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Pushbelt CVT efficiency improvement potential of servo-electromechanical actuation and slip control

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ABSTRACT

By using the ability to shift to high overdrive ratios, a CVT equipped vehicle can outperform the fuel economy of its counterpart with conventional automatic transmission. A further significant reduction of fuel consumption can be obtained by reducing transmission power loss. The main sources of power loss are well known to be losses inside the V-belt variator and losses caused by driving the hydraulic pump. Significant reduction of these losses will increase the attractiveness of the V-belt type CVT.

This paper first analyses the most important loss contributions present in a reference transmission and how they depend on actuation and control properties, like pump driving power, over-clamping and variator slip. Based on this analysis, actuation and control improvements will be proposed. The efficiency increase is presented, that can be expected when the hydraulic pump is replaced by a servo-electromechanical actuation system, and when variator slip control is introduced. The efficiency benefit is shown to be substantial, especially at part load conditions. At 25 % of rated load the CVT losses are halved, whereas at full load, a reduction of 25 % can be expected.

The proposed servo-electromechanical actuation system is currently in the realization phase, as well as the slip control technique. Experimental results will be reported on when available.

Keywords: continuously variable transmission, efficiency

INTRODUCTION

Since its introduction over 15 years ago, the CVT has contributed to the improvement of fuel economy of passenger cars. It gives a better fuel economy than the stepped automatic transmission (AT) due to its wider ratio coverage and its ability to shift ratio without compromising comfort. Another advantage is that it allows for the torque converter to lock up at lower vehicle speeds than when an AT is applied. Unfortunately, the metal V-belt CVT does not have a much better efficiency than the AT. An important reason for this is that it needs significantly higher hydraulic pressure for variator actuation purposes, (typically up to 50 [bar]) than an AT does, which leads to a larger oil pump drive torque. Since the oil pump usually rotates at a speed proportional to engine speed, quick ratio shifts at low engine speed dictate rather large pump displacement. For a typical CVT, rated at 200 [Nm], the pump displacement amounts to about 19 [cc/rev]. Making use of the double piston principle in combination with a two stage torque sensor, allows for a more favorable pump displacement of 11 [cc/rev] at 310 [Nm] rated torque [1]. Additionally, the power loss in the variator (defined as belt, pulleys and shaft bearings) is not small. This leads to transmission efficiencies which barely exceed 90 [%] at rated load. Under part load conditions, efficiency drops even further. This is caused by limitations in the actuation system, preventing the clamping force to drop to sufficiently low values, causing over-clamping. In order to prevent variator slip, over-clamping is applied, causing increased variator torque loss. On various driving cycles used for fuel consumption benchmarks, a large time fraction is spent in part load conditions. Therefore, improving part load ef ciency receives considerable attention. Improving ef ciency at rated load will be bene cial for high torque situations like top speed, launch and fast acceleration maneuvers.

In order to reduce actuation system power consumption, Bradley and Frank [2] proposed a servo-electro-hydraulic system, largely eliminating excessive oil o w. The pumps need to deliver enough o w to compensate for seal leak. A reduction of actuation power of about one order of magnitude was reported.

By using variator slip as a control variable for determining the distance between actual clamping force and the slip threshold at which the variator is irreversibly damaged, Faust et. al. [1] indicate a transmission ef ciency improvement of almost 2 [%].

This paper will present the transmission ef ciency improvement that can be expected after replacing the hydraulic pump with a servo-electromechanical actuation system and after introduction of variator slip control. Also the fuel economy bene t in a number of operating points will be given. First, the modelling of the ef ciency of the reference transmission will be presented.

REFERENCE TRANSMISSION EFFICIENCY MODELLING

For benchmark purposes, the 2 liter class Jatco CK2, manufactured by Jatco Transtechnology Co. Ltd. was selected. This type of transmission will also be used as a carrier for the servo-electromechanical actuation system, that is currently in the realization phase. The reference transmission comprises a torque converter as drive-off element, a DNR set, a hypotrochoid gear pump, a push-belt type variator with ratio coverage of 5.4, followed by the na l reduction and differential gear [3]. The majority of the presented data will be limited to the over-drive ratio, since, this ratio is most relevant for fuel economy evaluations.

EFFICIENCY ANALYSIS METHOD

In order to assess the ef ci ency improvement potential of the proposed modi cations , rst the power losses occurring in the reference transmission were carefully modelled. The main power losses are caused by the hydraulic pump and by the variator and were obtained by means of measurements. Losses in the nal reduction were calculated by means of literature data and models [4]. These loss components will be treated separately in the following sections.

PUMP LOSS One of the major loss sources within the transmission is the power absorbed in driving the hydraulic pump. This loss was measured on a spin-loss test rig. This test rig allows to drive the transmission with unloaded output. For these measurements, the clutch plates of the DNR set were removed to eliminate the torque loss due to friction between clutch plates. In order to control pump torque independent from the original transmission controller, a new transmission control system was developed, allowing full control over the CK2, including clamping pressure, variator ratio and torque converter lock-up. The dependence of pump power loss on input speed (equal to engine speed), pump output pressure (equal to clamping pressure in secondary variator cylinder) and temperature, was measured in a wide operating range and mapped into lookup tables.

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

Variator torque loss Variator torque loss may be de ned by:

$$T_{loss} = T_{pri} - r_G T_{sec}, \tag{1}$$

with r_G de ned as the speed ratio $\omega_{sec}/\omega_{pri}$ at zero variator output load. Here ω_{pri} and ω_{sec} denote the angular speed of the primary and secondary variator shaft, respectively. Measurements of T_{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 ratio's at a constant secondary clamping force of 8 [kN] are given in Fig. 1. These measurements show that T_{loss} is independent from input torque at constant clamping force, which is in line with literature data [5]. It was therefore decided to measure torque loss under unloaded conditions on a CK2 transmission on the same test rig that was used for pump loss determination. By removing the intermediate shaft, power loss in the n al reduction was eliminated from the measurement. By switching the transmission into drive and applying torque converter lock-up and by subtracting pump torque loss from the measured data, variator torque loss as a function of input speed, ratio and clamping force could be determined over a wide operating range. A result from these measurements is shown in Fig. 2. Here, the data is used to calculate

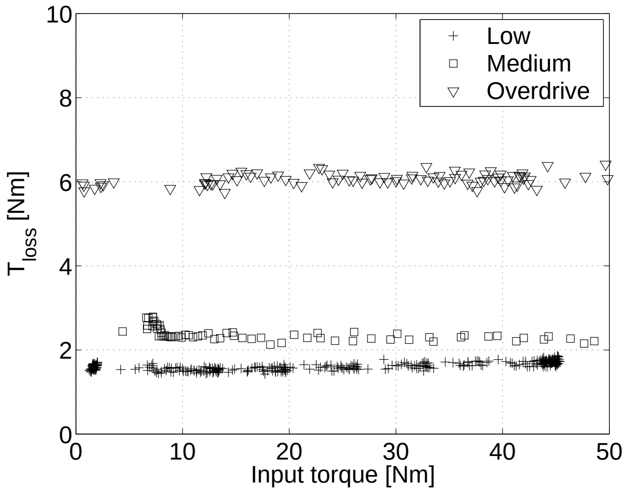

Figure 1: *Typical results of T_{loss} measurements at three different ratios at secondary clamping force of 8 [kN] and 225 [rad/sec] input speed.*

variator ef ciency (including slip loss, to be described in the next section) as a function of safety factor S_f, which is de ned as the ratio of actual clamping force and the minimum clamping force, required for transfer of a speci c input torque at a speci c variator ratio. This graph shows that S_f must be reduced to the lowest possible value in order to obtain the best variator ef ciency. Usually, lowering of the safety factor is severely limited, especially for part load conditions. Very often, the secondary or line pressure, directly related to secondary clamping force, cannot be lowered to values below p_{sec_min} = 5-7 [bar]. This is caused by the fact that from the line pressure the auxiliary pressure is derived, which should not drop below 5-7 [bar]. Low safety value at low nominal torque increases the risk of variator slip caused by torque peaks originating from road irregularities. This forms another reason why safety values are seen to be considerably higher than 1 in practice.

Variator slip loss Variator slip loss may be de ned as follows:

$$\omega_{loss} = \omega_{pri} - \frac{\omega_{sec}}{r_G} \qquad (2)$$

A relative slip number can be de ned by:

$$s_\omega = \frac{\omega_{loss}}{\omega_{pri}} = 1 - \frac{r_\omega}{r_G}. \qquad (3)$$

with $r_\omega = w_{sec}/w_{pri}$. The dependence of slip on torque can effectively be approximated by a linear relationship for $S_f > 1$ [6]. The dependence of the slope of this linear relationship on ratio and clamping force was captured in a lookup table.

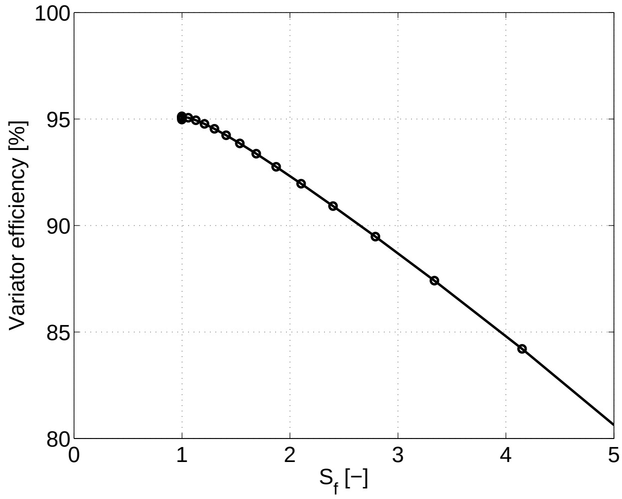

Figure 2: *Efficiency versus safety (over-drive, 1500 [rpm]).*

FINAL REDUCTION LOSSES The nal drive of the CVT is similar to that found in any conventional transmission and can be modelled in a similar way. The calculations have been based on literature data [4] and references therein. Bearing loss, oil churning loss and gear meshing loss have been taken into account.

REFERENCE TRANSMISSION LOSS BREAKDOWN

The ef c iency analysis mentioned in the previous sections, was used to generate Fig. 3, showing the power loss breakdown of the reference transmission in overdrive at input speed of 1500 [rpm].

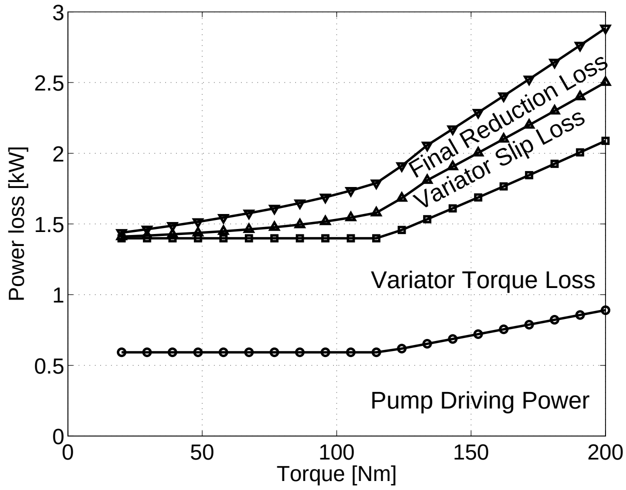

Figure 3: *Loss breakdown of reference transmission (over-drive, 1500 [rpm]).*

The clamping force strategy, as used in the controller of the reference transmission was closely followed: the safety factor was held at S_f = 1.3. The minimum pressure level that could be achieved was 6.5 [bar]. Obviously torque loss in the variator and pump loss are the most important loss sources. The change in slope occurring at about 120 [Nm] is caused by the limitation in hydraulic pressure to drop below 6.5 [bar]. This leads to rather severe over-clamping at low torque levels ($S_f \sim 3$ at 50 [Nm]), causing both pump driving loss and variator torque loss to be rather high.

EFFICIENCY IMPROVEMENT OPTIONS

From the previous sections it may be concluded that for the optimization of transmission ef ciency by actuation and control measures, the following aspects should be taken into account:

- Over-clamping should be avoided or, equivalently, the safety factor S_f should be kept close to 1, also at low torque levels.
- In order to realize this, hydraulic limitations for lowering clamping force should be avoided
- Actuation power must be reduced rigorously

In order to realize this, a servo-electromechanical actuation system with variator slip control has been proposed, which will be described in the following sections.

THE SERVO-ELECTROMECHANICAL ACTUATION SYSTEM (EMPACT) In order to assess the apparent advantages of an all-mechanic system, an electrically driven variator actuation system has been proposed by van de Meerakker et al. [7]. This system is schematically shown in Fig. 4 and is characterized by the following features:

- Both moveable sheaves are actuated by means of electric motors (Mp and Ms)
- Both motors are stationary when both clamping force and variator ratio are constant. This is accomplished by applying two differential gear sets, serving as a decoupling mechanism between the rotating transmission shafts and the transmission housing.
- By coupling two moving ring gears of the adjustment mechanisms, shifting energy can be exchanged between one movable sheave which is operating with positive power and the other operating at negative power.
- Clamping force loops are closed on the rotating shaft and not supported via the transmission housing, thereby avoiding highly loaded thrust bearings.
- A separate low pressure pump for auxiliary hydraulic functions is also required.

From mid 2004 onward, a prototype transmission with electromechanically actuated variator will be evaluated on a test rig.

VARIATOR SLIP CONTROL As shown in Fig. 2, variator torque loss can be reduced by minimizing over-clamping. However, this increases the risk of variator slip. In order to limit variator slip as de ned in the section on variator slip loss, variator slip control has been proposed [8]. With this type of control, safety values very close to 1 can be realized. It is important to note here that the EMPAct system calls for a novel way of estimating the clamping force, since a hydraulic pressure signal is absent. Slip control seems therefore to be a logical approach for assessing the clamping status of the variator. As mentioned previously, applying low safety values, increases the risk of variator slip, especially at part load conditions. Road irregularities may cause short torque peaks in the drive train, causing a temporary macro-slip condition. Recent experimental results [9] have shown however, that the push-belt variator is much more robust against macro-slip than previously assumed. It will be the subject of further research to what extent this property can be used when applying slip control.

EFFICIENCY MODELLING FOR A TRANSMISSION WITH EMPACT SYSTEM

The planetary gear sets of the EMPAct system on the primary and secondary transmission shaft are both loaded and all sun gears are spinning at the shaft speed under steady state (constant torque, clamping force and ratio) conditions. Both servo-motors Mp and Ms generate torque, but do not rotate in steady state. In order to assess the losses to be expected in the gear mechanisms, a model was constructed on the basis of literature data [4]. Gear meshing losses prove to be the major contribution to the loss in the gear sets. The spindles, driving the moveable sheaves, are assumed to have zero loss, which is valid under stationary conditions. This assumption facilitates the calculation of the load on the planetary gear sets. Losses in the stationary servo-motors were assumed to be negligible. The loss associated with driving an electric pump to feed the auxiliary functions like solenoids and clutches, was also taken into account.

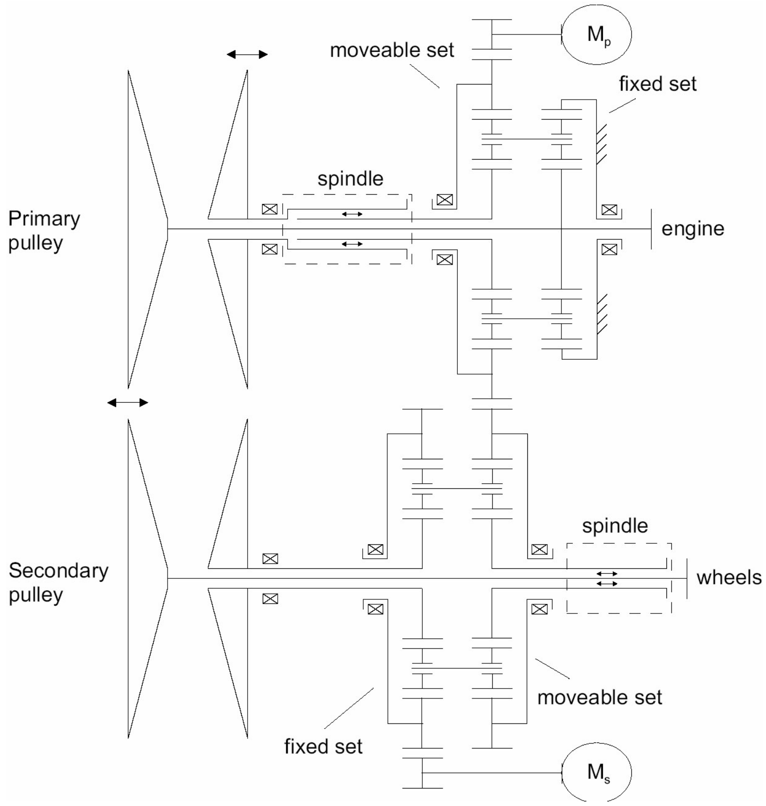

Figure 4: *Schematic representation of the servo-electromechanical actuation system.*

LOSS BREAKDOWN FOR CVT WITH EMPACT SYSTEM

The ef c iency analysis mentioned in the previous sections, was used to generate Fig. 5, showing the loss breakdown of the transmission with EMPAct system and slip control in over-drive at an input speed of 1500 [rpm].

In comparison to Fig. 3 the following differences can be noted:

- The change in slope at about 120 [Nm] has disappeared. This is associated with the ability of the EMPAct system to reduce clamping force to low levels.
- Especially the power loss at part load is strongly reduced.
- Actuation power is strongly reduced
- Part load variator torque loss is reduced considerably, which must be attributed to the reduction of over-clamping.
- Variator slip loss has increased somewhat by the reduction of the safety factor to values close to 1. This is most clearly seen at high input torque levels.

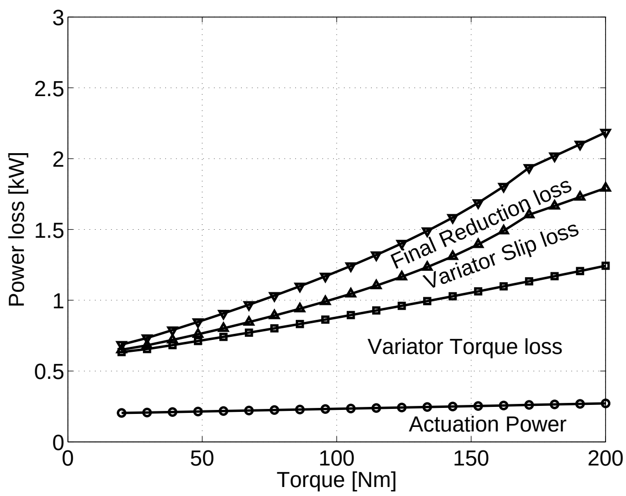

Figure 5: *Loss breakdown of transmission with EMPAct system and variator slip control (over-drive, 1500 [rpm]).*

From Fig. 6 the effects of various measures on power loss can be estimated. The dotted curve indicates the very important effect of removing the clamping force

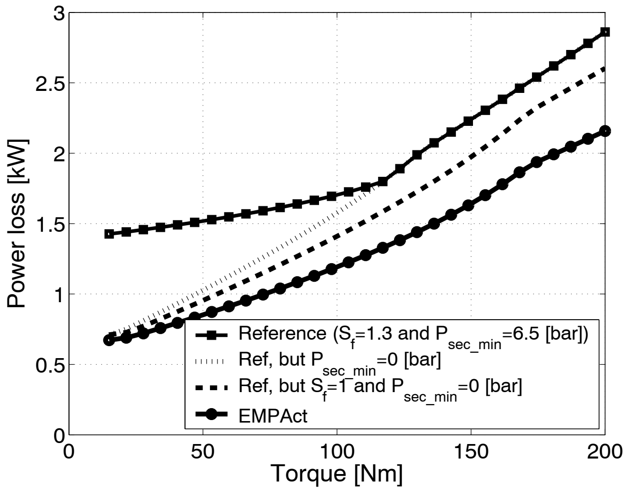

Figure 6: *Loss reduction by various measures in comparison to the Reference transmission.*

limitation, so that the secondary pressure can be lowered to p_{sec_min} = 0 [bar]. The dashed curve represents a further lowering of safety to S_f = 1, meaning that variator slip control must be applied, in order to avoid gross slip. The curve indicated with EMPAct shows the most important loss reduction at higher torques, due to the reduction of actuation power.

The total power loss data represented by the highest curves in Fig. 3 and Fig. 5 can easily be converted into ef ciency. The result of this operation can be seen in Fig. 7. This gure clearly shows the important ef-

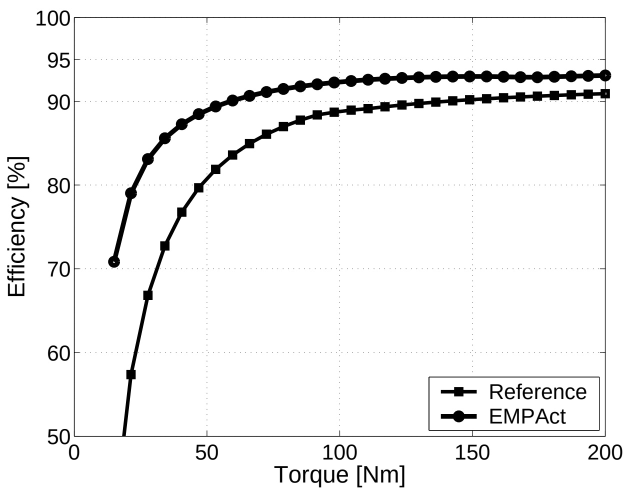

Figure 7: *Transmission efficiency comparison for the Reference transmission and the transmission equipped with EMPAct system and variator slip control.*

ciency gain that can be expected with the EMPAct system, in combination with variator slip control.

Reduction of power loss in the transmission will have a positive effect on vehicle fuel economy. By calculating fuel consumption in steady state operating points, the effect on fuel consumption with respect to the reference transmission can be judged. The calculations are based on a typical fuel ef ciency map of a 2.0 l. MPFI gasoline engine and the road load data valid for a typical passenger car. Results of these calculations, which should be regarded as indicative, are shown in the following table:

Vehicle speed	Fuel consumption reduction
[km/hr]	[%]
50	6.6
70	4.6
100	4.1
120	4.6

CONCLUSIONS

Both model calculations and test rig measurements have been used to construct a detailed ef ciency model of a commercially available CVT. By subtracting pump loss and adding loss due to the EMPAct system and by mapping variator loss as a function of over-clamping, the effect of the new actuation system and slip control on ef ciency could be estimated and an indication could be given for the consequences for fuel consumption. By applying a servo-electromechanical actuation system and variator slip control, signi ca nt reduction of power loss can be expected. This leads to a drastic improvement of ef ciency, especially at part load, which, in turn, leads to a signi can t fuel economy potential, especially on driving cycles where part load conditions prevail. Test rig evaluations will be used to prove the predicted ef ciency potential. Vehicle evaluations will be carried out in order to assess the practical use of both the EMPAct system and slip control.

REFERENCES

[1] Faust, H., M. Homm, M. Reuschel, "Ef ciency-Optimised CVT Hydraulic and Clamping System", proceedings International CVT congress, VDI berichte 1709, pp.43-58, Munich, Germany, 2002.

[2] Bradley, T.H. and A.A. Frank, "Servo-pump hydraulic control system performance and evaluation for CVT pressure and ratio control", proceedings International CVT congress, VDI berichte 1709, pp.35-41, Munich, Germany, 2002.

[3] Abo, K., M. Kobayashi, M. Kurosawa, "Development of a metal belt-drive CVT incorporating a torque converter for use with 2-liter class engines", SAE Technical Paper 980823, 1998.

[4] Changenet, C., M. Pasquier, "Power losses and heat exchange in reduction gears: numerical and experimental results", International conference on gears, Munich, 2002. VDI-Berichte 1665

[5] Ide, T., "Effect of Power Losses of Metal V-belt CVT Components on the Fuel Economy". Proceedings of the International Congress on Continuously Variable Transmission, CVT '99, Eindhoven, The Netherlands, 1999.

[6] Bonsen, B., T.W.G.L. Klaassen, K.G.O. van de Meerakker, M. Steinbuch and P.A. Veenhuizen, "Analysis of slip in a continuously variable transmission", Proceedings of IMECE03 2003 ASME International Mechanical Engineering Congress, Washington, D.C., November 15-21, 2003, IMECE 2003-41360

[7] K.G.O. van de Meerakker, P.C.J.N. Rosielle, B. Bonsen, T.W.G.L. Klaassen, N.J.J. Liebrand, Mechanism proposed for Ratio and Clamping Force Control in a CVT, in Fisita 2004; Barcelona, Spain, 10, (2004)

[8] Bonsen, B., T.W.G.L. Klaassen and K.G.O. van de Meerakker, Modelling Slip- and Creepmode Shift Speed Characteristics of a Pushbelt Type Continuously Variable Transmission, Proceedings of the International Congress on Continuously Variable Transmission, CVT '04, San Francisco, USA, 2004.

[9] Drogen, M. van, M. van der Laan, "Determination of Variator Robustness under Macro Slip Conditions for a Push Belt CVT", SAE Technical paper, 2004-01-0480, SAE world Congress, Detroit, 2003.