Magic Formula representation has been considered in this work [12]. Since a front wheel drive vehicle is considered, $F_{xr}$ will vanish in the Equation 7. $F_R$ is the total resistance force against vehicle motion with $F_{RR}$, $F_g$ and $F_A$ being rolling resistance, gravitational and aerodynamic resistance force respectively.

Engine speed, one of the inputs of internal combustion engine model, is calculated by the kinematical relations of powertrain:

$$\omega_e = \omega_{wh} n_{ds} n_{CVT} \qquad (9)$$

$$v = \omega_{wh} R_{wh} \qquad (10)$$

In which $v$, $\omega_{wh}$ and $R_{wh}$ and vehicle speed, wheel speed and radius respectively. The second input (pedal angle) is specified by the driver.

## 2. 4. Model Verification

In order that the results of current model built in SIMULINK environment be validated, justification of results in comparison with a suitable software such as Dymola was considered. This software is especially for dynamic simulation and it has some prepared models for mechanical components like gearbox, spring and damper. A simulation of vehicle longitudinal dynamics was prepared with this software with a constant gear ratio equal to maximum gear ratio of the gearbox. Obviously all parameters in both simulations were taken identical. For engine modeling a 2-D table was used for both simulations.

Results obtained from longitudinal dynamics simulations of vehicle with both software are shown in
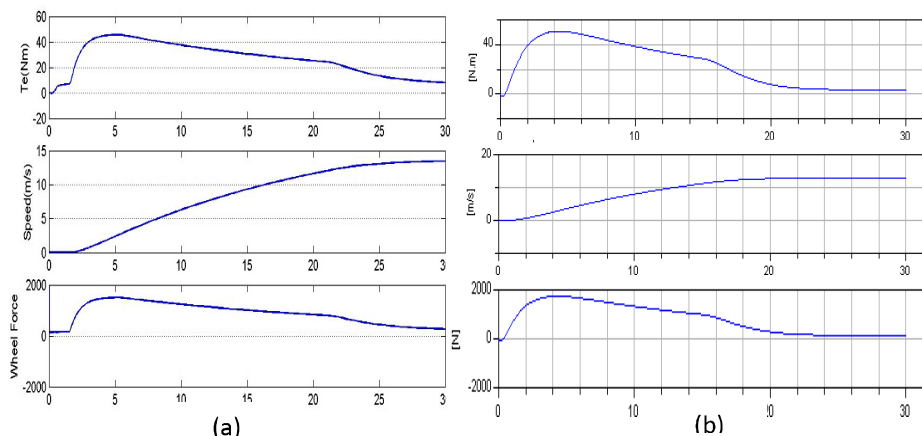


(a)                               (b)

**Fig. 1.** Results compared a) SIMULINK. and b) Dymola

figures 1.a and 1.b. The results including the variation of engine torque, vehicle speed and longitudinal force with time are similar to a large extent. Although some little differences between the results may be seen but overall the two simulations show similar trends and values.

## 3. OBJECTIVE FUNCTION

Despite the fact that the mechanical efficiency of CVTs in general is less than that of common manual gearboxes, nevertheless this type of transmission can play an effective role in reducing fuel consumption by transferring engine operating points near to optimum working conditions. In Figure 2 are shown the engine map and important parameters such as constant power and optimum consumption curves.

To reduce fuel consumption, engine must be kept close to minimum specific fuel consumption contour at any power output, shown by Optimum Fuel Consumption Curve (OFCC) of Figure 3. In CVTs continuous gear ratios makes it possible to smoothly move over OFCC. However, there are still restrictions in the gear ratios that put limitations on engine operating points.

The required engine operating point can be achieved by proper control of input parameters, namely throttle angle and gear ratio. No dynamics between pedal and throttle is considered unless it is assumed that throttle follows the pedal with a delay.

One objective for the controller could be bringing engine operating points as close to OFCC as possible, at various road conditions and for all pedal inputs

applied by the driver. This strategy, however, may cause behaviors unexpected by the driver such as unexpectedly increasing or decreasing of engine speed for different pedal inputs. The said strategy does not take into consideration the driver's demands and only accounts for fuel economy. In order to have a compromise, the control strategy must consider both the driver's demand and fuel consumption at the same time.

## 4. DESIGN OF CONTROLLER

According to the objectives described earlier, the controller strategy is to track OFCC for reducing fuel consumption. However, modifications must be made to also consider driver's demands. In this strategy the main goal is reduction of FC so for each driving cycle and every input pedal applied by driver, the engine must be kept close to optimum fuel consumption curve.

Pushing accelerator pedal by the driver will have different outcomes in vehicles with manual gearboxes and CVT equipped vehicles. In the former case the consequence is simply a proportional acceleration, whereas in the CVT case it is not the same, owing to the existence of controller actions.

The controller must decide the engine throttle and gearbox ratio values in order that engine optimum operation is achieved at every instant. There are, however, limitations in the throttle values or gearbox ratios that controller can choose.

Information regarding the fuel consumption of engine is often available in lookup table formats. At each engine torque, the target engine speed is determined by the controller, taking into account the engine and gearbox limitations.

The gear ratio $n_{CVT}$ defined as below:

$$n_{cvt}(t) = \frac{\omega_{ds}(t)}{\omega_e(t)} \tag{11}$$

has following limitation due to upper and lower fixed ratios:

$$n_{cvt_{min}} \le n_{cvt} \le n_{cvt_{max}} \tag{12}$$

As Fig. 3 shows, the inputs of controller are engine torque, engine speed and driver's pedal angle whereas the output is CVT ratio.
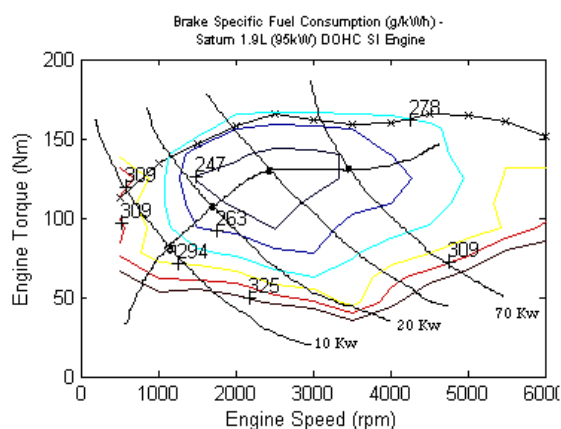


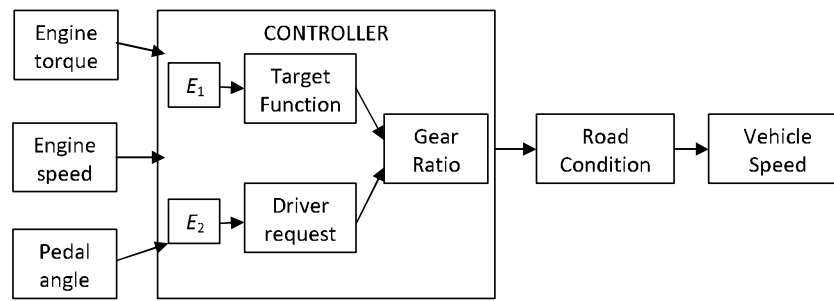**Fig. 2.** Optimum fuel consumption curve (OFCC)

**Fig. 3.** Schematic of controller operation

The inputs to the control system are converted to controller inputs $E_1$ and $E_2$ with following equations:

$$E_1 = K_1(\omega_e - \omega_d) \tag{13}$$

$$E_2 = K_2(T_e - T_d.) \tag{14}$$

$$T_d = T_{max} \theta / 100 \tag{15}$$

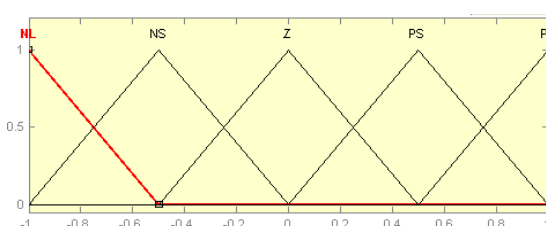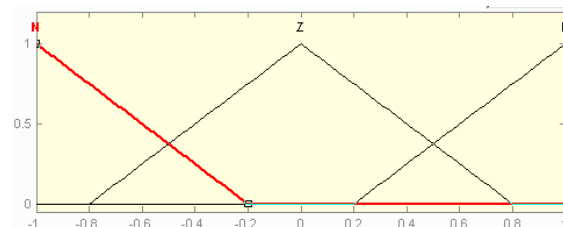$$\frac{d}{dt} n_{CVT} = K.u \tag{16}$$

$\omega_e$ and $\omega_d$ are current and target engine speeds respectively. $K_1$ and $K_2$ are two constant gains for adjusting the values of controller inputs to lie between +1 and -1 for the purpose of fuzzy membership function definition. $K$ is an amplification gain relating the final output to preliminary controller output $u$. $T_e$ is the current engine torque and $T_d$ is desired engine torque. It is worth mentioning that in equation 15 the desired torque of engine is taken proportional to the throttle opening. In this equation $T_{max}$ is the maximum engine torque in present engine speed and $\theta$ is the throttle opening in percent.

The first controller input $E_1$ is proportional to difference between the actual and target engine speed. The desired engine speed is chosen according to objective function and engine operating point relative to optimum FC curve. The second input, $E_2$, is proportional to difference between the actual and desired engine torques corresponding to the pedal angle. This will account for the driver's demands at different driving conditions.

The complexity of system equations and the nature of gear shifting that is historically a human experience, makes the Fuzzy control algorithm a good candidate for this decision making. Designed membership functions for controller inputs and outputs are shown in Figures 4-6.

Design of the rules was primarily based on subjective driving experience and then by trial and error and evaluation of different driving cycles, errors were removed until satisfactory results generated. The governing Fuzzy rules are prepared in a table format shown in Table1. The resulting surface is also shown in figure.7.



**Fig. 5.** Membership functions of second input ($E_2$)



**Fig. 4.** Membership functions of first input ($E_1$)
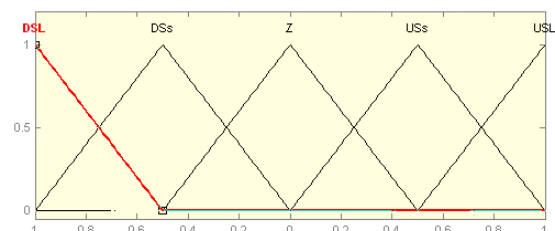


**Fig. 6.** Membership functions of output (U)

**Table 1.** The Fuzzy rules

| STATE | | E1 | | | | |
|---|---|---|---|---|---|---|
| | | NL | NS | Z | PS | PL |
| | N | $US_L$ | $US_S$ | Z | $DS_S$ | $DS_L$ |
| | Z | $US_S$ | Z | Z | $DS_S$ | $DS_L$ |
| E2 | P | $US_S$ | Z | $DS_S$ | $DS_L$ | $DS_L$ |
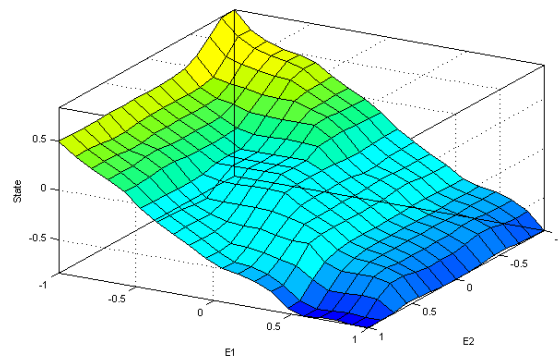


**Fig. 7.** Fuzzy inferred surface

## 5. SIMULATION RESULTS

A proper benchmark is necessary in order that the effect of suggested control strategy on the fuel consumption could be compared. Advisor software was found useful owing to fact that this software is well known in the field of vehicle energy and emission performance. Advisor uses a backward facing solution using speed-time driving cycles as input and calculating driveline and engine loads in a quasi-static manner.

Simulation in this research, however, is based on a more realistic forward facing solution in which the input to the system is accelerator pedal. In other words, in order to follow a specified driving cycle, the driver must apply appropriate pedal angles at each instant so that vehicle speed versus time follows the desired driving cycle.

Two standard driving cycles, namely Extra Urban Driving Cycle (EUDC) and COMMUTER, are chosen to produce primary results. These cycles are suitable for simulating by the forward model by trial and error in pedal angle. The output time history of simulation resembles EUDC driving cycle to a great extent. Therefore the other outputs such as fuel consumption can be associated to the same driving cycle.

In Figure 8 comparison is made between the results obtained from forward model and Advisor during the course of this driving cycle. This figure shows the variation of vehicle speed, gear ratio, engine speed and engine torque with time. The initial and maximum gear ratios are kept the same for both models. The speed-time graph shows that for the forward model vehicle traverses the driving cycle with a small deviation. It is noticeable that the gear ratios and as a consequence other parameters are different for the two cases.

Results obtained for the COMMUTER cycle from
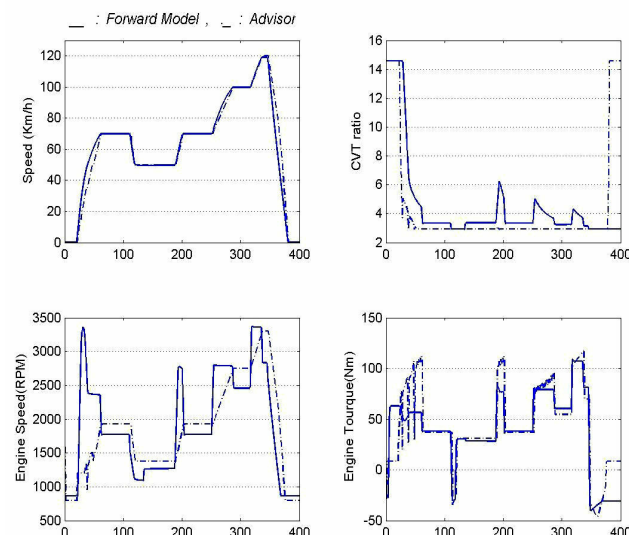


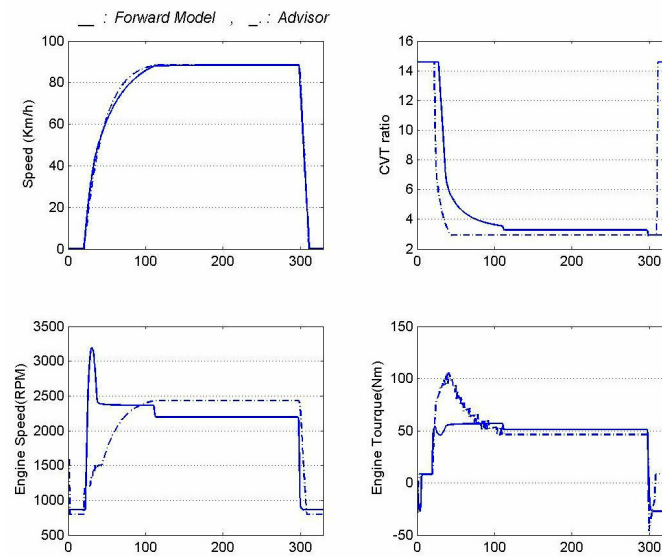**Fig. 8.** Results for EUDC driving cycle.

**Fig. 9.** Results for COMMUTER driving cycle.

both models are compared in Figure 9. Engine speed and torque graphs show that in steady state condition, in forward model gear ratio is selected in such a way that the engine speed is lower whereas engine torque is higher than those of Advisor. In transient conditions, the difference between the results of two methods is substantial.

Results generated in this section were chosen to investigate the behavior of controller in selecting gear ratios. It was found that the controller is capable of regulating the gear ratio according to design objectives.

## 6. THE CONCEPT OF PEDAL CYCLES

In a backward facing simulation, the driving cycles are input to the system. In forward facing simulations, however, a driving cycle could be generated from system outputs. For the sake of comparison between the fuel consumptions of a forward facing simulation with that of a backward facing simulation for a specific vehicle, the speed-time output of the former must be identical to the driving cycle input of the latter. If a standard driving cycle is concerned, for the forward facing model a pedal angle pattern should be found by trial and error so that the simulation output generates the specific driving cycle of interest. For most driving cycles owing to their complex speed–time structures, finding proper input patterns will not be impossible but surely will be cumbersome.

As a solution for the problem of comparing the results of the two methods, the pedal cycle concept was introduced. This concept is similar to speed-time cycle or driving cycle when defined for variation of pedal angle with time. By this method, the driver's pedal input versus time is defined for the forward
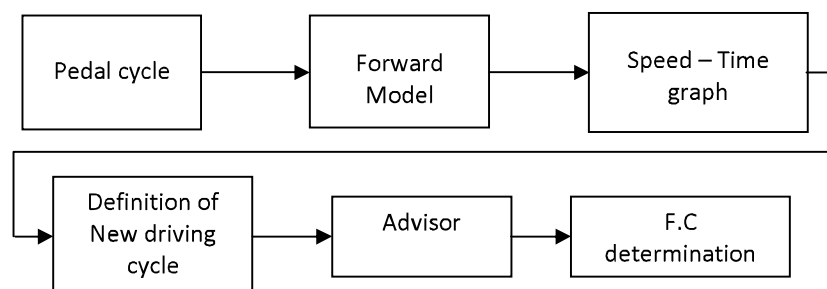


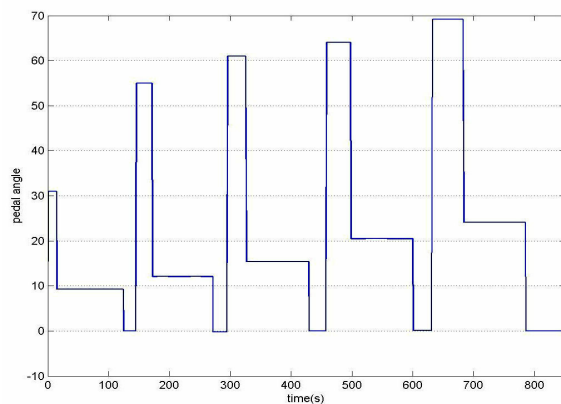**Fig. 10.** Algorithm of pedal cycle used in conjunction with Advisor

**Fig. 11.** Input Pedal angle designed for first cycle

facing simulation. The result would be a speed–time variation that is a driving cycle by definition. A backward facing simulation like ADVISOR can take this time history as an input file. Figure 10 clearly shows this procedure.

This technique will allow comparing fuel consumptions between the two forward and backward simulations in complicated driving cycles. It is also believed that the pedal cycle concept is a practical way of comparing the performance of actual vehicles on the roads or on the chassis dynamometers.

In the first step a pedal cycle was built for the purpose of simulating an existing standard driving

cycle, namely the "5 peak". Owing to sharp edges in this driving cycle, a forward looking simulation can hardly follow it. There should be a particular pattern for the accelerator pedal so that the resulting speed of the vehicle resembles the driving cycle as close as possible. Such pedal cycle will look like as that shown in Figure 11.

The speed–time graph in Figure 12 is the response of forward model to the said input pedal cycle. This output is quite similar to "5 peak" with only small differences. For fuel consumption calculation, this speed-time history is defined as a new driving cycle input to Advisor. With this concept, differences in inputs are eliminated and as a result, real difference between the forward model results and those of Advisor are obtained in the rest of figures.

Another pedal cycle shown in Figure 13 is chosen to observe the performance of the controller in low acceleration driving cycles. Short durations and low throttle openings produce low levels of engine torque leading to low longitudinal accelerations. The speed–time output is first obtained using the forward model and then it is fed into Advisor to obtain fuel consumption results. The output results obtained from both forward and backward simulations is compared in Figure 14.

Although a single driving cycle is used for both cases, outputs of the two models have significant
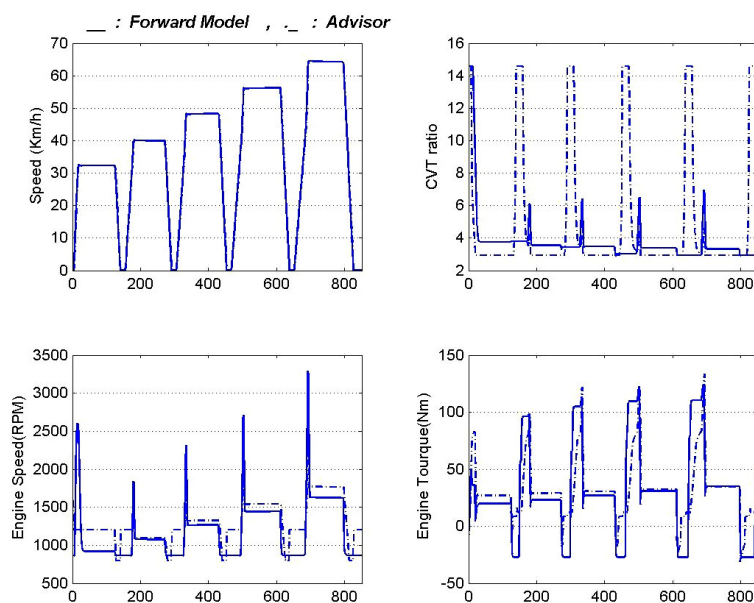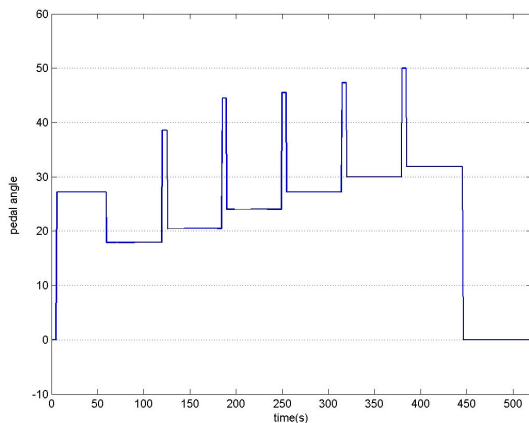


**Fig. 12.** Results obtained from forward model and ADVISOR in First pedal cycle

**Fig. 13.** Input Pedal angle corresponding to second pedal cycle

differences. Results show clear differences in gear ratios, engine speed and torque values during the simulation period. These differences in fact mean the operating points of engine are different in the two cases and it is expected that fuel consumption to vary accordingly. The dynamic nature of the forward simulation in contrast with the quasi static nature of the backward simulation is believed to cause the main differences between the two models.

By using the pedal cycle concept, both the forward and backward models use exactly equal driving cycles and as a result, the real difference between the two models can be made clear. This allows one to study the

effectiveness of the controller in reducing the vehicle fuel consumption in different driving conditions.

## 7. SENSITIVITY

Results obtained so far indicated that for different driving cycles, FC values had considerable differences. In order to investigate the effect of influencing factors on the reduction of FC, two cases will be examined in this section.

### 7. 1. Cycle Length

It was shown earlier that different driving/pedal cycles had different effects on the fuel consumption. The intention is to investigate the effect of cycle length. In other words for a previously considered cycle, namely the first pedal cycle, the length will be extended. One possible extension is repeating the same cycle. The results obtained for this combination is provided in table 2.

The first result is that the absolute FC for each model in longer cycle decreases. At the same time repetition of pedal cycle has caused more reduction in FC for the model introduced here, compared with that of Advisor. The engine temperature is found to play the main role in this respect. In fact in the cycle with original length, the effect of cold start is more
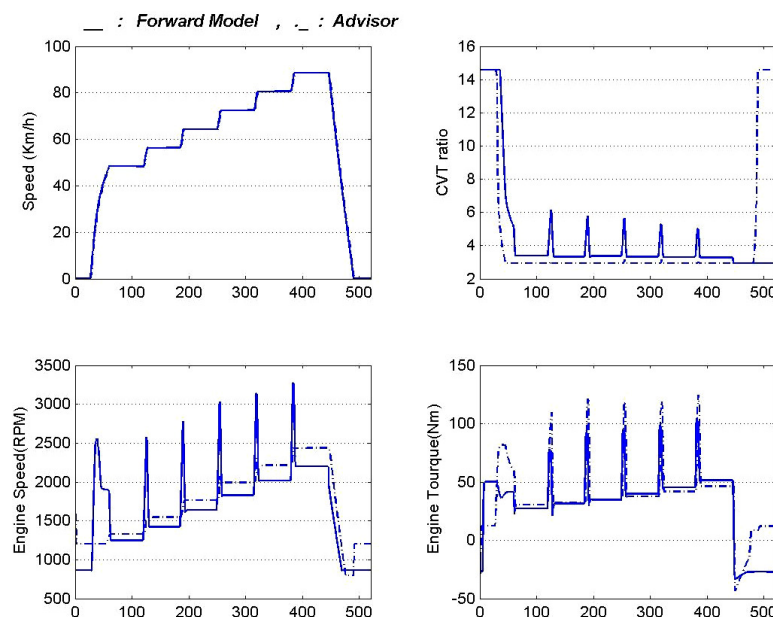


**Fig. 14.** Results for second pedal cycle.

Table 2. FC results showing the effect of cycle length

| Driving cycle | Model | FC (gal) | FC (L/100Km) | FC reduction |
|---|---|---|---|---|
| First Pedal cycle | Advisor | 0.1588 | 7.2 | 1.2% |
| | Current Model | 0.1569 | 7.1 | |
| First Pedal cycle repeated (1700s) | Advisor | 0.3156 | 7.0 | 1.5% |
| | Current Model | 0.3108 | 6.9 | |

dominant since in this case stable engine temperature can not be reached.

## 7. 2. Membership Functions

Because the number of membership functions is effective on the performance of fuzzy controllers, the effect of this design parameter on the results will be examined. The original number of membership functions in this work was obtained by trial and error and therefore is not an optimum choice. The number of membership functions of the second controller input, $E_2$, is increased from 3 to 5 and for output membership functions from 5 to 9. Using the same strategy; number of fuzzy rules will increase to 25 as shown in below Table 3.

Results showed that by increasing membership functions, gear changing sensitivity was increased relative to original case. Results for EUDC driving cycle, however, showed that increasing number of membership functions has rather small effect on FC reduction. That is only 1.784% compared to the base case with 1.64% total FC reduction (see Table 4).

## 8. DISCUSSIONS

The effectiveness of control strategy under different driving/pedal cycles has been investigated. For the different inputs of pedal angle applied by the driver, the outputs in terms of vehicle speed-time history were generated. With the objective of fuel consumption reduction, the fuzzy control system was responsible to determine suitable gear ratios at different driving conditions. The response of system to different standard driving cycles was first investigated by finding proper pedal inputs.

The design of fuzzy controller was performed in such a way that for changes in driving conditions and driver's demands, the best gear ratio be chosen to shift engine on optimum fuel consumption curve of engine map. For this goal, the two influencing factors $K_1$ and $K_2$ (see equations 13 and 14) which control the system response and stability were chosen properly. It was observed that increasing these two values resulted in high accuracy of output response but the model became unstable. On the other hand, decreasing the two gains caused more stability but low accuracy.

Table 4. Effect of increasing membership functions on fuel consumption results

| STATE | | E1 | | | | |
|---|---|---|---|---|---|---|
| | | PL | PS | Z | NS | NL |
| E2 | NXL | NXL | NXL | NXL | NXL | NL |
| | NS | NS | NS | NS | NS | NS |
| | Z | Z | Z | Z | Z | Z |
| | PL | PL | PL | PL | PL | PS |
| | PXL | PXL | PXL | PXL | PXL | PL |

Table 3. Fuzzy rules

| Model | Fuel consumption(gal) | | FC reduction |
|---|---|---|---|
| | Current Model | Advisor | |
| Main controller | 0.1323 | 0.1345 | 1.640 % |
| Membership functions increased | 0.1321 | 0.1345 | 1.784 % |

Optimum values, therefore, were found by trial and error in such a way that created a balance between stability and output accuracy.

The sensitivity of introduced forward model to the different driving conditions and pedal inputs can be seen from gear ratio results of the model in comparison to those of Advisor. For example every time that pedal is pressed, the gear ratio is increased allowing for acceleration (e.g. pedal angle in Figures 11, 13 and CVT gear ratio in Figures 12, 14). This increase in gear ratio after a few seconds causes engine speed to increase, so that the controller decreases the gear ratio in order to minimize the fuel consumption. In other words, the controller fulfills both driver's demands and fuel economy at the same time. This achievement must be considered in view that fuel consumption reduction with using software methods of controlling transmission is only about a few percents, and is due to transferring unsuitable engine working conditions with changing gear ratios to optimum points at different situations.

The reduction of fuel consumption was found different for various driving cycles. Amount of the reduction in FC has been found to depend on average acceleration of driving cycle. In Table 5 is shown the dependency of fuel consumption to average acceleration of driving or pedal cycle. The maximum reduction of FC as can be seen in Table 5 is for second pedal cycle that demonstrates a low acceleration driving cycle. By using a driving cycle with higher average acceleration, the reduction percentage is reduced.

Further investigations were also conducted in order to study the effect of influencing parameters such as membership functions and cycle length (time). Results indicated that the amount of FC reduction is independent from vehicle weight and the number of membership functions, although increasing of

membership functions causes more sensitivity of controller to pedal angle and road conditions but have no effective on fuel consumption. The cycle duration on the other hand, plays a role in the amount of absolute fuel consumption due mainly to the engine cold-start effects. In other words, for the longer cycles the fuel consumption reduction will be more effective since the effect of cold-start at the start of cycle will be shared in a longer period, making the average FC smaller.

## 9. CONCLUSIONS

In this paper a Fuzzy control algorithm was developed for the gear ratio selection of a CVT system with main objective of reducing fuel consumption. A forward facing simulation was used for controller evaluation. The model input is pedal angle and its output is speed-time history of vehicle. Owing to Advisor's strength in fuel consumption estimation, it was used for this purpose. Advisor, however, is a backward facing simulation and its input is speed-time graph and its outputs are vehicle and engine parameters. In order that the two simulations are linked a new concept named "pedal cycle" is introduced in this work. A pedal cycle is given as input to the forward simulation and then its output is fed as input to Advisor.

Two pedal cycles and two standard driving cycles were considered for the evaluation of control algorithm. Results obtained for the fuel consumption showed that the algorithm has been successful in proper gear ratio selection resulting in less fuel consumptions. Also driver demand is satisfied at the same time at various road conditions as vehicle does not miss the driving cycles.

The extent to which the controller succeeds in reducing fuel consumption depends on the average

**Table 5.** Fuel consumption results

| Driving/pedal cycle | Average acceleration | Model | FC (gal) | FC (L/100Km) | FC decrease |
|---|---|---|---|---|---|
| EUDC | 0.38 | Advisor | 0.1345 | 7.3 | 1.6 % |
| | | Current Model | 0.1323 | 7.2 | |
| COMMUTER | 0.28 | Advisor | 0.1186 | 7.0 | 1.9 % |
| | | Current Model | 0.1163 | 6.9 | |
| First pedal cycle | 0.51 | Advisor | 0.1588 | 7.2 | 1.2% |
| | | Current Model | 0.1569 | 7.1 | |
| Second pedal cycle | 0.15 | Advisor | 0.1397 | 6.5 | 3.1% |
| | | Current Model | 0.1354 | 6.3 | |

driving acceleration and cycle length. It is well understood at high accelerations the fuel consumption will raise. The controller is found to be more successful in reduction of fuel consumption at low longitudinal accelerations. At high acceleration cycles, fuel demand is high and controller is not able to reduce it since otherwise the cycle will be missed.

The results of driving the vehicle at different driving and pedal cycles show that the introduced control system is able to reduce fuel consumption up to 3.5% depending on type of driving cycle. Cold engine effects reduce amount of improvement at short length cycles, whereas at long cycles the improvement is better.

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