Performance optimisation of the push-belt CVT by variator slip control


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Abstract: Continuously variable transmissions (CVTs) are applied in an increasing number of vehicles. Large ratio coverage allows for reduced engine speeds, which adds to both highway driving comfort and reduced fuel consumption. It becomes increasingly important to further improve the performance in terms of efficiency, robustness and torque capacity of the CVT. This paper describes the possibilities of improving the CVT by minimising variator clamping forces. This is accomplished by using slip control technology. This technique allows for the best possible transmission efficiency, combined with improved robustness for slip damage.

This paper first describes the relation between variator slip and functional transmission properties. The conditions for optimum performance regarding efficiency and robustness are identified.

This leads to the development of a variator slip controller. The remaining sections describe experimental results on two test rigs and in a production vehicle. The paper concludes with an outlook into further developments.

Keywords: control; CVT; efficiency; robustness; slip.


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1 Introduction

Continuously variable transmissions (CVTs) are increasingly used in automotive applications. They have an advantage over normal automatic transmissions, due to their very large ratio coverage and absence of comfort issues related to shifting events as they occur in automatic transmissions. This enables the engine to operate at more economic operating points. Despite these advantages, V-belt type CVTs still have rather large potential in transmission efficiency. Also torque capacity (currently at about 350 Nm) needs expansion.

The main reason for the low efficiency of modern production CVTs is the high clamping forces in the variator necessary to transfer engine torque. To prevent belt slip at all times, the clamping forces in modern production CVTs are usually much
higher (typically 30% or more) than needed for normal operation. Higher clamping forces result in additional losses in both the hydraulic and the mechanical system. This is due to increased pump losses and friction losses because of the extra mechanical load that is applied on all variator parts. Excess clamping forces also reduce the endurance life of the V-belt type element, since the net pulling force in this element is larger than strictly necessary for the transfer of engine power. Also the contact pressure between V-belt and sheaves is higher than strictly necessary, leading to increased wear. This excess loading leads to heavier components, thereby compromising power density.

Reducing clamping forces, therefore, leads to improvements in various design areas: increase in transmission efficiency, increased torque capacity, reduced maximum hydraulic pressure and improved power density (Micklem et al., 1996). Preventing slip damage however, is still an important prerequisite for any clamping force control, although recent studies indicate that extended slipping between belt and pulley is allowed under certain conditions (van Drogen and van der Laan, 2004).

This paper is subdivided in five major sections:

- Section 2 describes the relationship between variator slip and its functional properties, like torque and slip losses and effective friction coefficient.
- Section 3 shows the derivation of a generic simplified model of variator slip dynamics. Some basic insights regarding slip control will be developed here.
- Section 4 describes slip control implementation on a test rig, containing only a variator and two motors for driving and braking, respectively.
- Section 5 shows results obtained with slip control on a dyno, comprising a full drive train with commercially available CVT. Efficiency improvement will be shown in this section.
- Section 6 reports on the experience obtained with variator slip controller implemented in a production vehicle. Some imperfections of the implemented slip controller will be identified.

2 Variator slip and functional properties

This section describes results from an experimental study into the efficiency of the push-belt variator and its capability to transfer torque. The relationship between belt slip and these functional properties was a particular subject of study.

2.1 Variator losses

Contrary to form closed transmission components like gears, the variator not only loses power by torque loss, but slip loss also occurs. Although slip loss is generally much smaller than torque loss in a V-belt variator, it cannot be neglected, since slip loss tends to increase when over-clamping is reduced. In the following two sections the methods by which these losses were analysed will be described.
2.1.1 Variator torque loss

Variator torque loss may be defined by:

\[ T_{\text{loss}} = T_p - r_{s0} T_s, \]  

with \( r_{s0} \) defined as the speed ratio \( r_s = \omega_s / \omega_p \) at zero variator output load. Here \( \omega_p \) and \( \omega_s \) denote the angular speed of the primary and secondary variator shaft, respectively and \( T_p \) and \( T_s \) represent primary and secondary torque, respectively. This relationship is identical to the one used for gears, except for the assertion that the speed ratio should be evaluated at zero load, so as to make sure that slip losses are excluded from \( T_{\text{loss}} \). Here it is implicitly assumed that the belt running radii do not change when torque is changed. Measurements of \( T_{\text{loss}} \) have been carried out on a variator test rig. During the measurements, variator ratio, clamping force and input torque were varied. Typical results for three different ratios at a constant secondary clamping force of 8 kN are given in Figure 1. These measurements show that \( T_{\text{loss}} \) is independent from input torque at constant clamping force, which is in line with literature data (Ide, 1999).

![Figure 1](image.png)

2.1.2 Variator slip loss

Variator slip loss may be defined as follows:

\[ \omega_{\text{loss}} = \omega_p - \frac{\omega_s}{r_{s0}}. \]  

A relative slip number can be defined by:

\[ \nu = \frac{\omega_{\text{loss}}}{\omega_p} = 1 - \frac{r_s}{r_{s0}}. \]  

For the common case where positive power flows from primary to secondary shaft, the variator efficiency can now be derived:
Neglecting torque loss, the relation between secondary clamping force $F_s$ and input torque $T_p$ can be represented by (Vroemen, 2001):

$$
\eta = \left(1 - \frac{T_{loss}}{T_p}\right) \left(1 - \frac{\omega_{loss}}{\omega_p}\right),
$$

(4)

where $R_p$ denotes the belt running radius on the primary sheave and $\beta$ denotes the pulley wedge angle. This effective friction coefficient or traction coefficient is known to depend weakly on ratio, clamping force and shaft speeds, but depends strongly on the amount of slip. Figure 2 shows how $\mu_{eff}$ and variator efficiency $\eta$ depend on slip at fixed ratio and clamping force $F_s$. The safety factor can be defined by

$$
S_f(\nu) = \frac{\mu_{eff,max}}{\mu_{eff}(\nu)},
$$

(6)

where $\mu_{eff,max}$ represents the maximum value of $\mu_{eff}$ that is obtained for all positive values of $\nu$. A high safety factor indicates that the clamping force is much larger than necessary to transfer the input torque. A high value for $S_f$ is associated with very low slip and also low values for $\mu_{eff}$, as can be concluded from Equation (6). As can be seen in Figure 2, the maximum efficiency is attained at, or very close to, the slip value where the traction curve suddenly changes slope. This is the slip value where $\mu_{eff}$ is very close to $\mu_{eff,max}$ and slip has a low value. This leads to the conclusion that in order to obtain the best possible transmission efficiency, slip must be controlled very close to $S_f = 1$. In contrast, in production CVTs, the over-clamping factor $S_f$ rarely exceeds 3 to 5 under part load conditions. Close inspection of Figure 2 indicates that in overdrive at $S_f = 1$, the efficiency reaches 94%, whereas at $S_f = 3$, the efficiency comes down to 82%.

**Figure 2** Efficiency $\eta$ and effective friction coefficient $\mu_{eff}$ vs slip $\nu$ measured at input speed of 300 rad/sec for variator ratios low (0.43), medium (1) and overdrive (2.25). The slip values at which variator efficiency reaches its maximum are indicated.
Regarding robustness for shock loads, if $S_f > 1$ the transferable torque will increase when the amount of slip is suddenly increased, for instance by driving over a bump in the road. For this no control action is needed. If, however, this slip increase occurs when $S_f = 1$, slip will increase without limit, possibly leading to severe variator damage. This can only be prevented if a quick control action is taken, by which the clamping force is increased to fit the required torque level. It must be noted here that limited excursions into the macro-slip area may be allowed for the push-belt variator, as indicated in van Drogen and van der Laan (2004).

In the following sections approaches to slip control are presented, which allow the variator to be operated at $S_f \approx 1$ while guaranteeing robustness for shock loads.

### 3 Modelling slip dynamics

For controller development, the torque generated on both shafts of the variator can conveniently described on the basis of Equation (5):

$$T_{p,s} = \frac{2F_r \mu(\nu) R_{p,s}}{\cos \beta}.$$  

(7)

Note that by using this description, torque losses are neglected. Although this limits model accuracy, the effect of torque loss is assumed to be small and not significant for the description of the variator dynamics. It is also assumed that speed ratio changes due to the axial motion of the variator sheaves are much smaller than those associated with slip. This assumption may impose limitations on the control strategy derived below, for those cases where fast ratio changes occur. This assumption allows the contribution of $\dot{r}_{s0}$ to $\dot{\nu}$ to be neglected. The slip dynamics can now be derived using Equation (3) and $R_s = \omega_p/\omega_s$, resulting in

$$\dot{\nu} = -\frac{\dot{r}_s}{r_{s0}}.$$  

(8)

$$\dot{r}_s = \frac{\dot{\omega}_s \omega_p - \omega_s \dot{\omega}_p}{\omega_p^2}.$$  

(9)

The dynamics of the variator as shown in Figure 3 can now be described by

$$\dot{\omega}_p = \frac{T_e - T_p}{J_p}.$$  

(10)

$$\dot{\omega}_s = \frac{T_s - T_d}{J_s}.$$  

(11)

Substituting Equations (7), (10) and (11) into Equation (9) leads to:

$$\dot{\nu} = \frac{1}{\omega_p} \left( -\frac{2F_r \mu(\nu)}{\cos(\beta) J_s r_{s0}} + \frac{T_d}{J_s r_{s0}} \right) + (1 - \nu) \left( -\frac{2F_r \mu(\nu)}{\cos(\beta) J_s} + \frac{T_e}{J_e} \right).$$  

(12)
This equation is non-linear in $\nu$ and will be linearised around different operating points. With the linearised model, a state space representation will be derived (Klaassen et al., 2004). Firstly, the effective friction coefficient is taken piecewise linear as indicated in Figure 4. The sections of the curve where the traction is increasing with increasing slip is the so-called microslip area, since this only occurs at very small values for slip. These are Sections II and III in Figure 4. The part of the traction curve where the traction is almost constant or slightly decreasing with slip, shown as Sections I and IV in Figure 4, is called the macro-slip area. Indicating the different regions with index $i$, the traction coefficient can be represented by:

$$\mu(\nu) = k_{1,i}\nu + k_{2,i}.$$  \hspace{1cm} (13)

Defining the state space as $x = \nu$ and $u = [F_s \ T_s \ T_d]^T$ the system can be linearised around a certain operating point $x = \nu_0$, resulting in the linear system:

$$\dot{x} = Ax + Bu,$$ \hspace{1cm} (14)
where \( \hat{x} = x - x_0 \) and \( \hat{u} = u - u_0 \). The linearised matrices \( A \) and \( B \) can now be derived (assuming \( \nu_0 \ll 1 \) and neglecting higher order terms):

\[
A = \frac{-2R_0 F_0 k_{1,0}}{\omega_0 J_0 \cos \beta} \quad \text{with} \quad J_0 = \left( \frac{r_{00}}{J_e} + \frac{1}{J_s r_{00}} \right)^{-1},
\]

(15)

and

\[
B = \frac{1}{\omega_0} \begin{bmatrix}
-\frac{2R_0 \mu(v_0)}{J_0 \cos \beta} \\
\frac{1}{J_e} \\
\frac{1}{J_s r_{00}}
\end{bmatrix}^T.
\]

(16)

The derived linearised system will be used for controller design. The system matrix \( A \) indicates that stability requires \( k_{1,0} \) to be positive. This is only the case in the micro-slip region, which is the main reason for its common use in production CVTs. A control action is needed in order to stabilise the system in the macro-slip region. The model has three inputs, but only the clamping force \( F_s \) can be controlled on implementation. In a vehicle application, the input torque \( T_e \) is controlled by the driver via the throttle pedal and the output torque \( T_d \) is determined by road conditions. Therefore they must be regarded as disturbances acting on the system. An example of a controller will be given later in Section 4.

In order to control slip, it must be measured accurately. Measuring slip on similar force closed components like wet plate clutches can be carried out easily, by measuring the speed difference of the in- and outgoing shafts. For a pushbelt variator, \( r_{00} \) must also be known, as can be seen from Equation (3). This quantity \( r_{00} \) is not directly available when the variator is loaded. If the variator geometry is assumed not to change when the torque is varied, \( r_{00} \) may be estimated from a measurement of the position of a moveable sheave. The geometric ratio \( r_g \), reconstructed from a linear displacement sensor is correlated to \( r_{00} \) for all relevant operating conditions. Since even small offsets in slip due to temperature changes or elastic deformations due to clamping force variation will cause severe control errors, all these effects must be taken into account in the calibration phase. Later sections will give additional detail on the measurement methods employed. It is important to mention a slip detection technique for a V-belt variator as it was reported in (Faust et al., 2002). In this technique the response of the variator in- and output speeds is detected when a cylinder pressure is excited with a sinusoidal excitation. This method has the important advantage that it does not rely on very accurate measurements, but on a signal that is directly proportional to slip. However, it must be expected that the method is not sensitive to changes in slip in the macro-slip region.

4 Slip control implementation on a test rig

The first attempt for slip control was carried out on the test rig shown in Figure 5 and schematically represented in Figure 6. This test facility comprises two identical asynchronous electric motors with the following specification: Maximum power:
78 kW, maximum torque: 298 Nm and maximum speed: 525 rad/sec. The pressures in the hydraulic cylinders of the variator could be controlled independently. The belt slip measurement system comprises three elements, two rotary encoders on both the input and output shaft, and a linear encoder on the secondary pulley position.

**Figure 5** Test setup as used in the experiments

![Figure 5](image1)

**Figure 6** Layout of the test setup

![Figure 6](image2)

**Table 1** Measurement equipment for angular speeds and sheave position

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Sensor</th>
<th>Resolution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sec. pulley position</td>
<td>Heidenhain ST3078</td>
<td>1 µm</td>
</tr>
<tr>
<td>Angular velocity</td>
<td>Heidenhain ERN 1381</td>
<td>1/2048 rad</td>
</tr>
<tr>
<td>Torques</td>
<td>HBM T20WN 200 Nm</td>
<td>0.2 Nm</td>
</tr>
</tbody>
</table>
This system is accurate enough to detect 0.1% slip, at normal operating speeds. A sixth order polynomial is used to approximate $r_{s0}$ as a function of the secondary pulley position as measured with the linear encoder. These measurements are performed by quasi-statically changing the ratio from low ratio to overdrive ratio and back at 5 kN secondary clamping force with zero output torque. A least squares method is used to fit the polynomial through the measured data. The oil sump temperature is controlled at 80°C. In this setup, the effect of elastic pulley deformation was found to be small enough to allow it to be ignored.

For the measurement of the pulley position contact is needed between the sensor and the rotating pulley. The environment in the transmission causes problems with contact-free methods and limits the space available to the sensor equipment. The pulley position is measured at the outer rim of the pulley, therefore the position of the pulley has to be adjusted for elastic deformation of the pulley itself. This is accomplished by carrying out the calibration process at different clamping force levels. The accuracy that can be achieved with this system is 0.1% with $\omega_p = 50$ rad/sec and better for higher values of $\omega_p$.

4.1 Controller design

The control system consists of three independent controllers, angular speed control, ratio control and slip control. The control scheme is presented in Figure 7. The non-linear function N uses the measured output from plant P to calculate slip and geometric ratio for the feedback loop. In plant P $F$ and $\Delta F$ are transformed to a primary and secondary pressure, which is, in turn, used in the hydraulic actuation system of the variator. This transformation takes both centrifugal forces in the oil as well as a spring force into account, if a spring is present.

![Figure 7 Control approach](image)

The controllers are designed independently although there are some interactions. The influence of the various controllers is examined below.
4.1.1 Angular speed control

Angular speed is controlled using the motor torque input signal. PI control is used to control the motor speed. The integral term is used to eliminate steady-state errors. The proportional feedback is used to make the system asymptotically stable.

The following control law is used for speed control:

\[ T_e = \left( P_r + \frac{L}{s} \right) (\omega_{\text{ref}} - \omega_p). \]  \hspace{1cm} (17)

4.1.2 Ratio control

The ratio of the variator depends on the balance of the primary and secondary clamping force. If the primary and secondary clamping force are in balance, the ratio is constant. If an extra clamping force is applied to either the primary or secondary pulley, the variator will shift. It is not recommended to decrease the clamping force at one side, because this will increase the slip in the variator and, therefore, the influence on the slip controller would be destabilising. Because the balance will shift, depending on the operating point of the variator, a certain interaction will still exist. Because the equilibrium of forces in the belt changes for different levels of slip in the micro-slip area and only slightly in the macro-slip area, the influence of the slip control on the ratio control will be small in the macro-slip area. The following control law is used for ratio control:

\[ \Delta F = \left( P_r + \frac{L}{s} \right) (r_{\text{ref}} - r_g). \]  \hspace{1cm} (18)

The shifting process in the variator exhibits a lot of damping, therefore no separate damping term is needed. Because the damping is high, the maximum bandwidth that can be reached is about 1 Hz. The integral term is used to establish the balance in a certain operating point, since the primary and secondary clamping force are almost never equal to each other in the variator. In the future a feed-forward controller can be used to compensate for the balancing problem.

4.1.3 Slip control

Slip in the system is controlled using the clamping force level. Other options are to control the input torques or the ratio to maintain a certain slip level. Controlling slip by the input torques is not a feasible solution, because the drive line side of the variator is not controllable in an automotive application. When the slip is controlled using the geometric ratio the system would be too slow, because of the slowness involved with shifting the variator. Also the control range of the ratio is limited. This could cause very high amounts of slip when the system has reached its limit.

For operating points where \( k_{1i} \) is zero or negative, i.e. the macro-slip area, the controller has to stabilise the system without destabilising the system in the other situations. The purpose of the slip controller is to operate the variator in the optimal operating point, which is a slip level depending on the ratio the variator is in. The range is between 1 and 3% slip. Since the application of this system is mainly in automotive applications, it is assumed that the road load torque \( T_d \) is not controllable.
The clamping force level is controlled by controlling the pressure in the hydraulic cylinders. At the same time the ratio is controlled by controlling the difference between the primary and secondary clamping force. To minimise the effect of the slip control on the ratio controller and vice versa, the slip controller gives the minimum clamping force needed and the ratio controller the clamping force difference. From these values the primary and secondary clamping forces and, therefore, the pressures can be derived.

A non-linear state feedback controller is used:

\[ F_p = \max(P_c \mu_c(\nu), F_{\text{min}}) \quad \text{with} \quad \mu_c(\nu) = \mu_{\text{eff}}(\nu)(|\nu| - |\nu_{\text{ref}}|). \]  

The function \( \mu_c(\nu) \) is shown for two values of \( \nu_{\text{ref}} \) in Figure 8, together with the original traction curve, \( \mu_{\text{eff}}(\nu) \). The system will be stable if the traction curve is continually increasing for all increasing values of \( \nu \) for which \( |\nu| > |\nu_{\text{ref}}| \) is true. As is shown in Figure 8 this is possible with the proposed feedback controller for values of \( P_c > 0 \). For values of \( \nu \) for which \( |\nu| \leq |\nu_{\text{ref}}| \), Equation (19) ensures that a minimum clamping force level \( F_{\text{min}} \) will be applied.

**Figure 8** Traction curve and the feedback strategy at \( \nu_{\text{ref}} = 0\% \) and at \( \nu_{\text{ref}} = 2\% \)

### 4.1.4 Process analysis

The sensitivity of the controlled system was measured by adding bandlimited noise to the controller output signal. By estimating the transfer function of the noise to the plant input signal an estimate for the sensitivity is obtained. The plant characteristics can be deduced from the sensitivity function with \( P = \frac{1 - S}{SC} \). The measured sensitivity of the system for the clamping force control is given for 0.5% slip in Figure 9. The ratio of the CVT was near overdrive \( (r_s \approx 2) \). The primary speed was 50 rad/sec. The sensitivity for 0.5% slip is low for very low frequencies and has a dip at 1.5 Hz.
The process frequency response function (FRF) is shown in Figure 10. The process at 0.5% has an eigenfrequency of 1.5 Hz. From the model a first order system would be expected, since two states are rigid body modes (eigenfrequency equals zero). The measurement results suggest the same. The slope of the process FRF goes to roughly $-1$ for high frequencies, with a phase shift of $-90$ degrees.

4.2 Results

In Figure 11 the response of the system to torque-steps of up to 30 Nm are shown. In an increasing sequence torque steps are imposed on the variator. The slip level remains within acceptable limits during the test. Slowness in the hydraulic system severely limits the performance of the system. Stick-slip behaviour was seen when the
delay in the system caused by low-pass filters was too high. Eliminating the filters rendered better performance. Good results were obtained using a hydraulic system with a 10 Hz bandwidth.

**Figure 11** Time response of the slip controlled system on torque changes

4.3 Conclusions from slip controller development on a test rig

Control of slip in a CVT is possible if the clamping force is used as the control output and a slip signal is available as control input. A limiting factor is the performance of the actuation system. It is shown that robustness for torque peaks from the drive line can be achieved by using the proposed control strategy. In the test setup the pulley position is measured using a position encoder and shaft speeds were measured by means of high resolution rotary encoders. This technology proved to be adequate for the implementation of slip control. Measurement of slip is a problem in normal automotive CVTs. No suitable feedback of pulley position, geometric ratio or belt speed is present. Also, the shaft speeds are measured by means of low resolution encoding technology, limiting both resolution and bandwidth of the measured signal. It is of great importance to investigate what specification is needed for the sensors that are to be used for evaluation of the slip signal. The following section will address this issue, since it covers the implementation of slip control in a production CVT with limited modifications to the transmission.

5 Slip control implementation in a production CVT

Based on the successful implementation of slip control on a test rig, as described in Section 4, it was decided to undertake an activity resulting in a slip controlled production CVT in a vehicle. The most important requirement of the slip controller
is that it has the ability to attenuate the load disturbances caused by torque peaks in the drive line. An additional problem in the controller design process is that the slip dynamics change for different values of ratio and angular speed. The system that controls the clamping force will be shown to have somewhat lower bandwidth as compared to the case described in Section 4. A robust gain-scheduled controller is therefore desirable, which can adapt the control parameters, depending on ratio and speed. To meet all requirements, a synthesis method for robust PI(D)-controllers with optimal load disturbance response is used (Panagopoulos et al., 2002). The designed slip controller is simulated and subsequently tested on a test rig.

5.1 Modelling actuation system dynamics

The clamping force in the Jatco CK2 is applied using hydraulic pressure cylinders attached to the movable pulleys (Abo et al., 1998). The oil pressure in the cylinders is regulated by a complex electro-hydraulic actuation system that is controlled by a PWM-based solenoid. The duty cycle of the PWM-signal determines the oil pressure that provides the clamping force, or the line pressure. The line pressure in the CK2 is limited between 0.66 and 4.2 MPa, between these values the pressure varies practically linear with the duty cycle. Modelling this electro-hydraulic system is a complex and time-consuming task, therefore the system’s dynamic response is determined using Frequency Response Function (FRF) measurements. A good estimation of the system’s response will then be used in the controller design process. For the FRF-measurements, the duty cycle of the solenoid is taken as the input and the line pressure as the output. The measurements were performed at different pressures, ratios and engine speeds. All measurements showed practically identical system responses, only with slightly different gains for low frequencies, but small enough to be neglected. Figure 12 shows the result of one of the FRF-measurements.

Figure 12 Measured and estimated FRF of the line pressure circuit in the Jatco CK2
The system is estimated with a third order low-pass filter with a cut-off frequency of 6 Hz, which is also plotted in the figure. The frequency of the PWM signal is 50 Hz, this causes a peak in the FRF. Because the bandwidth of the system is much lower than 50 Hz it is not taken into account in the estimation.

5.2 Slip controller design

5.2.1 The control design problem

With the linearised model of the slip dynamics as described in Section 3 and the estimated transfer function of the actuation system a slip controller can be designed. Using Equations (15) and (16) the variables that influence the slip dynamics the most can be found. There is a large difference in the system response between the micro- and macro-slip region. For slip control design, attention is mainly focused on the macro-slip region. In this region, ratio and primary speed have the greatest influence on the dynamics. A gain scheduled controller is designed by linearising the slip dynamics in a number of operating points and by calculating the controller parameters for each operating point. As mentioned above, the slip controller requires good load disturbance attenuation and must be robust to deal with model uncertainties. For this purpose a gain-scheduling PID-controller was proposed. However, due to the large amount of measurement noise in automotive applications the derivative term cannot be used. Therefore a gain-scheduling PI-controller is proposed, as shown in Figure 13. As can be seen, the gain is scheduled based on primary speed, ratio and slip. Slip is used to determine whether the system is in the micro- or macro-slip region. The setpoint also varies with the ratio, since the maximum traction coefficient is reached for different slip values, depending on the ratio, as can be seen in Figure 2.

Figure 13 Proposed gain scheduling PI slip controller

5.2.2 Robust PI-controller synthesis method

To design controller parameters easily for multiple operating points, while meeting both design requirements, a synthesis method for robust PI(D)-controllers with optimal load disturbance response is used (Panagopoulos et al., 2002). The method is based on a constrained optimisation problem that maximises the integral gain of the PI(D)-controller while making sure that the maximum sensitivity is less than a specified value. Using the maximum sensitivity as the main design parameter, a
A trade-off can be made between load disturbance response and robustness with respect to model uncertainties. The resulting controller parameters of this optimisation process can be obtained graphically for a PI-controller. It produces a series of ellipses in the controller parameter space, called the k-ki plane, for different frequencies of the system. These ellipses represent a boundary for the sensitivity constraint, and together they form a boundary surface in the k-ki plane. Choosing combinations of k and ki below this surface ensures a stable and robust closed loop system. For optimal load disturbance attenuation, the maximum value of the integral gain is determined from the figure. The proportional gain is then determined graphically. Figure 14 shows a typical result of the synthesis method. Using this synthesis method for different ratios in the micro- and macro-slip region, the gain-scheduling scheme presented in Table 2 is obtained.

Figure 14 Example of the sensitivity constraint in k-ki plane

<table>
<thead>
<tr>
<th>Ratio</th>
<th>Micro-slip region</th>
<th>Macro-slip region</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>P-gain</td>
<td>I-gain</td>
</tr>
<tr>
<td>0.43</td>
<td>1.7</td>
<td>30</td>
</tr>
<tr>
<td>1</td>
<td>1.9</td>
<td>53</td>
</tr>
<tr>
<td>2.25</td>
<td>3.6</td>
<td>110</td>
</tr>
</tbody>
</table>

The differences between the micro- and macro-slip region mentioned above, result in very different values for the controller parameters. This is because the system dynamics drastically change at the transition from the micro to the macro-slip region. The system matrix A in Equation (14) almost becomes zero in the macro-slip region. This means that a part of the system dynamics disappears, resulting in important changes in the systems gain.
Another reason is that in the macro-slip region the gain becomes scalable by the primary speed. This can be seen in Equation (16), considering the fact that system matrix A is practically zero. Therefore the gains in Table 2 for the macro-slip region are scaled by the primary speed (in rad/sec) in the controller. Based on ratio, slip, and primary speed, the proper controller parameters are used. Between the operating points shown in Table 1 interpolation will be used. To ensure the stability of the controller between these operating points, several measures were taken. In the micro-slip region load disturbance response is not very important since slip will not cause any damage in this region. However, many model uncertainties are present, because the slip dynamics depend on many variables in this region. Therefore a maximum sensitivity of 1.2 is chosen in the controller synthesis method, which is relatively low. In the macro-slip region a maximum sensitivity of 1.8 is chosen. This is much higher since there are fewer model uncertainties in this region and a good load disturbance response is required. Additionally the worst-case values of the controller parameters were taken to ensure stability for every operating point.

5.2.3 Controller implementation

In order to successfully implement the controller described in the previous section, an integral anti-windup is added. This is necessary because the output of the controller is limited between the minimum and maximum pressure level of the CK2. To prevent slip caused by the engine torque $T_e$, a feed forward term is added based on Equation (7), which calculates the minimal clamping force to transmit the given engine torque. The engine torque is estimated using the engine speed and the throttle valve position. This feed forward is needed because the bandwidth of the slip controller is not sufficient to compensate for the fast dynamics of a combustion engine. With these additions the slip controller is ready for implementation.

5.3 Results

5.3.1 Test setup

Figure 15 shows a schematic representation of the test rig on which the slip controller has been implemented. It is designed to perform realistic drive train experiments, using a 2.0-litre combustion engine as the power source and a flywheel, an eddy-current brake, and a disc brake to simulate road loads. The torques on the input and output shaft of the transmission are measured using telemetry systems. These torque sensors were used only for the measurement of efficiency. The Jatco CK2 can be controlled with the Transmission Control Unit (TCU) that is used in a car or with the newly developed slip controller. This is very useful for efficiency comparison. The angular speeds of the primary and secondary shaft of the CVT are measured to determine the ratio $r_s = \omega_s / \omega_p$. The no-load ratio $r_{0s}$ is determined using a Linear Variable Differential Transformer (LVDT) to measure the displacement of the primary pulley, as described earlier. The system is accurate enough to detect 0.1% slip, which should be sufficient to implement the slip controller.
5.3.2 Efficiency measurements

Slip control is developed to improve the efficiency of CVTs, therefore, efficiency measurements are carried out to show the efficiency benefits of slip control in a production CVT. The efficiency when using the TCU is compared to the efficiency when using the slip controller. The efficiency comparison is carried out at fixed ratios and with a constant engine speed of 300 rad/sec. The slip value is controlled between 0.5% for ratio 2.25 (overdrive) and 1.5% for ratio 0.43 (low). At these slip values the maximum efficiency of the CK2 is reached. The engine torque is gradually increased and plotted against the efficiency. Figure 16 shows that the efficiency improvement when using the slip controller is quite significant, especially for low engine torques. Since the average engine torque in normal drive cycles is usually relatively low, this is a very promising result. For ratios until ratio 1.4, the efficiency improvement is a little lower than for low ratios, but still in the order of 10 to 5%. For ratios higher than 1.4, the efficiency improvement becomes less. When driving in overdrive, there is hardly any improvement. This is caused by the minimum pressure level in the CK2 of 0.66 MPa, which results in a minimum clamping force of almost 10 kN. For normal engine torques hardly any slip will occur in overdrive with this clamping force level, therefore the benefits of slip control cannot be fully exploited in the current CK2. Lower clamping forces are required for slip control in ratios near overdrive.

Figure 15 Test rig layout

Figure 16 Comparison of efficiencies between TCU and slip control, measured for ratio 0.64, at an engine speed of 300 rad/sec
5.3.3 Load disturbance measurements

The previous section shows that slip control significantly increases the efficiency of a CVT. This was to be expected, based on previous studies. The next step is to perform experiments where torque peaks are introduced in the driveline, thus testing the performance of the slip controller with load disturbances. Also interesting in these tests are the amounts of slip that occur using slip control and whether this damages the belt and pulleys. Experiments were performed at fixed ratios and with a fixed engine speed of 200 rad/sec. The slip was controlled at the same values that were used for the efficiency measurements. The eddy-current brake provided a constant torque high enough to reach a slip value at the transition between the micro and macro-slip region. Torque peaks were then introduced by suddenly engaging the disc brake. Limitations in the disk brake actuation system restricted the rise time of these torque peaks, however. Figure 17 shows the result of one of these measurements. The figure shows that the torque peaks cause belt slip, which was expected at the transition of the micro- and macro region. But instead of reaching destructive levels, the slip is quickly reduced to non-destructive levels because of the control action.

Figure 17 Slip controller performance with torque peaks acting on the driveline, measured for ratio 0.43 (low), at an engine speed of 200 rad/sec

The slip controller is able to deal with torque peaks of up to 1000 Nm in the drive shaft, although this causes the slip level to peak above 5% for short periods of time. Visual inspection however, showed that the belt was not damaged after such tests. This would mean that short peaks in slip do not cause belt damage. Additional tests should be performed to investigate if this is true for all operating points of the CVT. These tests should also include faster disturbances in torque, as they may from road irregularities. Another important aspect that should be considered is the long-term effect of slip control with respect to belt damage. If necessary, the bandwidth of the controller could be increased to get better load disturbance response, resulting in lower peak values of the belt slip. This can be achieved by improving the gain scheduling scheme with more operating points and using higher maximum sensitivities. If this is not sufficient, an alternative actuation system with a higher bandwidth should be used.
5.4 Conclusion from slip control implementation in a production CVT

The developed slip controller shows efficiency improvements of the Jatco CK2 of up to 30% at low engine torques. Even larger improvements are expected if lower clamping forces could be applied, which makes it possible to use the benefits of slip control for higher ratios and even lower engine torques. This should be considered in future CVT research and design. Using the slip controller it is possible to operate a CVT with minimal clamping forces, while preventing damage to the belt and pulleys. Relatively high slip levels (5–15%), which were present for short periods of time during the tests, did not lead to damage to the system. The slip controller is able to attenuate load disturbances of up to 1000 Nm in the drive shaft and perhaps even more. This is true for the current test conditions, but more research is necessary to investigate the long-term effect of using slip control in a wide range of operating points with respect to belt damage.

6 Slip control implementation in a production vehicle

From the experiments on the test rigs it became clear that slip control has great potential for both efficiency and robustness improvement of production CVTs. In order to perform more realistic experiments, especially with regard to the variators robustness and fuel consumption, the slip controller is implemented in a production vehicle. The car that is used is a Nissan Primera 2.5i with CK-Kai (very similar to CK2) as presented in Figure 18. It provides a maximum engine torque of 250 Nm and delivers a maximum power of 115 kW. The slip control strategy was identical to the one that was developed on the dyno, as reported in Section 5.

6.1 System description and requirements

The slip controller was implemented with a dSPACE Autobox. In order to be able to compare the slip control strategy with the conventional clamping force strategy, a switch-box was installed. The clamping force was controlled by the slip controller, whereas ratio and torque converter lock-up were controlled by the original TCU. The slip controller was tested on a test track with many different drivers and some results will be presented below.
6.2 Test track results

The vehicle was tested on the Bosch proving grounds in Boxberg, Germany. Over 50 persons drove the car over a handling course to test the slip controller. These tests showed that under normal driving conditions the slip controller worked perfectly. When driving in a more aggressive way however, the slip controller was not always able to attenuate slip peaks fast enough. This resulted eventually in wear of the belt and pulleys. Figures 19 and 20 show the two major problems that occurred during the test drive.

- Closing of the torque converter caused relatively large slip peaks and vibrations in the drive line. Because the slip controller increases the clamping force with increasing slip, these slip peaks did not cause damage to the belt and pulleys. The vibrations in the drive line, however, cause a very uncomfortable driving experience. Figure 19 shows an example of such a measurement. Opening and closing of the torque converter cause an important change in the dynamics of the drive train. This effect should be taken into account for further development.

- Large slip peaks occurred often with fast variator shifts. This is because the slip controller was designed for quasi-stationary shifting behaviour (see Section 3). With aggressive driving this assumption becomes invalid since large and fast downshifts occur with fast variations of the throttle pedal position. Shifting of the variator can trigger belt slip, as can be seen in Figure 20. Shifting dynamics should, therefore, be taken into account for future slip control development. This is especially interesting due to the observation reported in Ide et al. (1996), where it is found that shifting dynamics is enhanced when the variator slips.

Figure 19 Slip peak due to torque converter lockup
Figure 20  Slip peak associated with a fast downshift

7 Conclusions and recommendations

- Push belt variator slip control is a feasible technology. Stabilisation of slip in the macro-slip regime proved possible.
- Slip control is needed when the best transmission performance in terms of efficiency and power density are to be obtained.
- Lowering of the belt clamping force to very low values is a prerequisite in order to obtain efficiency improvements under low torque conditions.
- Slip control development requires a good understanding of the functional properties of the push belt variator.
- Presentation of slip control in a production vehicle with only minor modifications, proves the viability of this technology also for industrial application.
- Control problems that were identified during test driving, could be attributed to shortcomings in control logic, due to insufficient modelling detail.
- Although slip control showed good functionality in a production vehicle, additional hardware modifications for good performance under more demanding conditions may be necessary.
- Future research must focus on: reduction of clamping forces under low load conditions, while maintaining a fast hydraulic response; improvement of actuator dynamics; improved slip measurement technology; included shift dynamics in slip dynamics modelling.
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References


Nomenclature

$I$ Integral control factor
$J_e$ Engine side or primary inertia
$J_s$ Vehicle side or secondary inertia
$J_t$ Effective system inertia
$T_e$ Engine or drive torque
$T_p$, $T_s$ Primary, secondary variator torque
$T_d$ (Vehicle) drag torque
$T_{loss}$ Torque loss
$F_{min}$ Minimum clamping force
$F_s$ Secondary clamping force
$P$ Proportional control factor
$R_p$, $R_s$ Primary, secondary belt running radius
S\textsubscript{f} \quad \text{Over-clamping or safety factor}

k\textsubscript{1}, k\textsubscript{2} \quad \text{Linear fit coefficients of } \mu(\nu)

r\textsubscript{g} \quad \text{Geometric ratio}

r\textsubscript{s} \quad \text{Speed ratio}

r\textsubscript{s,0} \quad \text{Speed ratio at } T\textsubscript{sec} = 0

\beta \quad \text{Pulley sheave angle}

\eta \quad \text{Variator efficiency}

\mu\textsubscript{eff} \quad \text{Effective friction coefficient}

\nu \quad \text{Slip}

\omega \quad \text{Angular speed}

\omega\textsubscript{loss} \quad \text{Speed loss}

\textbf{Indices}

i \quad \text{Index of segment in traction curve}

0 \quad \text{Equilibrium value}

p, s \quad \text{primary, secondary}

r \quad \text{Ratio control}

\text{ref} \quad \text{Reference}

v \quad \text{Velocity control}

\nu \quad \text{Slip control}