ELECTRONIC CONTROL OF CONTINUOUSLY VARIABLE TRANSMISSIONS.

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The technology of a Continuously Variable Transmission (CVT) has been around for many years now. While its reliability, durability, efficiency and controllability have been problems in the past, all but controllability have been greatly improved recently. In this paper, we present an electronic control strategy for a mechanical CVT that consists of a wet multi-plate clutch and a pushbelt variator. Using a virtual engineering approach, the ultimate powertrain control problem (optimal shifting) is translated in a set of hierarchical control problems. To realize optimal shifting, the concept of variogram control is implemented by two PI-controllers. Next, the characteristics of the variator and the clutch are taken into account, by an explicit static feedforward. PI-controllers are added, to deal with modeling errors and uncertainty. Finally, the behavior of the electro-hydraulic actuator is optimized, by redesigning the valves and by adding actuator control.

I. INTRODUCTION

In this paper, we present a virtual engineering approach for electronic control of a CVT with a wet multi-plate clutch and a pushbelt variator. A topdown approach is used to translate the powertrain control problem into a set of hierarchically related subproblems. Once all components and their interactions are modelled, these models are used to design controllers for all subsystems. Major design advantages of our approach are:

- The proposed control strategy is fairly general and can easily be adapted to each car/engine combination that contains a CVT.
- The topdown approach leads to a controller with an hierarchical structure, where each subcontroller controls a specific subsystem.
- Compared to the classical prototyping methods, our model based controller design method is cheaper and faster, so more alternatives can be evaluated, for lower cost.
- Unlike the classical mechanical and hydraulical control systems, our control parameters can be changed fast and for lower cost.

Besides substantial design advantages, our approach also leads to an increase in performance of the powertrain and its subsystems:

- Variogram control makes it possible to implement an optimal drive-concept.
- Due to computer control, a throttle dependent stall rotation speed can be implemented, which permits to realize a smooth and fast drive away behavior.
- Taking into account the highly non-linear characteristics of the variator and the clutch, the implemented controller will be less conservative then the classical hardware controllers.
- The implementation of actuator control drastically reduces the influence of the oil temperature and the engine rotation speed.

This paper is organized as follows: in the first section, two optimal drive-concepts are presented. To realize an optimal drive-concept, the idea of variogram control is introduced. Next, it is shown how the characteristics of the wet multi-plate clutch and the pushbelt variator can be taken into account. Finally, the influence of the actuator on the powertrain control loop is investigated. The different controllers are combined in a single controller with an hierarchical structure. Closed-loop experiments demonstrate that the proposed algorithm works efficiently.

OPTIMAL DRIVE-CONCEPTS

In practice, combustion engines are characterized by two curves. The torque-speed characteristic relates the delivered engine torque T_e to the engine rotation speed N_e and the throttle position α (see figure 1), while the BSFC diagram represents the brake specific fuel consumption b_e as a function of the engine rotation speed N_e and the delivered engine torque T_e (see figure 1). For each throttle position α^*, the following optimal engine rotation speeds $N_e(\alpha^*)$ can be derived:

$$N_{e_P}(\alpha^*) \quad = \quad \text{the speed of maximal power (sport mode)} \qquad (1)$$
$$N_{e_b}(\alpha^*) \quad = \quad \text{the speed of minimal consumption (economy mode)} \qquad (2)$$

The optimal operating lines, corresponding to these different optimal speeds, are shown in figure 1. Notice that only a CVT can operate along these lines, since continuous shifting is required to track them. To obtain optimal shifting, the curves defined by equations (1)-(2) must be plotted as a function of the vehicle speed (so-called variogram). Because of comfort specifications, these curves need to be smoothed. The final variogram (for economy mode) is shown in figure 2.

POWERTRAIN LEVEL

At the powertrain level, a car with ideal components is considered (see figure 3). First, an accurate physical model is derived. Next, it is shown how the variogram can be used to implement an optimal drive-concept.

Modeling

The clutch is modelled as a modulated friction element. The torque T_{cl} that is transmitted through the clutch is given by:

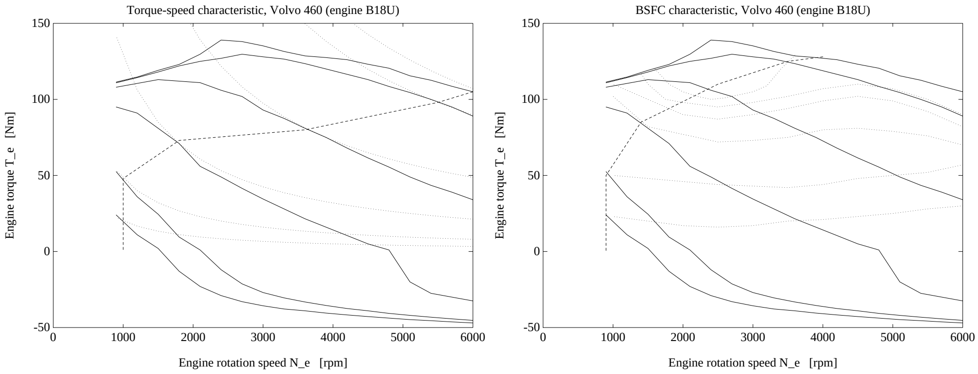

Figure 1: *Left: the experimentally determined torque-speed curves $T_e(N_e)$ for various throttle positions α (full lines) and the optimal operating line for the sport mode (dashed line). Right: the BSFC characteristic is obtained by adding lines of constant brake specific fuel consumption (dotted lines) and the optimal operating line for the economy mode (dashed line).*

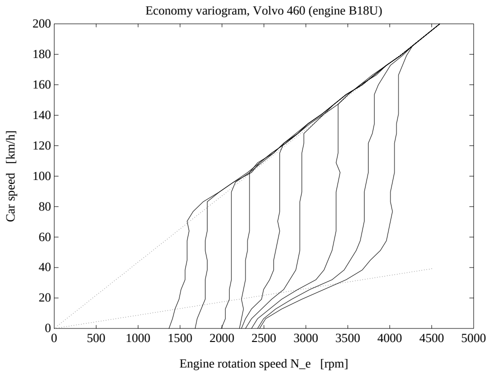

Figure 2: *The economy variogram for a Volvo 460 (engine B18U) with a CVT (full lines). The minimal and maximum transmission ratios are indicated also (dotted lines).*

$$T_{cl} = \mu_{cl} Q \tag{3}$$

The friction coefficient μ_{cl} mainly depends on the clutch slip $s = N_e - N_p$, the difference between the engine rotation speed N_e and the primary rotation speed N_p. Q is determined by the oil pressure in the clutch piston.

The variator and the final drive reduction are modelled by a single modulated transformer with a variable transformation ratio $i = i_x i_d$. The continuously variable ratio of the variator is indicated by i_x. i_d is the fixed ratio of the final drive reduction. The ratio i varies between a minimal ratio i_{min} and a maximal ratio i_{max}. It holds that:

$$N_p = N_f i \tag{4}$$

$$T_p = \frac{T_f}{i} \tag{5}$$

with T_p the primary torque and T_f the torque applied to the wheel shaft.

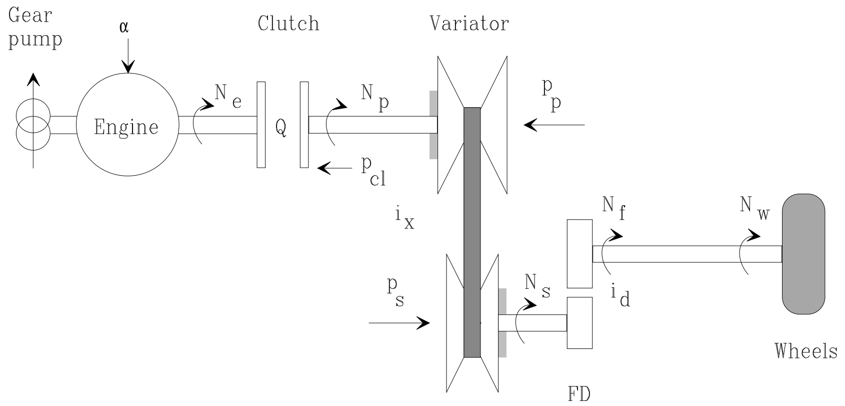

Figure 3: *The powertrain contains a mechanical CVT with a wet multi-plate clutch and a metal pushbelt variator. The variator ratio i_x can be varied by changing the pressures p_p and p_s. The clutch control signal Q is determined by the pressure p_{cl}.*

Control design

As shown in the previous section, the variogram represents an optimal drive-concept. For a powertrain with a CVT, we propose the control configuration of figure 4. Table 1 summarizes the proposed strategy.

Phase	Computation of Q	Computation of i_x
Clutch phase $\frac{N_{p,m}}{N_{w,m}} > i_{max}$	From $N_{e,ref} - N_{e,m}$	Keep $i = \frac{i_{max}}{i_d}$
Variator phase $\frac{N_{p,m}}{N_{w,m}} < i_{max}$	From $s_{ref} - [N_{e,m} - N_{p,m}]$	From $N_{e,ref} - N_{e,m}$

Table 1: *The proposed control strategy for powertrain control.*

During the clutch phase (dotted lines in figure 4 are active), $\mathrm{PI_{cl}}$ controls the clutch, to realise the desired shifting traject, while the variator ratio remains constant. During the variator phase (full lines are active), the variator controller $\mathrm{PI_v}$ takes care of variogram tracking. To avoid energy losses, $\mathrm{PI_{cl}}$ is used to keep the clutch slip small. An hysteresis element, driven by the measured clutch slip $s_m = N_{e,m} - N_{p,m}$, is used to generate the reference clutch slip s_{ref}. Smooth switching between both phases is guaranteed if the initial reference clutch slip is well-chosen.

POWERTRAIN COMPONENT LEVEL

In this section, it is shown how the characteristics of the wet multi-plate clutch and the pushbelt variator can be taken into account. First, reduced static models of both systems are derived. Next, these models are inverted to obtain static feedforward controllers.

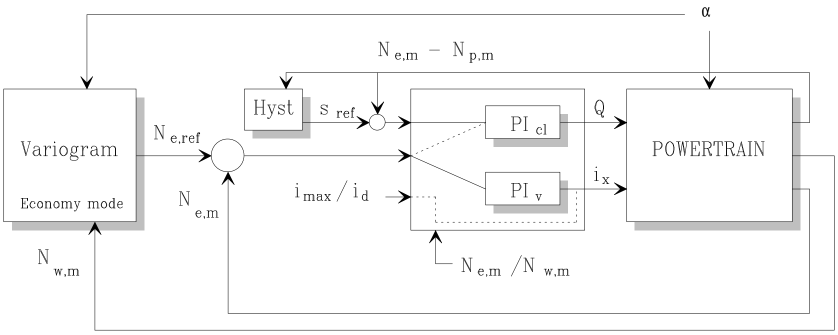

Figure 4: *Variogram control: the reference engine rotation speed $N_{e,ref}$, which is obtained from the measured wheel speed $N_{w,m}$, the measured throttle position α and the desired drive-mode, is compared with the measured engine rotation speed $N_{e,m}$.*

The wet multi-plate clutch
The torque transmitted through the clutch is proportional to the pressure in the piston (see figure 5).

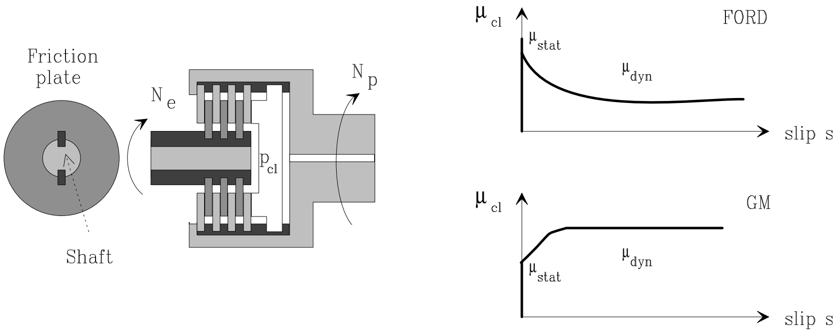

Figure 5: *A cross section and a longitudinal section of the wet multi-plate clutch (l). To achieve smooth coupling, the friction coefficient μ_{cl} in wet multi-plate clutches is used as a real design parameter (r).*

Only a rough static model is derived. For a fixed engine rotation speed N_e and primary rotation speed N_p, the clutch equation determines the torque that is transmitted through the clutch:

$$T_{cl} = n\mu_{cl}\left[p_{cl} - \frac{F_0}{A_{cl}}\right] A_{cl} R_m \tag{6}$$

with p_{cl} referring to the clutch pressure. Table 2 explains the meaning of the parameters. From equations (3) and (6), it follows that the clutch control signal Q is determined by:

$$Q = n\left[p_{cl} - \frac{F_0}{A_{cl}}\right] A_{cl} R_m \tag{7}$$

The pushbelt variator
The pushbelt variator (see figure 6) may be considered as the heart of the CVT. The circumferential force of the driving torque T_p at the primary pulley is transmitted by a flexible medium to the secondary pulley.

Symbol	Parameter
F_0	preload of the spring
A_{cl}	pressure surface of the clutch piston
R_m	mean friction radius of a plate
n	number of active friction surface pairs

Table 2: *Parameters in the clutch equation.*

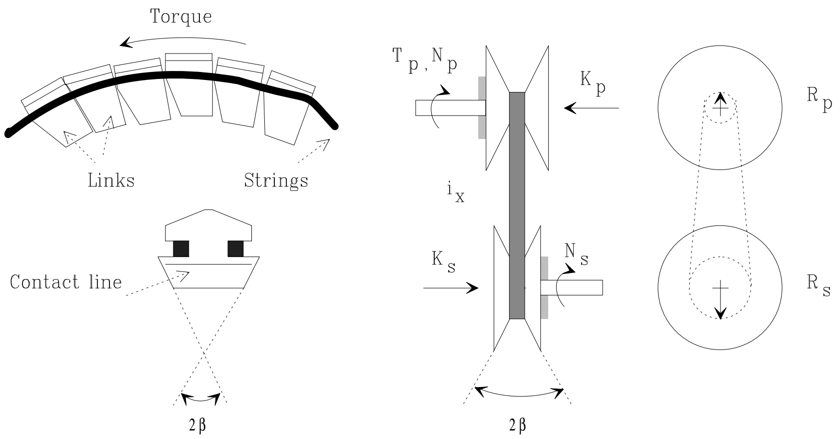

Figure 6: *To transmit power, a pushbelt variator uses a belt that exists of metal links and metal strings. Power is transmitted by pushing, through the line contact of neighbouring links (l). The variator ratio $i_x = R_s/R_p$ can be changed by the pinch forces K_p and K_s (r).*

In steady state, the ratio between the secondary radius R_s and the primary radius R_p is equal to the variator ratio i_x. This ratio is adjusted by axial shifting of one of both conical pulley sheaves. For a fixed ratio i_x, integral equations can be used to compute the static behavior of the variator. In principal, one describes the static equilibrium of the pinch forces K_p and K_s, the belt stresses (determined by the driving torque T_p), the centrifugal forces on the belt mass (determined by the primary rotation speed N_p) and the dry friction between belt and pulleys. By solving these implicit equations in i_x, it is possible to obtain discrete values of the following multivariate function:

$$i_x = i_x(\frac{K_p}{K_s}, N_p, T_p) \tag{8}$$

The static values of the resulting pinch forces K_p and K_s are determined by the static equilibrium at each pulley:

$$\begin{aligned} K_p &= p_p A_p && (9) \\ K_s &= p_s A_s + d_s[R_{s,0} - R_s] && (10) \end{aligned}$$

Table 3 explains the meaning of the used parameters. The rotation speeds N_p and N_s are also related to the radii R_p and R_s by:

$$i_x = \frac{N_p}{N_s} = \frac{R_s}{R_p} \tag{11}$$

Symbol	Parameter
A_p	pressure surface of the primary cylinder
A_s	pressure surface of the secondary cylinder
d_s	equivalent stiffness of the secondary spring
$R_{s,0}$	reference radius of the secondary pulley

Table 3: *Parameters in the static equilibria of the pulleys.*

Furthermore, for a constant belt length L and a fixed shaft distance d, it holds that:

$$L = 2\sqrt{d^2 + [R_p - R_s]^2} + \pi[R_p + R_s] \tag{12}$$

From equations (9) and (10), it follows that the primary pressure p_p and the secondary pressure p_s can be used as control signals for the variator. To avoid the belt from slipping on the pulleys, two additional constraints are taken into account:

$$K_{p,min} = \frac{f_r |T_p| \cos(\beta)}{2\mu_v R_p} \leq K_p \tag{13}$$

$$K_{s,min} = \frac{f_r |T_s| \cos(\beta)}{2\mu_v R_s} \leq K_s \tag{14}$$

A safety factor $f_r \geq 1$ is used. The angle β of the conical sheaves is indicated on figure 6. The dry friction coefficient between the belt and the pulleys is denoted by μ_v. Equations (8)-(14) are used to generate the pressures p_p and p_s. Since only static models were considered, additional feedback control is required. The speed error $N_{e,ref} - N_{e,m}$ is used as an input for the feedback controller.

ACTUATOR LEVEL

As shown in the previous section, three hydraulical signals are required to control the wet multi-plate clutch and the pushbelt variator. An electro-hydraulic actuator is used to generate the desired signals. A typical loop is shown in figure 7.

Modeling

Bond graph modeling is used to describe the static and dynamic behavior of all hydraulic servo valves in the valve-body.

Although extensively used for valve analysis and design, the bond graph models are too complex for control design. Within the range of interest (30 bar output pressure), the dynamics are accurately described by a linearized system:

$$\frac{p_s(s)}{p_{servo}(s)} = \frac{K(N_e,T)(s - z(N_e,T))}{s^2 + \alpha(N_e,T)s + \beta(N_e,T)} \tag{15}$$

where all parameters are smooth functions of the engine speed N_e and the temperature T.

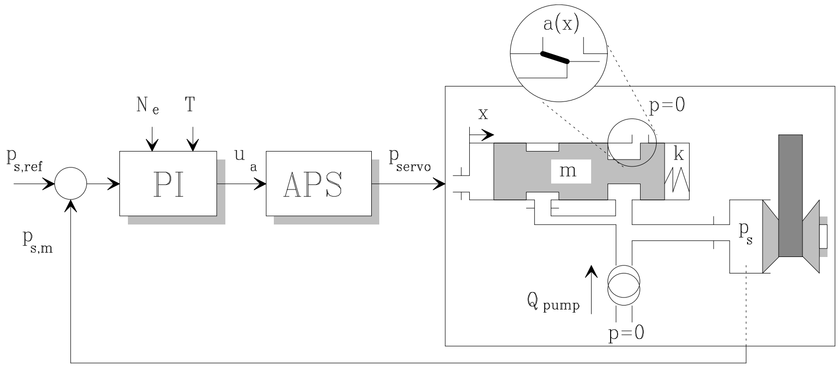

Figure 7: *The secondary circuit of the electro-hydraulic actuator: an ideal analogue pressure source (APS) drives the hydraulic servo valve (three way valves in PWM-mode are used in cars, for reliability and cost constraints). The servo pressure p_{servo} determines the secondary pressure p_s. The pump flow Q_{pump}, delivered by an engine driven gear pump, is proportional to the engine speed. The oil temperature T and the pump rotation speed N_e are measured, slowly varying disturbances. An (N_e, T)-scheduled PI-controller is used to reduce their influence on the secondary pressure p_s.*

Control design

The speed and temperature dependency of the valve-body pressures is the main reason for doing actuator control. Here, we will use Ziegler-Nichols-like PI-control.

The following equations are used to tune the PI-controller in a set of operating points:

$$K_p(N_e, T) = \left[\frac{0.9}{R(N_e, T)L(N_e, T)}\right] p_f(N_e, T) \tag{16}$$

$$T_i(N_e, T) = [3.3L(N_e, T)]\, i_f(N_e, T) \tag{17}$$

A precise definition of the reaction rate R and the lag L is given in [Ziegler, 1942]. To meet all design specifications, the factors $p_f(N_e, T)$ and $i_f(N_e, T)$ are introduced. It is shown in [Vanvuchelen, 1993] that a well-tuned (N_e, T)-scheduled PI-controller guarantees robust performance over a wide range of operation ($20\,^o\mathrm{C} \leq T \leq 90\,^o\mathrm{C}$, $1500\,\mathrm{rpm} \leq N_e \leq 4500\,\mathrm{rpm}$).

COMBINED RESULTS

In this section, all previously derived models and controllers are combined in a single hierarchical model and a single hierarchical controller. Closed-loop experiments demonstrate that the proposed model based approach works satisfactorily. The applied throttle signal and the slope of the road are shown in figure 8. Optimal driving is guaranteed by variogram control (see figures 8 and 9). The controller computes the clutch control signal Q and the variator control signal i_x (see figure 10). The hydraulical signals p_{cl}, p_p and p_s are required to realize Q and i_x (see figure 11). The electro-hydraulical actuator is assumed to be ideal.

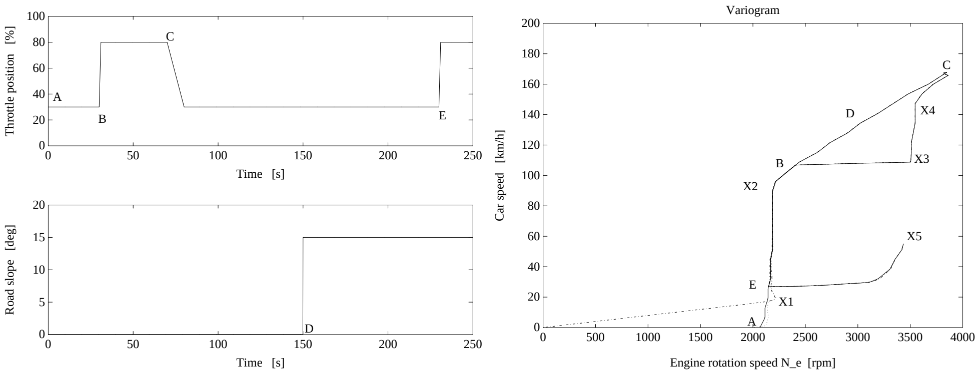

Figure 8: *Inputs (left): throttle position and slope of the road. Outputs (right): the desired variogram (full line) and the realized variogram (dotted line).*

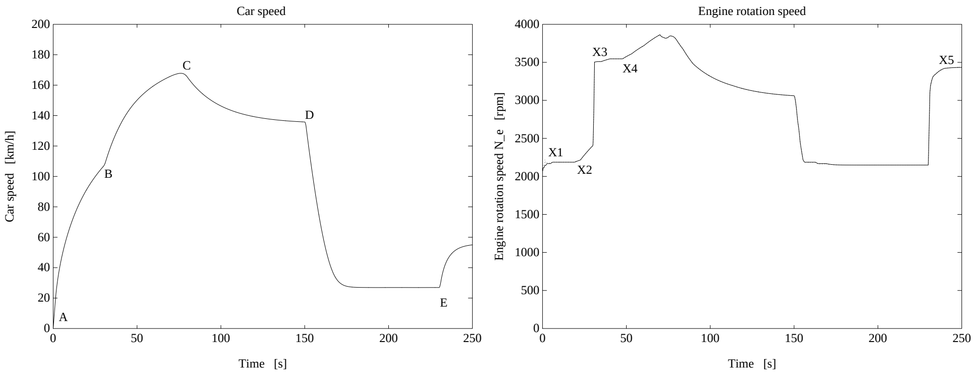

Figure 9: *The car speed (left) and the engine rotation speed N_e (right).*

CONCLUSIONS

A virtual engineering approach is used to tackle the powertrain control problem of combustion engine cars. It is shown that, due to the engine characteristics, an optimal drive-concept can only be realized with a CVT.
We presented an electronic control strategy. A topdown approach is proposed to translate the powertrain control problem into a set of hierarchically related subproblems. Bond graph modeling, static modeling and linear identification are used to obtain models of the powertrain, the wet multi-plate clutch, the pushbelt variator and the electro-hydraulic actuator. Computer Aided Control System Design (CACSD) methods are then used to design controllers for all subsystems. Closed-loop experiments demonstrate that the proposed approach works well.

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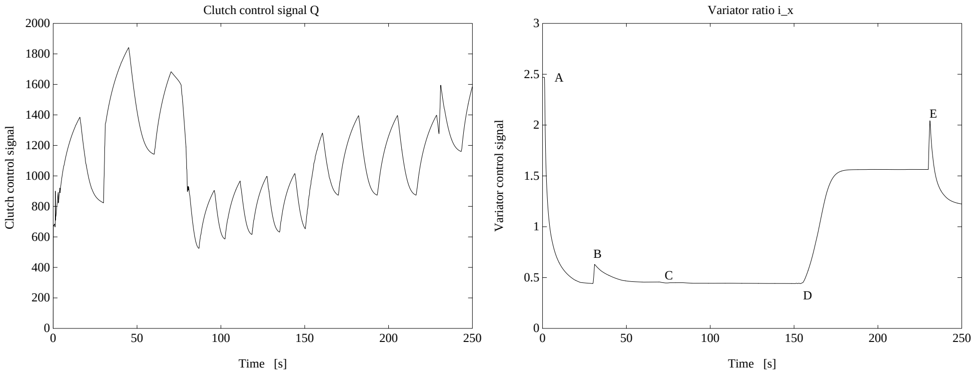

Figure 10: *The clutch control signal Q oscillates, since the hysteresis block is active during the variator phase (left). The variator shifts continuously (right).*

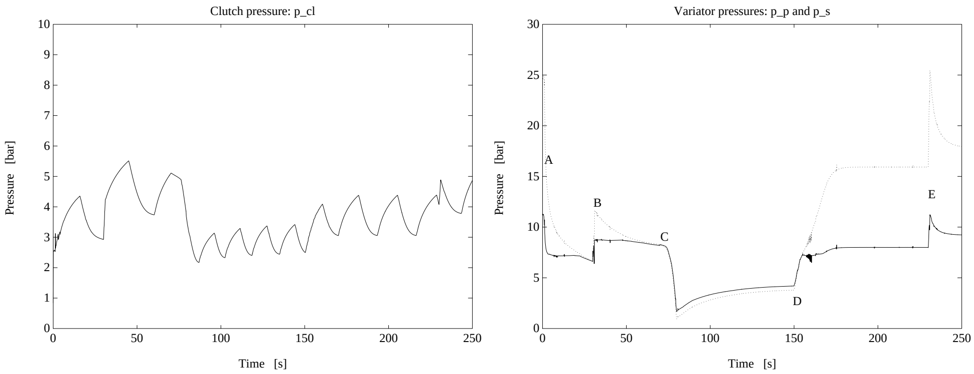

Figure 11: *The applied clutch pressure p_{cl} (left). The applied primary pressure p_p (full line) and the applied secondary pressure p_s (dotted line) are shown also (right).*

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