Model-Based Variator Control Applied to a Belt Type CVT

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The purpose of variator control is to take care of both primary speed and clamping force control by excitation of a hydraulic system generating the pulley clamping forces. Increasing demands on fuel consumption and driveability call for a continuous improvement of variator control performance. Since the system behaviour is nonlinear and shows a considerable level of interaction, a model-based approach is chosen to improve the results compared to traditional control strategies. This paper presents the basic principles of a model-based variator control strategy and the results of its application in a CVT-vehicle.

Keywords / push belt CVT, efficiency, driveability, model-based control, electronic variator control

1. INTRODUCTION

To achieve optimal fuel economy and optimal driveability of a CVT, accurate and fast control of the primary (variator input) speed is a prerequisite. In addition, the clamping force needs to be continuously adjusted to an optimal level to ensure slip-free operation without unnecessarily increasing the power losses in the CVT. The target of variator control is to take care of both primary speed and clamping force control by excitation of a hydraulic system generating the pulley clamping forces.

Since the behaviour of hydraulics and variator is nonlinear and shows a considerable level of interaction between clamping force and ratio control, a model-based approach is chosen to improve the results compared to the standard, empirical and non-coupled control strategy.

This paper presents the basic principles and structure of model-based Co-ordinated Variator Control (CVC®) and the results of its application in a CVT-vehicle. The CVC structure is such that it can easily be adjusted to work with any hydraulic layout.

In Section 2 the targets for the performance of the variator controller are formulated. Section 3 describes the variator control system under consideration, its model and the model verification. The model-based variator controller design is discussed in Section 4. In Section 5 the measurement results of application of the CVC in a CVT-vehicle are discussed. Finally, Section 6 presents the conclusions and an outlook to future activities.

2. VARIATOR CONTROL TARGETS

This study is part of the EcoDrive project which aims at improving the fuel consumption of mid-class CVT-vehicles, without compromising driveability and performance [1]. Based on this general objective, variator control performance targets are formulated.

2.1 Primary Speed Control Targets

Constant speed
For a given power demand at the vehicle wheels, the engine speed for realisation of optimal fuel economy can be derived from the engine's BSFC-map [2]. The line of optimum engine speed as a function of power demand is called the efficiency-line ('e-line'). The practical realisation of the e-line fuel consumption depends on the accuracy of the primary speed control in stationary situations.

Fig. 1 shows the results of simulations of the cumulative fuel consumption over an NEDC-cycle. The optimal (e-line) fuel consumption is chosen as a reference. The increase in fuel consumption, as a percentage of the e-line fuel consumption, is displayed for several primary speed offsets. The traditional primary speed control shows stationary primary speed errors of up to ±100 [rpm]. Based on the simulation results, the CVC accuracy target for primary rotational speed control was set to ±20 [rpm] in stationary situations. This corresponds to a maximum increase in fuel consumption of about 0.25 [%] as compared to the e-line fuel economy.

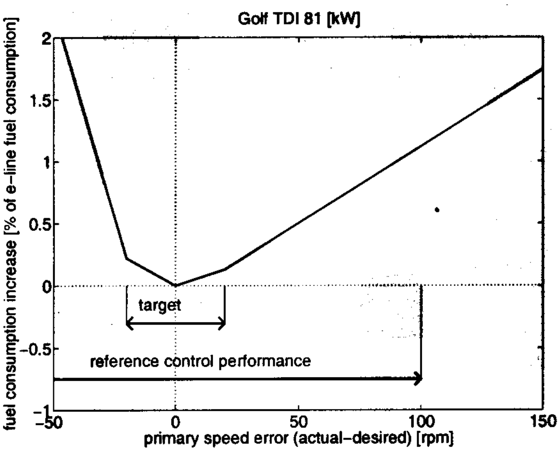

Fig. 1 Simulation results: increase in NEDC-cycle fuel-consumption as a function of primary speed error.

Speed transients

In most applications a trade-off exists between fuel consumption and driveability. When fuel consumption is minimised, driveability tends to decrease. Since this project aims at maintaining good driveability, co-ordinated steering of the engine torque and the primary rotational speed is used to improve driveability. To realise this, accurate tracking of the primary rotational speed trajectory is desired.

Besides the benefits of minimal fuel consumption in economy-mode, high primary speed control accuracy will also improve the CVT-performance when a high dynamic response is requested (*i.e.* with tip-mode programs, at a stage-shift in a two-stage CVT or with hybride drive lines).

2.2 Clamping Force Control Targets

In order to guarantee the torque capacity of the variator the clamping forces need to be higher than the slip-limit force. At known torque, a safety factor (over-clamping) of 1.3 based on this known torque will adequately prevent the belt from slipping. In vehicle applications, especially the torque peaks induced by the road (bumps, holes) are quite unpredictable. Therefore, at part-load increased safety is applied. This, however, decreases the CVT-efficiency. Studies show that a clamping force strategy that controls the safety at 1.3 of actual torque is a very effective way to improve fuel consumption. Simulation results, presented in [3], show a potential fuel consumption decrease up to 5 [%]. To realise this some design choices were made:

1. a torque-fuse operating at the secondary shaft
2. a hydraulic layout that enables a reduction of clamping forces.

In addition, the clamping force algorithm must be prepared:

- to control the 1.3 safety based on actual torque at the primary or the secondary pulley
- to control higher clamping forces when this is necessary to realise fast ratio shifts.

To further explain these demands, the variator control system will be discussed, followed by the in-depth explanation of the CVC.

3. THE VARIATOR CONTROL SYSTEM

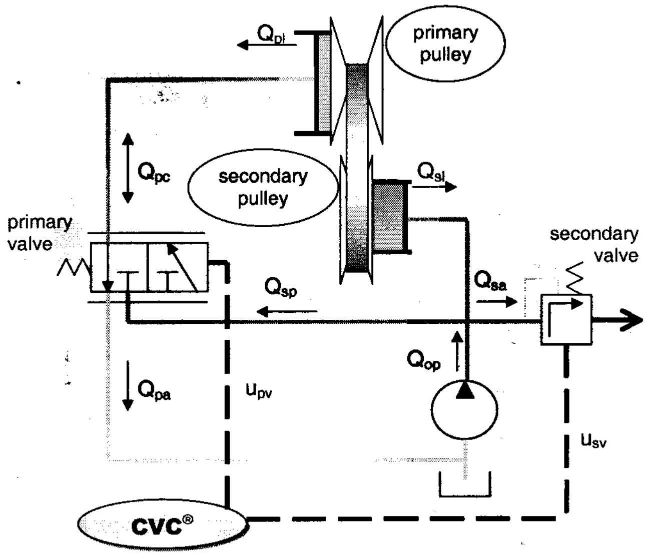

Fig. 2 The variator control system under consideration.

Fig. 2 shows the variator and hydraulics of a CVT, incorporating a Van Doorne metal V-belt. Both the torque capacity and the ratio change speed of the variator are influenced by the clamping forces applied to the primary (input shaft) and secondary (output shaft) pulley. These clamping forces are generated by a hydraulic system.

A pressure control valve directly controls the pressure (and consequently clamping force) in the secondary pulley, and is therefore called 'secondary valve'. A flow control valve ('primary valve') controls the flow to and from the primary pulley, influencing the primary pressure and clamping force level.

The pump, driven by the engine shaft, delivers the necessary flow directly to the secondary pressure circuit. The primary pressure circuit is fed from the secondary pressure circuit and returns to a low pressure level. The primary and secondary valve are individually controlled by solenoid valves.

3.1 Model Structure

For controller design the interest is in the relationship between the system inputs, the solenoid currents, and the system outputs, the clamping forces and the primary rotational speed. The variator control model describes this relationship. The model is built up out of three sub-models, describing the behaviour of respectively:

- the hydraulics
- the variator
- the drive line.

The model of the hydraulics and variator system will be explained below.

3.2 Modelling the Hydraulics

The primary circuit is described by a mass-balance, taking into account (flows indicated in Fig. 2):

- the flow through the primary valve either from the secondary circuit or to the lower pressure level (respectively Q_{sp} and Q_{pa})
- the volume change of the primary pulley caused by shifting
- a leakage flow from the primary circuit (Q_{pl}) and compressibility effects.

In a similar way, the secondary circuit is described by a mass-balance taking into account:

- the flow delivered by the pump (Q_{op})
- the flow through the secondary valve, from secondary circuit to auxiliaries (Q_{sa})
- the flow through the primary valve, from the secondary to the primary circuit (Q_{sp})
- the volume change of the secondary pulley caused by shifting
- a leakage flow from the secondary circuit (Q_{sl}) and compressibility effects.

3.3 Modelling the Variator

In the literature only a few approaches are available to describe the variator shift behaviour as a function of the clamping forces [4],[5]. A rather simple, experimentally determined shift speed model is described in [4]. A slightly adapted version of this model was used to describe the variator behaviour for CVC development:

$$\frac{di}{dt} = m(i) n_p F_s \left(kpks - \frac{F_p}{F_s} \right) \qquad (1)$$

Speed ratio i is defined as the ratio of the primary and secondary rotational speed (n_p/n_s). The ratio shift speed di/dt is described as a function of:

- the difference between kpks, defined as the clamping force ratio for holding a stationary speed ratio and the actual *applied* force ratio F_p/F_s
- the secondary clamping force level F_s
- the primary rotational speed n_p
- an experimentally determined factor m(i), which is assumed to depend only on speed ratio.

3.4 Model Verification

The model verification was carried out in several steps. As a start the variator shift speed model was verified on a variator test-rig. This allowed mapping of kpks and m(i) under well defined circumstances.

The over-all model (hydraulics, variator and driveline) was verified using one of VDT's demonstration vehicles, a Chrysler Voyager with P884 prototype CVT. The model was verified both in the time and in the frequency domain. Fig. 3 shows an example of measurement and model prediction results of a frequency response near Low and OverDrive (OD), in one plot. The model describes the system behaviour adequately.

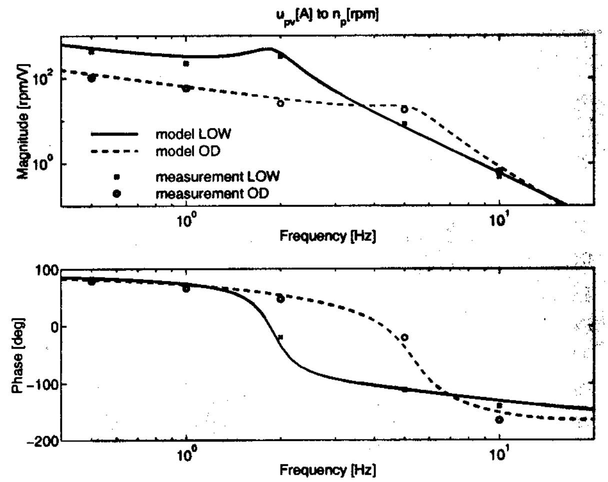

Fig. 3 Frequency response primary solenoid current to primary rotational speed near Low and OD.

4. MODEL-BASED VARIATOR CONTROLLER DESIGN

In this section the basics of the CVC design are discussed. Fig. 4 shows the general CVC structure. Only the main signal flow is indicated. The input of the CVC is the primary speed (or speed ratio) set-point. Based on this trajectory, the model-based controller determines the ratio and clamping force to be controlled and steers the hydraulics by means of the primary and secondary current. The measurement signals used to realise the control actions are not indicated in Fig. 4, these are e.g. primary and secondary speed, and secondary pressure.

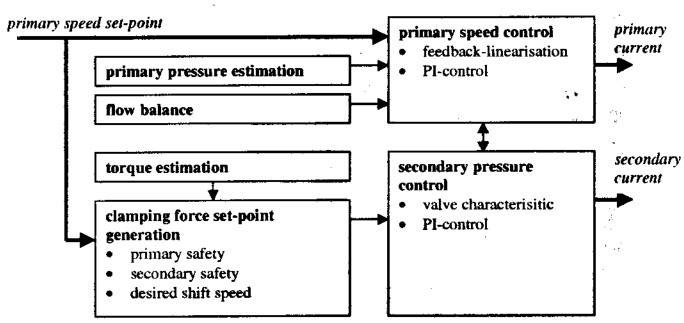

Fig. 4 The CVC structure.

The main CVC functions will be discussed in some more detail.

4.1 Model-based Primary Speed Control

The system behaviour from primary current to primary speed is highly nonlinear. This is illustrated by the verification measurement of Fig. 3: the transfer function of the relation depends on the operating conditions chosen (in this case Low or OD). Analysis of the system model can explain this behaviour. Consider a ratio change towards OD. To realise this, the primary valve must be opened to enable a flow from the secondary to the primary circuit. This flow causes the primary movable pulley to move in the OD direction, which causes the primary speed to decrease. Both the flow through the primary valve and the flow-ratio-primary speed relationship are nonlinear. The flow through the primary valve depends on the pressure difference over the valve. For the shift towards OD, this pressure difference is secondary minus primary pressure, which depend respectively on torque level and kpks. The flow-ratio relationship is geometrically determined and depends on the speed ratio. The ratio-primary speed relationship is also nonlinear, and depends on the secondary speed.

In traditional control, proportional control over the primary speed error is used to control the primary speed. With gain-scheduling, based on primary and secondary speed the influence of these quantities is taken into account. Since by means of gain-scheduling only a limited number of nonlinear effects can be taken into account, the primary control performance is limited and tuning is difficult. In the CVC all influences discussed above are taken into account by using a feedback-linearisation algorithm [6]. Its principle is that additional (measurement) signals are used to obtain a linear system behaviour as seen from the controller, see Fig. 5. One, easily tuneable, linear control law is now used for all operating conditions.

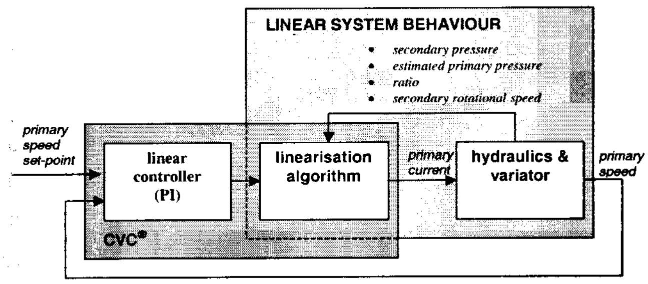

Fig. 5 The primary speed control algorithm.

Primary pressure estimation

For the linearisation algorithm the primary pressure is necessary. Since it is desired to avoid the use of a primary pressure sensor for reasons of costs, the primary pressure is estimated from measured ratio and secondary pressure, by means of the shift speed model (1).

Flow balance

In the linearisation algorithm the flow through the primary valve is calculated explicitly. This allows the shift speed to be balanced with the available oil flow. When flow shortage is detected, for example by monitoring the difference between desired and actual secondary pressure, shift speed limitation is established.

4.2 Model-based Clamping Force Control

The clamping force is controlled by means of the secondary valve. In the CVC there are three principles that determine the clamping force to be set:

- a secondary safety demand (stationary, normal shift speeds)
- a primary safety demand (fast shifts towards LOW)
- a desired shift speed demand (fast shifts towards OD).

All three of these demands can be translated in a minimal bound for the secondary clamping force. The maximum of these is translated in a secondary pressure set-point.

Stationary situations, normal shift speeds

The secondary safety demand directly provides a desired secondary clamping force.

Fast shifts towards Low

For shifts towards Low, the primary clamping force is reduced, but should not reach the minimal primary force level that guarantees the torque capacity. From the minimum allowed primary clamping force and a desired shift speed, the minimum secondary clamping force is derived by use of the variator shift speed model (1). As an example, Fig. 6A displays, for a specific situation, the maximum possible shift speed towards low as a function of the secondary pressure level. The maximum shift speed is determined by the minimum primary clamping force, corresponding to a primary safety of 1.3. For this situation, if a ratio change speed of 0.5 [1/s] is desired, the secondary pressure level must be increased to 12.6 [bar] instead of the 2.1 [bar], which corresponds to a stationary situation and a secondary safety of 1.3.

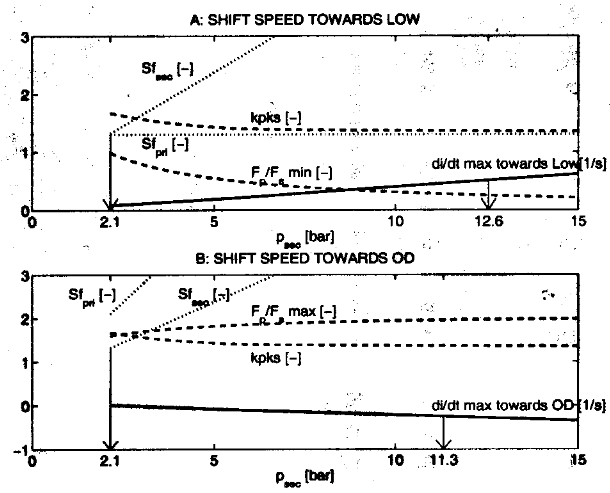

Fig. 6 Variator shift model simulation of the achievable shift speed as a function of secondary pressure level (P884 variator, speed ratio 0.5 [-], primary speed 2000 [rpm], primary torque 50 [Nm]).

Fast shifts towards OD

For shifts towards OD the primary clamping force is increased, so primary safety is not an item. The clamping force ratio however, is limited to a maximum of 2, i.e. the pulley surface ratio, because the maximum primary pressure level equals the secondary pressure level in the layout under consideration. The shift speed model (1) indicates that shift speed is not only determined by the difference between F_p/F_s and kpks, but also by the absolute value of F_s. To reach a desired shift speed, it is sometimes necessary to increase the secondary clamping force beyond the level necessary for secondary safety

1.3. Fig. 6B shows a simulation of the maximum achievable shift speed to OD in the same operating point as in Fig 6A. The shift speed towards OD is determined by the maximum value of F_p/F_s. It can be seen that for this situation and this variator lay-out, shifting towards OD is not possible at a secondary pressure of 2.1 [bar], corresponding to a secondary safety of 1.3. To reach a shift speed of -0.25 [1/s], the secondary pressure must be increased to 11.3 [bar].

Secondary pressure control

The secondary pressure itself is controlled by means of PI-control, improving the control performance by taking into account the secondary valve characteristic.

5. MEASUREMENT RESULTS

The CVC was tested and developed in a Golf VR6 with a prototype P884 transmission. First, a measurement program was carried out to record the performance of the reference controller. In a later stage, the same measurements were carried out with CVC. Fig. 7 shows both results for take-off with 30 [%] throttle.

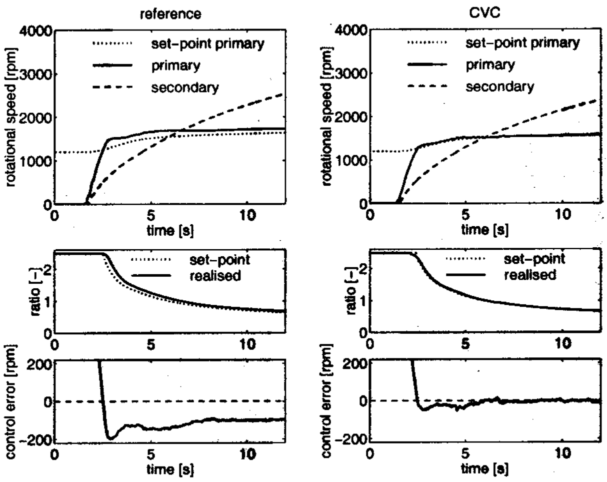

Fig. 7 Take-off with 30 [%] throttle, traditional control vs. CVC.

This measurement shows the benefits of the CVC for primary speed control accuracy. The original controller shows a maximum control error of -200 [rpm] slowly decreasing to -100 [rpm]. With the model-based algorithm, the error is reduced to a maximum of -50 [rpm] reducing to ±10 [rpm]. With the CVC it is possible to control a stationary primary speed with an accuracy of ±10 [rpm], due to the integral action. This allows very accurate e-line speed control, as described in Section 2.1.

Fig. 8 shows a fast brake measurement with CVC. The pump of this transmission is designed to enable complete downshift to ratio Low at a deceleration of 0.6 [g] at an engine (=pump) rotational speed of 1200 [rpm]. To demonstrate the CVC, in Fig. 8 a faster brake action (0.75 [g]) at 1200 [rpm] engine speed is shown. At the first instances of the brake action, accurate ratio tracking is realised. Primary safety is taken care of by increasing the secondary pressure above its level desired for secondary safety. At 2.2 [s] flow limits are reached (see the break-down of secondary pressure). From this moment on primary safety is taken care of by limiting the shift speed. Although Low is not reached, optimal shift speed is realised while guaranteeing the torque capacity by ensuring the primary safety.

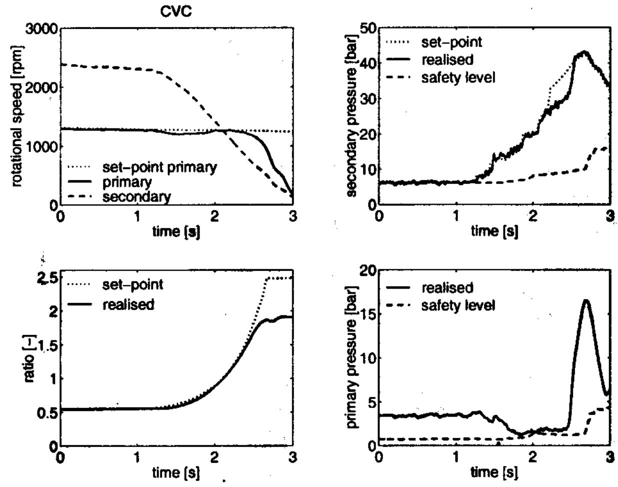

Fig. 8 Emergency brake 0.75 [g] at 1200 [rpm] CVC (safety level=required pressure for safety 1.3).

Fig. 9 shows an CVC measurement of tip-mode shifts at a constant vehicle speed of 50 [km/h]. It can be seen that the secondary pressure is increased in order to realise the high shift speed towards OD. For the shifts towards Low, the primary pressure level touches but never crosses its lower bound, determined by primary safety. Again the ratio control is very accurate, keeping the dynamics of the set-point in mind.

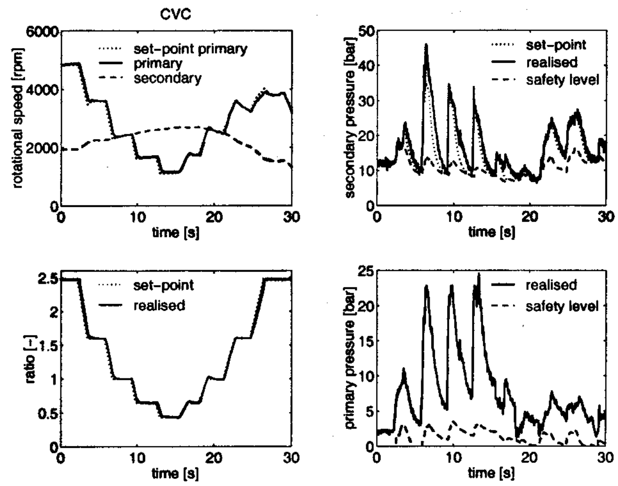

Fig. 9 Tip-mode shifts at 50 [km/h] CVC (safety level=required pressure for safety 1.3).

6. CONCLUSION

In this paper it is shown that variator control performance can be significantly improved by incorporating system knowledge in the control algorithm. For a simple hydraulic layout, without extra demands concerning sensor and actuator equipment, a fast and accurate primary rotational speed control is realised, while maintaining the clamping force even in extreme situations.

CVC® will be applied in all transmission development projects at Van Doorne's Transmissie. In order to demonstrate the efficiency potential, a CVT with dedicated layout and CVC is under development; a demo-vehicle of this will be available by the end of 1999.

ACKNOWLEDGMENT

The EcoDrive project is a joint project of Van Doorne's Transmissie (VDT), Netherlands Organisation of Applied Research (TNO) and the Eindhoven University of Technology (TUE). The project is subsidised by the Dutch governmental program E.E.T. (Economy, Ecology and Technology).

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