The Analysis of Applying CVT Gear Ratio Rate Control for Scooter Efficiency Improvement

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ABSTRACT

Continuously variable transmissions (CVTs) potentially ensure the selection of such a gear ratio that scooter fuel consumption can reach a minimum value. Traditionally, scooter CVT gearboxes are mechanically controlled, causing a gear ratio to be an engine revs function. This solution does not ensure optimum gear ratio. In this paper, the solution for fuel optimal control problem is presented. The results, obtained during brake stand tests of scooter powertrains, show the significant values of brake specific fuel consumption for the velocity that is maximum for a scooter (as stated in The Highway Code). With the introduction of a CVT gearbox, in which the selection of gear ratio can be controlled according to the worked out strategy, the solution for fuel consumption problem is found. Electromechanical actuators ensure the selection of a gear ratio independently of engine revs. Such type of construction solution makes working out the suitable control strategy, which ensures decreasing of scooter fuel consumption, possible. Presented strategies do not use precise optimization techniques. The CVT efficiency has a strong influence on a transient operation. In the paper, the control strategy that results in fuel consumption decrease of over 15% is presented. The strategy was worked by fuel consumption map for a defined scooter exploitation model. The ways of realization of the worked out strategy were tested on the brake test stand.

Key Words: CVT, gear ratio rate, control strategy, rubber V-belt, drivability

INTRODUCTION

Continuously Variable Transmission (CVT) is more and more often used in automotive application. Its large transmission ratio coverage enables the engine to operate at more profitable operating points. For every power level in an internal combustion engine, there
is one-speed-torque combination which achieves optimal fuel efficiency. Using a continuously variable range of transmission ratio, a line connecting these operating points can be followed by high drive unit efficiency. In spite of this fact, the use of CVTs in the automobile industry has remained marginal. Two-wheeler and snowmobile are major sections of the automotive industry using CVTs. In this application, a rubber dry belt is commonly used. Dry belts are usually used because a high friction coefficient is created between a belt and pulleys so that clamping force can be much smaller than it is in lubricated variants.

Description of the rubber V-belt cooperation with the bevel gears assemblies is a very complex issue. The complexity is a result of the belt transverse and longitudinal flexibility, the rubber hysteresis, the V-shape cross section of the belt and changes of the forces acting on the belt in one driving cycle. Furthermore, the changes of the forces acting on the moving vehicle influence the transmitted power and gear ratio of the CVT transmission (Cammalleri, 2018), (Julió and Plante, 2011). Further difficulties arise from the internal combustion engine vibrations (usually with one cylinder), which is a common source of motive power in this kind of vehicles.

Unfortunately, the problem which appears in non-lubricated belt pulley contact results from the lack of cooling of this contact, which causes high limits to the torque capacity of this type of the variator (Bertini et al., 2014). But these types of CVTs can be small, light and ideal for application in small motorbikes and scooters. Moreover, hydraulic control of the CVTs axial thrust is not necessary for thrust creation and, that is why hydraulic losses are eliminated. Despite these advantages, CVTs are perceived as the most inefficient transmission system. This inefficiency is the result of the application of mechanically controlled variators. Centrifugal rollers and torque cams have been widely used in rubber belt CVTs as the mechanical actuators (Chen et al., 2000), (Jantos, 2001).

The gear ratio change in the CVT transmission is usually a result of the centrifugal regulator operation. The pressing force is generated by the rollers of the control unit. The force mostly depends on the engine angular velocity and the profile of ramp curvature. In the case of the driven wheel of the transmission, the axial force is exerted by the coil spring and the torque regulator. Usually, in the papers dealing with CVT operation, the axial force resulting from the belt-bevel gears cooperation is not associated with the force resulting from the regulator assembly (Srivastava and Haque, 2009).

The performance of CVT depends on the characteristics of these actuators. By using mechanical actuators, it is impossible to shift the transmission ratio in such a way as to make speed-torque combination point remain on an engine optimal operating line (Bonsen et al., 2005), (Smetsers, 2008).

**OPTIMAL OPERATION LINE**

Optimal operation line (OOL) tracking is the most fuel-saving way to operate the drive line. The OOL can be calculated from engine map by minimizing the fuel consumption for a set of output power values. A large number of different control strategies exist. The most popular approaches are speed envelope, single track and off the beaten track. A simple approach for transient ratio shift control is to maintain the engine on the “quasi-static” peak efficiency curve which corresponds to the assumption that any arbitrary speed torque combination can be realized instantaneously. This approach is called single track strategy (Srivastava and
Haque, 2009) (Figure 1). This strategy does not use precise optimization techniques but relies on heuristic arguments. Since the vehicle performance of the single track strategy is limited, Pfiffner and all (Pfiffner et al., 2003) have proposed another approach called off the beaten track strategy. A driver can select two modes: the economy and the performance modes. These two trajectories represent different driver-selectable modes. They are presented in C 2.

![System trajectories of the single track and single track modified strategy.](image1)

Figure 1: System trajectories of the single track and single track modified strategy. The maximum gear ratio stationary driving resistance curve $\Lambda$, the peak efficiency curve for quasi-static operation $\Omega$ and the peak efficiency curve for stationary operation $\Gamma$ (Pfiffner et al., 2003)

![Engine map trajectories of the two modes off beaten track strategy on an accelerator step.](image2)

Figure 2: Engine map trajectories of the two modes off beaten track strategy on an accelerator step (Pfiffner et al., 2003)

In many cases, the actual engine torque is not controlled along the “quasi-static” peak efficiency curve but is brought directly to the final steady-state operation point. Consequently, the dynamics of the vehicle increases with the fuel consumption rising simultaneously.
Presented strategies do not use precise optimization techniques. Unfortunately, in many cases, the demanded power level is higher than engine’s capability. The engine operates at the maximum torque line until the demanded power is reached. When hard acceleration is commanded, engine speed flares, corresponding torque consumed and vehicle’s acceleration decreases. Sun and Luo, among others, introduced the drivability notion for the description of the vehicle behavior during sudden acceleration. Drivability (Sun and Luo, 2012) is a subjective assessment and hard to measure. Figure 3 shows four objective parameters which have obvious relations with CVT drivability assessment. In Figure 3 to is the moment when the throttle was opened. The acceleration decreased at first reached its minimal value \( a_{\text{min}} \) and began to rise. The four objective parameters related to drivability assessment were: acceleration delay \( \Delta t_1 = t_1 - t_0 \), duration of acceleration \( \Delta t_2 = t_2 - t_0 \), acceleration reduction \( \Delta G = a_0 - a_{\text{min}} \) and a peak value of acceleration \( G_{\text{max}} \). If the values of \( \Delta t_1, \Delta t_2, \Delta G \) are very small and \( G_{\text{max}} \) is high the assessment of drivability is good. These parameters are strong related to gear ratio rate.

![Figure 3: Objective parameters for drivability evaluation (Sun and Luo, 2012)](image)

The CVT vehicles drivability is the result of gear ratio rate. It is impossible to improve the drivability by conventional control because gear shift rate depends directly on throttle position and outer resistance torque. To solve the problem described it is necessary to develop the actuator which makes the ratio change possible independently of engine revs. Moreover, the actuator that was worked out should operate according to the defined control algorithm to elaborate the control algorithm obtaining powertrains characteristics is necessary. The test stand for receiving such types of characteristics is presented in later part of the article.

**Test Stand**

Figure 4 shows the test stand. The test stand consists of: engine with CVT gearbox, flywheel, water brake, engine equipment and apparatus for measuring following variables: the engine torque, angular velocity of the engine, angular velocity of inner and outer shafts of CVT, the driven wheel winding radius and fuel consumption. The winding radius was measured using the optical sensor. The sensor’s signals were recorded with the frequency of 100 Hz, using the analog-to-digital converter coupled to the PC. The driving pulley winding radius was calculated on the base of the belt length and the pulley’s centers distance. The flywheel was used to simulate the conditions during the start of the vehicle.
The experiments were carried out using complete drivetrain of the TGB 101S scooter which is propelled by a two-stroke, one-cylinder gasoline engine. The engine displacement is 49 cm$^3$ and maximal power is 3.6 kW at 7500 revs/min. The weight of the empty vehicle is 81 kg.

The results of tests are shown in Figure 5. In popular motorbikes and scooters with two stroke engine the two minimum of brake specific fuel consumption (BSFC) very often appeared. The same situation was noticed for the tested engine. Due to this engine characteristic, it is very difficult to approximate OOL (optimal operation line) line by using a simple function of engine revs.

When the characteristics from Figure 5 and 6 are used the real operation line (ROL) can be designated. The real operation line (ROL) is presented in Figure 7 (spotted line). The optimal operation line can also be defined (Figure 7). The real operation line is beyond engine high-efficiency area. A big difference between these lines can be observed. Maximum scooter velocity is reached by 8500 rev/min of the engine. The BSFC value at this point is very high. This big difference is the consequence of a mechanical governor employed.

Figure 4: Test stand view and scheme

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Figure 5: BSFC characteristics and external characteristic (yellow line) of the test engine

Figure 6 shows the quasi-static dependence between engine revs and scooter velocity for the tested CVT. As it can be seen the mechanical regulator starts the gear ratio change at about 6800 revs/min. This value corresponds with the BSFC minimum.

Figure 6: Quasi-static characteristic of the tested CVT

As it is shown in Figure 7, the value of resistance power for the maximum scooter velocity is about 1.6 kW. In fact, it can be performed with a partly opened throttle and a lower engine speed. Consequently, the BSFC value is much advantageous. To ensure the tracking of the OOL the special actuator system has to be employed. However, the tracking of OOL influences the scooter drivability due to the lack of its dynamic response. Since drivability is also important for ratio control design, the special strategy should be worked out for an efficient combination of fuel economy and drivability.
It seems that when TGB 101S scooter is considered the procedure is simple for OOL. The change of the axial gear ratio is sufficient for obtaining maximum velocity. Unfortunately, the change of a gear ratio to achieve the maximum scooter velocity at lower rpm can cause a noticeable decline in the scooter’s performance. The required power scooter drive system to reach its limited maximum velocity is very often 30-40% lower than the maximum power of the employed engine. It allows the designers of a scooter drive system to take into account the possibilities of application of the partly opened throttle for maximum scooter velocity. Introducing electromechanical actuators can improve the drive system efficiency because gearbox ratio is no longer a function of engine revs. The ratio control strategy of CVT should ensure slight lowering of scooter performance together with simultaneous significant lower fuel consumption. The way of the exploitation of such a vehicle is different from the exploitation of other vehicles. These vehicles usually move with the wide opened throttle (WOT) from small velocity to maximum velocity. This way of scooter motion requires taking it into consideration when the ratio control strategy of CVT is evaluated. In this case, to decrease fuel consumption, the engine revs corresponding to maximum scooter velocity should be lowered to 6800 revs/min.

To obtain this value of engine revs, the gear ratio should be lowered by about 20%. The value of BSFC is the smallest at these revs. As the power of an engine at lowered revs exceeds the power calculated from the scooter’s motion resistance the position of engine throttle has to be diminished. In another case, the scooter’s velocity will exceed permitted velocity. The engine work with partly opened throttle will be sufficient. Partly opened throttle causes further lowering of fuel consumption. Employing an electric throttle is essential here.
To avoid the excessive slip in the CVT gear during electromechanical control the gear ratio rate tests in conventional CVT were performed. The maximum gear ratio rate values were assumed as a limit for gear ratio rate in electromechanical control. The results are shown in fig 8. The maximum value of gear ratio rate in both directions under conventional control is about 0,7 (1/s).

![Figure 8: Gear ratio (dotted line) and gear ratio rate (continuous line)](image)

For powertrain tested the maximum gear ratio rate that was measured caused a small diminishing of engine revs (Figure 9).

![Figure 9: Throttle position, revs of the engine, revs of the clutch, revs of outer CVT shaft during rapid acceleration](image)

As shown in Figure 9 drivability for tested CVT vehicle is acceptable. The main aim of introducing the new actuator and new control strategy is to diminish fuel consumption. If gear ratio rate for new control strategy is lower than the measured one, the decrease of drivability is not to expect. The description of the realized electromechanical actuator is attached in Appendix 1.

**SYSTEM MODELING**

Taking into account peculiar characteristic of exploitation of these types of vehicles the following control algorithm may be suggested (Figure 10). Engine revs are constant during vehicle acceleration in the chosen point of engine characteristic. When the vehicle reaches
maximum velocity, the angle of throttle opening decreases to make the motion resistance power balanced with the engine power. To keep the engine revs constant the gear ratio rate will be applied.

Figure 10: Suggested control strategy

To analyze the suggested strategy, the mathematical model of drive line was prepared (Grzegożek and Szczepka, 2012). The drive line is modeled only in its longitudinal behavior, and no drive train elasticities are taken into account. The efficiency of drive line is assumed as equal 1. Figure 11 shows a sketch of this system.

Figure 11: Drive line structure

The gear ratio of the CVT is defined by:

\[ i(t) = \frac{\omega_2}{\omega_1}. \]

The equations of drive trains motion are the follows:

\[ M_r - M_1 = J_i \frac{d\omega_1}{dt}, \]
\[ M_2 = i(t) \cdot M_1, \]
\[ M_2 = J_2 \frac{d\omega_1}{dt} + M_{op}. \]

The resulting drive trains dynamic equation is as follows:

\[
\frac{d\omega_1}{dt} = \frac{M_e - M_{op} \cdot i(t) - J_2 \cdot \omega_1 \cdot i(t) \cdot \frac{di}{dt}}{J_1 + J_2 \cdot i^2(t)},
\]

where:

- \( J_1 \) – inertia moment of an engine,
- \( J_2 \) – equivalent inertia moment of a scooter,
- \( M_e \) – engine torque,
- \( M_1 \) – input moment to CVT,
- \( M_2 \) – output moment out CVT,
- \( M_{op} \) – resistance moment of motion,
- \( M_R \) – resistance moment related to gear ratio rate.

It can be seen from above equation that there is the resistance torque related to gear ratio rate. The torque \( M_R \) is as follows:

\[ M_R = J_2 \cdot \omega_1 \cdot i(t) \cdot \frac{di}{dt}. \]

Figure 12 shows the resistance torque related to gear ratio rate during acceleration with mechanical CVT regulator.

![Figure 12: Resistance moment related to gear ratio rate (continuous line) during acceleration with mechanical CVT regulator.](image)

The principle of the control algorithm is that \( M_R \) is equal, during gear shifting to \( M_e - M_{op} i(t) \) to obtain the constant engine revs. When the engine operates in a selected point of its characteristic the decrease of fuel consumption can be obtained.

When in motion, city scooters equipped with small power engines have usually wide opened throttles. The control strategy should take into account the character of the drive...
which in fact is composed of extreme acceleration, drive with constant speed, breaking and reaccelerating until the maximum velocity is reached. This cycle is repeated several times; in each case the time in which the vehicle velocity is constant is different. Two values that define the run of the cycle are: the angle of opened throttle and the real gear ratio of CVT.

The start-on is usually independent of control systems. The centrifugal clutch is used for the start on. Subsequently, the vehicle is accelerated until the velocity corresponding to engine revs slightly higher than maximum torque is reached. Then the change of a ratio in CVT begins until the minimum ratio is reached. The balance of power on a scooter wheel with a power of motion resistance defines vehicle velocity. In many cases, this balance appears when the revs of the engine are much higher than revs of the engine for maximum power. These revs which are much higher are caused by the necessity of achieving the maximum vehicle velocity. The difference between engine revs of maximum torque and maximum power for typical engines which is very small is the reason why engine revs should be much higher than revs of engine’s maximum power. It is necessary for gaining a wider range of velocity changes. Unfortunately, these changes increase in vehicle fuel consumption. Moreover, the value of resistance power is on average 40% smaller than the maximum value of the power of the engines. When an actuator is mechanically steered the ratio value is the function of engine revs, and thus its value is defined by a regulator structure. When an electromechanical actuator is used simultaneously with electric throttle, the characteristics of the engine can be better employed, and the fuel consumption can be lowered. The use of electromechanical actuator should ensure the same or at least similar performance to the traditional version. As lowering fuel consumption requires smaller revs of an engine, it is indispensable to lower total gear ratio of a vehicle. In order not to change the drive power too much the vehicle acceleration should be done with possibly the highest value of engine torque. When the rate of the gear ratio change is controlled, the whole process of acceleration can be performed with constant engine revs as it appears in revs of an engine maximum torque. In such cases, even when the total gear ratio is lowered, it should ensure effective performance. The control strategy algorithm is presented in Figure 13.

![Figure 13: Control strategy algorithm](image)

The results of applying this control method to the mathematical model of the drive train are presented in Figure 14 and 15. As it can be seen in Figure 14 the performance of vehicles is almost the same. The characteristics shown in Figure 15 present that when the velocity is 45 km/h, and this control strategy is used, specific fuel consumption decreases by about 20% when compared with traditional method. As it was stated before the velocity of these types of vehicles is close to 45 km/h. Such lowering of specific fuel consumption may also be considered as fuel consumption per hour.
THEORETICAL ANALYSIS OF FUEL CONSUMPTION ACCORDING TO IDC

Small scooters, motorcycles and other vehicles of that type are driven in town in a specific way where extreme acceleration and braking are dominating. This kind of driving was described as Indian Drive Cycle (Chaudhari, 2004) in Asian countries, where the number of scooters and motorcycles exceeds 42 million.

The mathematical model, which was worked out, was used for the analysis of fuel consumption of the scooter equipped with CVT by the suggested research procedure. The course of the run of the engine torque as well as fuel consumption per hour in revs function were approximated by a polynomial. Moreover, additional function courses were formulated to present the previously described parameters in the function of an opening angle of a throttle. In Figure 16, the run of fuel consumption per hour for a different opening angle of throttle was presented and compared to the runs measured in test stand (spotted line). These functions were used in motion simulation for the scooter equipped with original as well as the optimum system of gearbox control.
In optimization case, the run of gear ratio rate is controlled by the relevant algorithm that ensures proper engine functioning at the highest efficiency, whereas in original characteristics the gear ratio rate is changed to present the run of the analyzed cycle in the best way. As it was in the case of real vehicle research according to the road cycle, the angle opening of throttle was manually performed. As shown in Figure 17 the velocity runs were with the considered cycle.

In IDC cycle 15% reduction of fuel consumption was acquired. Moreover, the algorithm controlling optimized gear ratio rate makes the gear ratio rate of the maximum rate not larger than in original solution possible.
Applying optimum control unit gives significant savings during the scooter motion with maximum velocity (45 km/h) if is compared this with fuel consumption of scooter with the conventional control unit. It is shown in Figure 18.

![Figure 18: Scooter velocity in time function comparison between original and proposed control strategy](image)

The scooter acceleration is a little higher than acceleration with the conventional unit. When a scooter is driven with maximum acceleration and velocity (range of test distance 1 km) its fuel consumption is:

a) 4.58 l/100km for a scooter equipped with the original control unit,

b) 2.32 l/100km for a scooter equipped with the optimized control unit.

In this case, fuel consumption decrease exceeds 50%. In the above runs, the minimum velocity is 5 km/h. It is due to two facts:

- control unit starts to gear change at the vehicle velocity that exceeds 10 km/h,
- centrifugal clutch (which is not tested during this research) influences the vehicle driven with less than 5km/h.

**CONCLUSIONS**

- The main novelty of this work is the use of the CVT gear ratio rate for engine revs control. In this way, the fuel consumption decrease is obtained.
- To achieve optimal fuel consumption and good drivability of a scooter with CVT gearbox the special control unit, independent of engine revs, electronic throttle as well as appropriate control strategy should be used.
- The gear ratio rate is one of the parameters essential for vehicle performance. So far, the main emphasis was placed on the influence of gear ratio rate on vehicle acceleration values. The use of the proper gear ratio rate for gear changing may help in ensuring vehicle acceleration while, at the same time, securing optimal engine operation.
- The gear ratio rate value for the realized acceleration cycle did not differ much from the gear ratio range for the conventional system. In this way, the similar value of a belt slip and its efficiency was ensured.
- Fuel consumption in the analyzed cycle decreases by 15% when compared to conventional steering. It should be considered a great achievement.
The application of electromechanical control unit, making gear changing as well as gear ratio rate possible, enables to drive the vehicle with revs supporting optimum fuel consumption.

REFERENCES


Bonsen B and Steinbuch M and Veenhuizen P A 2005 CVT ratio control strategy optimization *Int. J. of Vehicle Design*

Cammalleri M. A new approach to the design of a speed torque controlled rubber V-belt variator *Proc. IMech E* vol 219 part D 2005

Chen T F and others 2000 Design Considerations for Improving Transmission Efficiency of the Rubber V-belt CVT *Int. J. Vehicle Design* vol 24 No 4

Chaudhari M K 2004 Motor Cycle Emission Control in India Asian Vehicle Emission Control Conference

Grzegożek W Szczepka M 2012 An Attempt of Fuel-Optimal Control of Scooter CVT Powertrains *Czasopismo Techniczne Mechanika* R.109 z.5-M pp 101-107

Jantos J 2001 Control of the Transmission Ratio Derivative in Passenger Car Powertrain with CVT *SAE Technical Paper* 2001-01-1159


Smetser H.” Implementation of a Suzuki Electronically controlled Continuously Variable transmission in a Formula Student racecar” Bachelor Final Project Eindhoven University of Technology, October 2008


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APPENDIX 1

The proposed actuator presented in Figure 19 is propelled by a DC 12V electric motor with a worm gear. The electric motor propels the cam with an Archimedes spiral outline which in turn acts with the roller pusher. The roller pusher is fixed to the engine crankshaft by an axial bearing and a bar. The movable CVT pulley is fastened to a three-arm carrier. The carrier is set on pivots fixed to a non-movable CVT pulley. The carrier is also joined to the actuator cover. The cam turn leads to a displacement of the cover as well as its carrier and in this way the axial position of the movable CVT pulley is changed. The electric motor power is about 100 W and the maximum axial force generated by the actuator is about 400 N.

Figure 19: View of the proposed actuator