Improving Pushbelt Continuously Variable Transmission Efficiency via Extremum Seeking Control

Stan van der Meulen, Bram de Jager, Erik van der Noll, Frans Veldpaus, Francis van der Sluis, and Maarten Steinbuch

Abstract—The control design for the variator in a pushbelt continuously variable transmission (CVT) is investigated. The variator enables a stepless variation of the transmission ratio within a finite range. A conventional control design for the variator is typically obtained by the use of a variator model, which incorporates large uncertainties and, therefore, limits the variator efficiency. In this paper, a control design for the variator is discussed, which improves the variator efficiency and limits the number of sensors. The relation between inputs and outputs of the variator is investigated, from which one input-output map is identified, which exhibits a maximum. This maximum indicates performance in terms of the variator efficiency. For this reason, this input-output map is maximized by means of extremum seeking control (ESC), which omits the use of a variator model. Experiments illustrate that the approach is feasible and show that a conventional control design for the variator is outperformed.

I. INTRODUCTION

The pushbelt continuously variable transmission (CVT) is a stepless power transmission device, which is able to provide infinitely many transmission ratios within a finite range. In comparison with stepped power transmission devices, e.g., manual transmissions (MTs) and automatic transmissions (ATs), this characteristic property of the CVT enables: 1) a more comfortable driveline behavior, since the power transfer is not interrupted and 2) a more efficient driveline behavior, since the number of internal combustion engine (ICE) operating points for a certain demanded power is not restricted, which improves the exploitation of the ICE characteristics and decreases the fuel consumption [1]. In order to further improve the performance of the CVT, the reduction of the power losses within the CVT is considered.

The primary sources of power loss concern the variator and the actuation system [2]. The variator consists of a metal V-belt, i.e., a pushbelt, which is clamped between two pairs of conical sheaves, i.e., two pulleys. The pushbelt consists of circa 400 V-shaped compression elements that are held together by two sets of either 9 or 12 thin tension bands. A transmission ratio change is enforced by the actuation system, which exerts forces on both axially moveable sheaves, i.e., one for each pulley. Since the majority of modern production CVTs is equipped with a hydraulic actuation system, the focus is on this type of actuation system. The variator losses include the power loss: 1) in the bearings of the variator shafts, 2) between the innermost band and the elements and between adjacent bands, i.e., a pushbelt internal power loss [3], and 3) between the elements and the pulleys [4] and [5].

The level of the variator power losses is strongly related to the level of the clamping forces. Hence, a reduction of the clamping forces leads to a reduction of the variator power losses. Furthermore, this indirectly results in a reduction of the actuation system power losses, since the hydraulic pressures are lowered. The main reason for the application of large clamping forces concerns avoidance of variator damage. Variator damage results from repeated occasions of a (too) high level of slip in the variator. Recently, research with respect to variator wear has shown that the variator is able to withstand substantial levels of slip [6], which enables a reduction of the clamping forces. When the clamping forces are decreased, the slip in the variator is increased. A transition of the variator behavior from open loop stable (micro-slip) to open loop unstable (macro-slip) occurs, since the friction force between the pushbelt and the conical sheaves is limited. Moreover, the variator efficiency initially increases, reaches a maximum, and ultimately decreases. The variator efficiency maximum occurs within the region for which the variator behavior is open loop stable [7].

In [8], the existence of a certain optimum for the variator efficiency as a function of the slip is shown by means of experiments. As a result, a straightforward approach is to control the slip in such a way that a certain slip reference is tracked, which corresponds to the optimum variator efficiency [9]. However, this approach involves two issues. First, the determination of the slip reference [10, Section 7.2]. Since the optimum variator efficiency depends on, e.g., the transmission ratio, the variator torques, and the variator wear, the determination of the slip reference is not straightforward and often time-consuming, which is typically caused by the complexity and the unreliability of the available variator models. Second, the measurement of the slip in the variator. This typically requires a dedicated sensor, e.g., measurement of the pushbelt running radius [11] or measurement of the axially moveable sheave position [9], which increases both the complexity and the costs. In addition, the computation of the slip in the variator on the basis of one of these measurements is extremely sensitive to deformations in the variator, which are unknown.

These considerations give rise to alternative approaches, which only utilize the sensors that are typically available in modern production CVTs. Moreover, a technique is desired that searches for the optimum variator efficiency and adjusts the control signals, i.e., the pulley pressures, accordingly. The approach that exploits the presence of a so-called extremum,
i.e., either a minimum or a maximum, in a specific uncertain input-output equilibrium map, to adapt the input in order to optimize the output, is called extremum seeking control (ESC) [12]. Since in general the primary pulley torque and the secondary pulley torque are not measured, the variator efficiency is unknown. As a result, adaptation of the clamping force (input) to optimize the variator efficiency (output) is not possible. For this reason, the input-output equilibrium map in which the clamping force is the input and the ratio between the pulley angular velocities is the output is considered, which also incorporates a maximum. The variator efficiency is indirectly improved, since the argument of this maximum results in a variator efficiency value that is close to the variator efficiency maximum [13].

The main contribution of this paper is that the optimization of the mentioned input-output equilibrium map by means of ESC is experimentally demonstrated, which improves the variator efficiency and limits the number of sensors. The remainder of this paper is organized as follows. The preliminaries are addressed in Section II. The experimental setup is introduced in Section III. The ESC approach is described in Section IV, together with closed loop experiments. Finally, the paper concludes with a discussion in Section V.

II. PRELIMINARIES

A. Variator Working Principle

In Fig. 1, the variator is schematically depicted. The variator consists of a metal V-belt, which is clamped between two pairs of conical sheaves, i.e., the primary (input, subscript "p") side and the secondary (output, subscript "s") side. On each side, one sheave is fixed and one sheave is axially moveable. The axially moveable sheaves are located on opposite sides of the pushbelt and are actuated by hydraulic pressure cylinders. Adjustment of the transmission ratio is achieved by simultaneous adjustment of the clamping forces that are exerted on the axially moveable sheaves. This varies the running radii of the pushbelt on the sheaves and, consequently, the transmission ratio. The torques that are exerted on the variator are denoted by $T_p$ and $T_s$. Furthermore, the angular velocities are denoted by $\omega_p$ and $\omega_s$, the clamping forces by $F_p$ and $F_s$, the axially moveable sheave positions by $x_p$ and $x_s$, and the running radii by $R_p$ and $R_s$. The variator is able to cover any transmission ratio in between the two extremes Low and High.

B. Definitions

In the definitions, the variator geometry is considered ideal. The geometric ratio $r_g$ and the speed ratio $r_s$ of the variator are defined by:

$$ r_g = \frac{R_p}{R_s} $$

$$ r_s = \frac{\omega_s}{\omega_p}. $$

The relative slip $\nu$ is defined by:

$$ \nu = \frac{\omega_p R_p - \omega_s R_s}{\omega_p R_p} = 1 - \frac{r_s}{r_g}. $$

see [8]. The variator efficiency $\eta$ is defined by:

$$ \eta = \frac{P_{\text{out}}}{P_{\text{in}}} = \frac{T_s \omega_s}{T_p \omega_p}, $$

where $P_{\text{in}}$ and $P_{\text{out}}$ denote the input power and the output power, respectively. This definition prevents the distinction between the sources of power loss within the variator and neglects the power for the hydraulic actuation system.

C. Control Objectives

The objective for control of the driveline is often a weighted optimum of fuel economy and driveability [14]. The objective for control of the variator, i.e., one of the driveline components, is twofold, see also Fig. 4. First, to track the speed ratio reference $r_{s,\text{ref}}$, which is prescribed by the driveline control system. Second, to optimize the variator efficiency, under the condition that variator damage is avoided. Here, the focus is on the latter objective. Since the number of sensors in modern production CVTs is limited, i.e., only the primary pulley angular velocity, the secondary pulley angular velocity, and the secondary pulley pressure are measured, the variator efficiency is unknown. As a result, direct control of the variator efficiency is prevented and indirect control of the variator efficiency is pursued. Of course, this approach is only useful when a conventional variator control system is outperformed, which is formalized below.

Consider the following definition and assumptions.

Assumption 1: The variator is geometrically fixed, i.e., the primary axially moveable sheave position $x_p$ is mechanically constrained.

For this reason, the secondary side is used for evaluation purposes.

A conventional variator control system in modern production CVTs employs a so-called safety strategy, which is either relative or absolute, in order to handle both uncertainties and transients [2]. The relative safety strategy and the absolute safety strategy are defined by:

$$ F_{\text{rel,s}} = \frac{\alpha_{\text{rel,s}} T_p \cos(\beta)}{2 \mu R_p} $$

$$ F_{\text{abs,s}} = \frac{\left(1 + (\alpha_{\text{abs,f}} - 1) \frac{T_{p,\text{max}}}{T_p} \right) T_p \cos(\beta)}{2 \mu R_p}, $$

where the traction coefficient $\mu$ is equal to $\mu = 0.09 [-]$. The factor 2 accounts for the number of friction surfaces, $T_{p,\text{max}}$.
denotes the primary external torque maximum, and $\beta$ denotes half the pulley wedge angle, which is equal to $\beta = 11$ [deg]. Furthermore, $\alpha_{rel, f}$ and $\alpha_{abs, f}$ denote the relative safety value and the absolute safety value, respectively. In this paper, the relative safety strategy is adopted for comparison purposes.

**Definition 2:** In a conventional variator control system, the secondary clamping force is given by:

$$F_{\text{safe}, s} = \frac{\alpha_f T_s \cos(\beta)}{2\mu R_s},$$  \hspace{1cm} (7)

where the relative safety value $\alpha_f$ is equal to $\alpha_f = 1.3$ [-].

**Remark 3:** In [2], the absolute safety value $\alpha_{abs, f} = 1.3$ [-] is applied, which is representative for modern production CVTs. A comparison between (5) and (6) reveals that $\alpha_{rel, f} = 1.3$ [-] identifies with $\alpha_{rel, f} \geq 1.3$ [-], where $\alpha_{rel, f} \gg 1$ [-] for large periods of time within the standardized driving cycles [2]. As a result, the relative safety strategy in Definition 2 evaluates the best-case performance of the absolute safety strategy in [2].

**Assumption 4:** The variator efficiency $\eta$ is a concave function of the secondary clamping force $F_s$ with a global maximum defined by $(F_s|_{\eta=\eta_{\text{max}}}, \eta_{\text{max}})$. 

**Assumption 5:** The speed ratio $r_s$ is a concave function of the secondary clamping force $F_s$ with a global maximum defined by $(F_s|r_s=r_{s,\text{max}}, r_{s,\text{max}})$. 

Consider Definition 2 and Assumptions 4 and 5.

**Assumption 6:** The relation between the secondary clamping force values is given by:

$$F_s|_{\eta=\eta_{\text{max}}} < F_s|_{r_s=r_{s,\text{max}}} < F_{\text{safe}, s},$$  \hspace{1cm} (8)

Consider the following proposition.

**Proposition 7:** If Assumptions 4, 5, and 6 hold, then optimization of the input-output equilibrium map in which the clamping force is the input and the speed ratio is the output improves the variator efficiency in comparison with a conventional variator control system.

In [13], several experiments are performed in order to show that Assumptions 4, 5, and 6 are satisfied.

### III. Experimental Setup

The experimental setup is depicted in Fig. 2 and consists of four main components. These are given by two identical electric motors, a pushbelt CVT variator, and a hydraulic actuation system. The experimental setup incorporates additional sensors in comparison with a modern production CVT, which are primarily used for analysis purposes.

#### A. Electric Motors

The electric motors (SIEMENS, type 1PA6184-4NL00-0GA03) are located on either side of the pushbelt CVT variator. The maximum power level is equal to 81 [kW] from 2900 [rpm] to 5000 [rpm], which is the maximum angular velocity. The maximum torque level is equal to 267 [Nm] below 2900 [rpm]. Both electric motors are equipped with a rotary encoder (HEIDENHAIN, type ERN 1387).

#### B. Pushbelt CVT Variator

Each shaft of the pushbelt CVT variator (VAN DOORNE’S TRANSMISSIE, type P811) is connected to one electric motor by means of two elastic couplings with a torque sensor (HBM, type T20WN) in between. The secondary axially moveable sheave position $x_s$ is measured with a length gauge (HEIDENHAIN, type ST 3078). Then, the geometric ratio $r_g$ follows from the variator geometry.

#### C. Hydraulic Actuation System

The hydraulic actuation system consists of several hydraulic pumps for actuation and lubrication. The pulley pressures in the hydraulic pressure cylinders are controlled by means of two servo valves (MANNESMANN REXROTH, type 4 WS 2 EE 10), which are fed from a shared accumulator. The maximum pulley pressure levels are equal to $p_{p,\text{max}} = 20$ [bar] and $p_{s,\text{max}} = 38$ [bar]. Each hydraulic pressure cylinder is equipped with a pressure sensor (GE DRUCK, type PTX 1400). Both hydraulic actuation circuits $G_{H_j}$, $j \in \{p, s\}$, see Fig. 3, are closed loop pressure controlled.

![Fig. 3. Control of hydraulic actuation circuits $G_{H_j}$](image)

The clamping forces $F_p$ and $F_s$ are mainly realized by the pulley pressures $p_p$ and $p_s$. However, centrifugal effects are also of relevance, since the oil in the hydraulic pressure cylinders often rotates with very high angular velocities. Furthermore, a preloaded spring is attached to the secondary axially moveable sheave, which provides the secondary spring force $F_{\text{spring}, s}$. This spring guarantees a certain secondary clamping force value when the hydraulic actuation system fails. On the basis of these contributions, the clamping forces $F_p$ and $F_s$ are given by:

$$F_p = A_p p_p + c_p \omega_p^2$$  \hspace{1cm} (9a)

$$F_s = A_s p_s + c_s \omega_s^2 + F_{\text{spring}, s}(x_s),$$  \hspace{1cm} (9b)

where $A_p$ and $A_s$ denote the hydraulic pressure cylinder surfaces, whereas $c_p$ and $c_s$ denote the centrifugal coefficients.
IV. Extremum Seeking Control

The presence of a maximum in the input-output equilibrium map in which the clamping force is the input and the speed ratio is the output is investigated by means of experiments in [13]. A quantitative prediction of this extremum by means of a variator model is a hard task, especially in real-time. This is caused by the complexity and the unreliability of the available variator models, in which several phenomena are typically simplified, e.g., the local friction models. The ESC approach omits a variator model and applies to situations where a nonlinearity exists in the control problem, which has either a local minimum or a local maximum. Here, the nonlinearity exists in the variator behavior, i.e., a physical nonlinearity, which appears through the input-output equilibrium map.

A. Assumptions

A certain stationary operating point is considered, with \( \omega_p = \bar{\omega}_p \), \( T_s = \bar{T}_s \), and \( r_g = \bar{r}_g \). Consider the general single-input single-output (SISO) nonlinear system:

\[
\begin{align*}
\dot{x} &= f(x,u) \\
y &= g(x),
\end{align*}
\]

where \( x \in \mathbb{R}^n \) is the state, \( u \in \mathbb{R} \) is the input, \( y \in \mathbb{R} \) is the output, and \( f : \mathbb{R}^n \times \mathbb{R} \to \mathbb{R}^n \) and \( g : \mathbb{R}^n \to \mathbb{R} \) are smooth. The state \( x \), the input \( u \), and the output \( y \) are defined by:

\[
x = r_s, \quad u = F_r, \quad y = x.
\]

A decrease of the secondary clamping force \( F_r \) results in a transition of the variator behavior from open loop stable to open loop unstable, since the friction force between the pushbelt and the conical sheaves is limited. The transition of the variator behavior from open loop stable to open loop unstable occurs for the secondary clamping force value \( \bar{F}_r \).

The following assumption is introduced with respect to the transition. Consider Assumption 4.

Assumption 8: The relation between the secondary clamping force values is given by:

\[
\bar{F}_r \leq F_r |_{y = \eta_{max}}.
\]

This assumption is supported by the experimental results in this paper and in [7, Fig. 3].

The following assumptions are introduced with respect to the SISO nonlinear system (10).

Assumption 9: There exists a smooth function \( h : \mathbb{R} \to \mathbb{R}^n \) such that:

\[
\dot{x} = 0 \iff x = h(u),
\]

for the interval \( u > \hat{u} \).

From the quasi-stationary experimental results in [13], it follows that Assumption 9 is satisfied.

Assumption 10: There exists \( u^* \in \mathbb{R} \) such that:

\[
(g \circ h)'(u^*) = 0 \quad (g \circ h)''(u^*) < 0.
\]

That is, the output equilibrium map \( y = g(h(u)) \) has a maximum for \( u = u^* \).

From the quasi-stationary experimental results in [13], it follows that Assumption 10 is satisfied.

B. Objective and Concept

In practice, a variator control system is typically implemented in terms of pulley pressures instead of clamping forces, where the interconnections are defined by (9). See the control configuration in Fig. 4. Here, \( G_V \) denotes the relation between \( p_s \) and \( r_s \) and \( T_{hp} \) and \( T_{hs} \) denote the relations between \( p_{p,ref} \) and \( p_p \) and \( p_{s,ref} \) and \( p_s \), respectively.

![Control configuration](image)

This suggests the introduction of the following definitions:

\[
u = p_s, \quad u_{ref} = p_{s,ref}.
\]

The following objective is formulated with respect to the SISO nonlinear system (10).

Objective 11: Design a feedback mechanism from \( y \) to \( u_{ref} \) without the knowledge of both \( u_{ref}^* \) and the function \( h \) that maximizes the steady-state value of \( y \).

This feedback mechanism is depicted in Fig. 5. Obviously, the feedback mechanism utilizes a sinusoidal perturbation \( \alpha_m \sin(2\pi f_m t) \), which is added to \( u_{ref} \), i.e., the estimate of the optimum input \( u_{ref}^* \). As a result, the input of the variator \( u_{ref} \) is defined by:

\[
u_{ref}(t) = \hat{u}_{ref}(t) + \alpha_m \sin(2\pi f_m t),
\]

where \( \alpha_m \) denotes the perturbation amplitude and \( f_m \) denotes the perturbation frequency. When the periodic perturbation is slow in comparison with the variator dynamics, the variator essentially manifests itself in the form of a static map. Consequently, there is no interference between the variator dynamics \( G_V \) and the feedback mechanism.

When \( u_{ref} \) is on either side of \( u_{ref}^* \), the periodic perturbation enforces a periodic response of the output of the variator \( y \), which is either in phase or out of phase with the periodic perturbation. With this information, the feedback mechanism from \( y \) to \( \hat{u}_{ref} \) is designed, which consists of the following operations:

\[
\begin{align*}
\xi_1 &= H_b(s)y \\
\xi_2 &= H_b(s)u \\
\xi_3 &= \xi_1 \xi_2 \\
\xi_4 &= H_i(s)\xi_3 \\
\hat{u}_{ref} &= \frac{1}{s}I\xi_4.
\end{align*}
\]

Here, \( H_b(s) \) denotes a band-pass filter, \( H_i(s) \) denotes a low-pass filter, and \( I \) denotes the integrator gain. The band-pass filter \( H_b(s) \) enforces the suppression of “DC components” and noise for \( y \) and \( u \), which results in \( \xi_1 \) and \( \xi_2 \), respectively. As a result, \( \xi_1 \) and \( \xi_2 \) are approximately two sinusoids, which are in phase for \( \hat{u}_{ref} > u_{ref}^* \) and out of phase for \( \hat{u}_{ref} < u_{ref}^* \). In either case, the product of both sinusoids \( \xi_3 \)
has a “DC component”. The low-pass filter $H_l(s)$ extracts the “DC component” of $\xi_3$, which results in $\xi_4$. Finally, $\hat{u}_{\text{ref}}$ results from integration of $\xi_4$, with integrator gain $I$. The initial condition for the integrator is equal to $\hat{u}_{\text{ref}}$, which corresponds to a certain stationary operating point. Observe that (21) contains the gradient information and (22) represents the gradient update law, which enables the adaptation of $u_{\text{ref}}$ towards the optimum input $u_{\text{ref}}$.

Obviously, the feedback mechanism incorporates five design options. These are the perturbation amplitude $\alpha_m$, the perturbation frequency $f_m$, the band-pass filter $H_b(s)$, the low-pass filter $H_l(s)$, and the integrator gain $I$. The selection of these design options is closely related to the proof of stability for the closed loop system, which is addressed in [12]. The feedback mechanism in [12] is similar to the feedback mechanism in Fig. 5. However, a high-pass filter is employed instead of a band-pass filter. The main reason for the application of a band-pass filter concerns the suppression of noise. When Assumptions 8, 9, and 10 are satisfied, convergence of the solution $(x(t), \hat{u}_{\text{ref}}(t), \xi_3(t), y(t))$ towards a certain neighborhood of the point $(h(u^*_{\text{ref}}), u^*_{\text{ref}}, 0, g \circ h(u^*_{\text{ref}}))$ is guaranteed by [12, Theorem 5.1] for a suitable choice of the design options.

**C. Design Options**

The band-pass filter $H_b(s)$ and the low-pass filter $H_l(s)$ are given by:

$$H_b(s) = \frac{1}{s^2 + \frac{2}{\pi f_m} 0.01 s + 1}$$
$$H_l(s) = \frac{1}{\frac{2}{\pi f_m} s + 1}.$$  \(\text{(23)}\)

Both the band-pass filter $H_b(s)$ and the low-pass filter $H_l(s)$ are discretized on the basis of a first-order hold discretization scheme, in order to realize a discrete time implementation.

Upper bounds are imposed on the perturbation amplitude $\alpha_m$ and the perturbation frequency $f_m$, in order to confine the size of the region to which the solution converges [12, Theorem 5.1]. On the other hand, a sufficiently large $\alpha_m$ and $f_m$ are required in order to excite the variator and to achieve convergence, respectively, see [15, Section 1.2.3]. The band-pass filter $H_b(s)$ is designed in accordance with the perturbation frequency $f_m$ and enforces the suppression of “DC components” and noise. The cut-off frequency of the low-pass filter $H_l(s)$ is a fraction of the perturbation frequency $f_m$, see [12]. Finally, the integrator gain $I$ is limited in order to confine the size of the region to which the solution converges [12, Theorem 5.1]. On the other hand, a sufficiently large $I$ is desired in order to accelerate convergence.

**D. Closed Loop Experiments**

The operation of the feedback mechanism is evaluated by means of closed loop experiments. The geometric ratio is equal to either Low or High. In order to obtain a concise presentation of the results, only the geometric ratio High is considered for a single secondary external torque reference. Similar results are obtained for the geometric ratio Low and alternative secondary external torque references. The experiment starts from a certain stationary operating point, which is defined by $\bar{\rho}_{p,\text{ref}} = 20$ [bar], $\bar{\rho}_{s,\text{ref}} = 6$ [bar], $\bar{\omega}_{p,\text{ref}} = 1000$ [rpm], and $\bar{T}_{s,\text{ref}} = 14$ [Nm]. The primary axially moveable sheave position $x_p$ is fixed, which is enforced by the choice of the primary pulley pressure reference $\bar{p}_{p,\text{ref}}$. The angular velocities and the torques are depicted in Fig. 6. The primary angular velocity error $\bar{\omega}_{p,\text{ref}} - \bar{\omega}_p$ is typically small due to closed loop velocity control, whereas the secondary torque error $\bar{T}_{s,\text{ref}} - T_s$ is possibly large due to open loop torque control. The secondary pressure is depicted in Fig. 7. The experimental results as a function of the secondary clamping force $F_s$ are depicted in Fig. 8.

For $0 \leq t \leq 50$ [s], the loop is opened in front of the integrator, i.e., $\hat{u}_{\text{ref}} = \bar{u}_{\text{ref}}$. For $t > 50$ [s], the loop is closed in front of the integrator, i.e., $\hat{u}_{\text{ref}} \neq \bar{u}_{\text{ref}}$. From Fig. 7, it follows that $u_{\text{ref}}$ decreases towards $u^*_{\text{ref}}$, which is approximately reached for $t \approx 100$ [s]. From Fig. 8 (top left), it follows that the feedback mechanism converges towards a small neighborhood of the extremum $r_s = r_{s,\text{max}}$, which corresponds to the global maximum. From Fig. 8 (bottom left), it follows that the feedback mechanism outperforms a conventional variator control system, since $\eta(F_s|r_s=r_{s,\text{max}}) > \eta(F_{\text{safety}}|s)$.

Furthermore, the level of slip in the variator is limited, as well as the oscillations that are introduced, see Fig. 8 (bottom right), which implies that variator damage is avoided. Finally, a change of the geometric ratio is observed in Fig. 8 (top right), which results from a change of deformations in the variator. In order to accelerate convergence, a choice of the initial condition for the integrator close to $u^*_{\text{ref}}$ is desired, which is provided by the conventional variator control system, for example.

**V. Discussion**

In this paper, a control design for the variator in a pushbelt continuously variable transmission (CVT) is proposed, which improves the variator efficiency and limits the number of sensors. From a series of experiments, several conclusions are drawn: 1) a global maximum exists for the variator efficiency as a function of the secondary clamping force, 2) a global maximum exists for the speed ratio as a function of the secondary clamping force, and 3) the argument in 2) results in a variator efficiency value that approximates the maximum in 1). These findings form the basis for the application of extremum seeking control (ESC), where the input-output equilibrium map in which the secondary clamping force is the input and the speed ratio is the output is optimized. In
of the variator are preserved. This is already indicated by experimental results in [13].

The primary control objective for the variator concerns tracking a prescribed speed ratio reference. The secondary control objective for the variator concerns improving the variator efficiency, which is highlighted in this paper and, therefore, a single-input single-output (SISO) control problem is obtained. The ESC feedback mechanism is employed, which requires further research in order to handle transients and transitions of the variator behavior from open loop stable to open loop unstable, which are possibly caused by disturbances. Ultimately, two control objectives are considered and two control signals are available, which yields a multi-input multi-output (MIMO) control problem. This requires the investigation of interaction between inputs and outputs, in order to make a well-founded decision with respect to the feedback control design.

REFERENCES