Extension of the geared neutral model

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Extension of the geared neutral model

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DCT 2005.43

Traineeship report

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Eindhoven, April, 2005
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1 Introduction

1.1 Goal

In the Automotive Engineering Science (AES) laboratorium, the so called "geared neutral" set up is being developed. This set up consists of a modified push belt CVT that can have a zero output speed while the input speed is non-zero, without the use of a clutch nor torque converter. Furthermore, slip in the variator has to be controlled to avoid overclamping.

To operate this set up, a control law based upon a realistic model has to be available. To this point, a dynamic model for the drive line is available. This model contains each part of the geared neutral drive line. Furthermore, a slip control strategy is included that actuates both primary and secondary clamp force.

In practice, this control strategy is not possible, for the primary and secondary clamp forces are interactive. In reality, realizing a pressure level takes a finite amount of time. The current model doesn't take this into account. Here, the clamp forces are applied with an infinite bandwidth. These problems are detrimental to a realistic model of the geared neutral behavior.

Adding a hydraulic model to the current model can simulate the geared neutral behavior in a more realistic way. The above mentioned problems will be met when a hydraulic model is added. Hence, the target of my internal traineeship is to:

"Extend the drive line model with an existing model of the hydraulic unit and develop a slip control strategy that agrees with this hydraulic model."

1.2 Partial questions

- What are the differences in the geared neutral set up, compared to a conventional push belt CVT? (section 3)
- What adaptations should be made to the drive line model with respect to the geared neutral set up? (section 4)
- What is the performance of the augmented drive line model (perform a simulation)? (section 10)
- Can the drive line model be simplified in order to evaluate it from a dynamical point of view? (section 8)

1.3 Report overview

The chapters in the report are organized as follows:

This report is split in four parts. The first part 'construction' covers an introduction to geared neutral control and the layout of the test rig. Also, the
differences between a conventional push belt CVT and the geared neutral usage is illuminated. The second part describes the drive line model and hydraulic model separately and combined. Also, the chain gears inertia is implemented.

It becomes clear that the control law can't be operated in the combined model. An alternative control strategy is suggested. This is described in part three, as well as an attempt to a dynamic description of the geared neutral set up. The fourth part describes a simulation of the final model, with the suggested control strategy from part three.

The report ends with the conclusions and recommendations.
Part I

Construction

2 Introduction to Geared Neutral

In this chapter the concept of geared neutral is explained. A complete chapter is devoted to this subject for it’s a rather complex concept and it’s important to understand it for the remaining of this report. First, a global view on geared neutral will be given. Second, the lay-out will be highlighted. Then, the constitutive relations are discussed that are used in the drive line model of geared neutral. Finally, the necessity of control of several variables related to Geared Neutral will be explained.

2.1 Historical perspective

The split torque geared neutral push belt CVT (STGN) consists of several technological features. Two features are discussed here.

The engine torque level of passenger cars is constantly increasing ([6]). A part of the engine torque is transmitted through conventional gears. The advantage is a reduced variator load and hence reduced losses occurring in the variator. A disadvantage is a reduced ratio coverage. This is taken into account by two additional clutches.

Another feature is the planetary gear set that is added to the output shaft of the CVT. The function of the planetary gear in this case is to subtract the angular velocity of both engine and the extra gears. By shifting the variator, the geared neutral situation can be obtained. In this situation the output velocity is zero. When the variator is shifted to lower or higher ratio the output speed is either positive or negative, corresponding to a forward or backward vehicle speed.

2.2 Lay-out

The STGN CVT is depicted schematically in figure 1. Like a conventional push belt CVT, the variator is at the left. Next to the variator, two conventional gears with ratio $F$ are added for the parallel torque path. The planetary gear $P$ subtracts the angular velocity of both variator, connected to sun gear, and the secondary chain gear, connected to the planet carrier. Furthermore, two clutches $C_1$ and $C_2$ are present for switching between a ‘LOW’ and ‘HIGH’ regime. In this report, clutch $C_1$ is assumed to be closed and $C_2$ is opened so the ‘LOW’ regime is active. This is necessary for geared neutral operation. The details are available in [6].
2.3 Constitutive relations

The planetary gear can be described in terms of angular velocities:

\[ \omega_{\text{sun}} = \omega_{\text{pla}} + P(\omega_{\text{pla}} - \omega_{\text{out}}) \]  

(1)

The relation between in and output speed is:

\[ \omega_{\text{out}} = \omega_{\text{in}} \frac{1}{F}[(1 + P)F - r_\omega] \]  

(2)

If \( r_\omega \) equals \((1 + P)F\), the output speed is zero for every input speed. The term \((1 + P)F\) is called "geared neutral ratio", \( r_{GN} \).

For the geared neutral situation, the control goal is to maintain a zero output speed. This can for example be done by controlling the variator ratio with relation 2. However, this is not practical due to small errors of the ratio leading to errors in the output speed with positive and negative sign. This leads to nervous behavior of a vehicle and unacceptable torque values in the transmission.

Therefore, another control variable is chosen, namely the output torque. However, the output torque is not directly measured. Instead, the output torque is derived from the clamping force acting on the pulleys in the variator by the following equation:

\[ T_{\text{sun}}(\xi) = \frac{2F_{\text{clamp}}\mu(\xi)R_{\text{sec}}}{\cos(\lambda)} \]  

(3)

Slip occurs in the variator. Therefore, slip is controlled to prevent overclamping and corresponding wear. The current slip is measured (percentage):

\[ \xi = 100(1 - \frac{r_\omega}{r_G}) \]  

(4)
Here, \( r_G \) is the so called "geometric ratio", the ratio of primary and secondary pulley radii.
3 Comparison of original and geared neutral CVT

3.1 Introduction

A regular CVT can be divided into two parts:

- the primary shaft, connected to the crankshaft via a torque converter or wet plate clutch, containing a planetary gear set and primary pulley
- the secondary shaft, containing the secondary pulley and final reduction gear and connected to a differential

For the geared neutral project, a modified push belt CVT is used. In the next sections the modifications are illuminated from both a constructional and control point of view.

3.2 From a constructional point of view

3.2.1 CVT application

A regular CVT has a torque flow from crankshaft to wheels, meeting (see figure 2): oil pump, torque converter, drive-neutral-reverse (DNR) set, primary pulley, push belt, secondary pulley, final reduction gear, differential.

Pulley actuation

Under regular circumstances the variator’s pulleys are actuated by two solenoids. The hydraulic unit covers two circuits, a primary and a secondary circuit, for the primary and secondary pulley, respectively. The secondary circuit is directly connected to the roller vane pump. The primary pressure is reduced from the
secondary pressure by the primary valve. Control of the secondary pressure is performed by the secondary valve. The ratio of the angular velocities is the variable that is controlled. This is done by controlling the secondary pressure and measure the primary and secondary angular velocities. One of the two pressures is kept at it's critical (belt slip) value, while the other is adjusted to minimize the ratio error. The pulleys in the variator differ with respect to their piston areas and allowed maximum pressure. As said before, the primary pressure is reduced from the secondary pressure. With safety valves, the maximum secondary pressure is set to 46 [bar], and the maximum primary pressure is about half times the secondary pressure, 21 [bar]. The opposite accounts for the piston areas, namely: 150 \([\text{mm}^2]\) versus 301 \([\text{mm}^2]\) (see table 3.2.1).

<table>
<thead>
<tr>
<th>Piston Area (\text{mm}^2)</th>
<th>Primary Pulley</th>
<th>Secondary pulley</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>301</td>
<td>150</td>
</tr>
<tr>
<td>Maximum allowed pressure (P_{\text{max}})</td>
<td>21 bar</td>
<td>46 bar</td>
</tr>
<tr>
<td>Resulting maximum force</td>
<td>6000 N</td>
<td>6000 N</td>
</tr>
</tbody>
</table>

Table 1: Pulley Comparison

So it can be concluded that both pulleys are capable of acting the same axial amount of force on the pushbelt.

3.2.2 Geared Neutral application

For the geared neutral application, due to the concept, the planetary gear set is required at the output axis and the torque converter has become obsolete. Therefore, the primary and secondary shafts are switched (see figure 1). And thus the primary and secondary shaft have become output and input shaft, respectively. Besides switching shafts, a parallel route is added. Two gears are attached to both input shaft and planet carrier. Because of the parallel system, the term “split torque” is used.

3.3 From a Control point of view

Because the layout of the construction is different and the aims of the geared neutral concept are different, also a different control law has to be applied.

3.3.1 CVT application

The secondary pulley prevents the push belt from slipping by clamping the sheaves with an amount of force. The stepper motor, to obtain the desired ratio, actuates the primary pulley. The primary clamp force can reach a value that is twice as high as the secondary clamp force, because of the piston surface that is also twice as large. This is necessary to reach overdrive ratio. In the CVT application, the ratio is thus controlled.
### 3.3.2 Geared Neutral application

In the Geared Neutral application, it is the amount of slip as well as $T_{\text{out}}$ that has to be controlled. In the drive line model, a slip control is present. As we shall see later, this model doesn’t work in practise. If the slip value equals its desired value, both pulley clamp forces are excited to obtain a desired output torque. When the slip value starts to deviate from the desired value, on top of this initial force, an extra amount of force is applied to control slip, which occurs on the pulley with the smallest radius (see also chapter 10). In the geared neutral situation, the primary pulley has the smallest radius.
4 Limitations

In the current setup, the CVT is used the other way around. This means that the primary pulley with the planetary gear is used as a secondary pulley and vice versa. This type of usage has several limitations, related to both construction and control.

4.1 Constructional limitations

In a normal setup, in overdrive ratio, the secondary pulley has an angular velocity that exceeds the primary one. For example, if the primary angular velocity is about 4000 rpm, the secondary angular velocity is 9800 rpm \( (r_{OD}=2.45) \). Both pulleys suffer from centrifugal forces, caused by oil in the cylinders. These centrifugal forces are computed as follows:

\[
F_{c,i} = f_{c,i} \omega_i^2 \quad i = p, s
\]

With equation (5), a graphical representation is made in figure (3).

![Graph of centrifugal forces in primary and secondary pulleys](image)

Figure 3: Centrifugal force in both primary and secondary pulley

In this graph it becomes clear that the secondary centrifugal force is little compared to the maximum clamp force due to a compensation chamber in the pulley. The primary pulley lacks such a room, hence the primary centrifugal force rises towards 16 [kN] for 6000 rpm. For high angular velocities the primary centrifugal force causes difficulty in controlling slip. The clamp force is constructed from the centrifugal force in addition to the pressure times piston
area. When the centrifugal force takes on large values, more effort has to be made to lower the pressure, in order to make it possible for the push belt to slip. For the geared neutral setup, slip occurs on the primary pulley (at the output shaft). This means that for the geared neutral setup, the centrifugal force represents a larger part of the total clamping force, thereby leaving less space for the pressure caused clamping force. For some primary angular velocities, the pressure caused clamping force should even be negative, in order to compensate for the centrifugal force and still be able to control slip. This of course is impossible. The only way to prevent this problem while maintaining this GN setup is to operate the primary pulley at lower angular velocities.

4.2 Model implications

The model file TESTRIG.MDL is written for the normal CVT application. In the GN setup however, the CVT is used the other way around. This means that labels as well as ratio definitions are rewritten. For example, in TESTRIG.MDL, Ide’s model is used to compute the geometric ratio. This is done with data from the primary pulley. Because of the alternative CVT usage, suddenly data of the secondary pulley is used. This leads to a different geometric ratio definition and therefore Ide’s model is changed. This seems trivial, but it should be done in a very careful way.

Another option for this problem is to keep the labels and definitions in tact. The hydraulic part of the model contains the right labels and definitions. Hence, combining the two models would cause communication problems. Therefore a “translating block” can be inserted, that for example inverts ratio definitions and relabels primary and secondary parameters, such as forces, pressures, radii etc. This is not recommended because it will cause a lot of confusion. Another disadvantage of this option occurs when other models are added that also contain the right labels and definitions. For those models another “translating block” has to be inserted, complicating the model even further. Therefore, the first option, rewriting the labels of TESTRIG.MDL, is chosen.
Part II  
Modeling  

5 Mechanical model  

5.1 Introduction  

A dynamic model for a STGN is available. This model is called: TESTRIG. It includes the components that were already mentioned in chapter 3.1 and is shortly described in chapter 5.3. During the traineeship, it became clear that the chain gears inertia might be of significant importance. Therefore, in section 5.2 is described how the inertia of the chain gears are modeled. Second, the algebraic equations for each drive line component are discussed after the chain gears inertia were modeled.  

5.2 Modeling chain gears  

In TESTRIG, only an inertia for the engine and an output inertia (fly wheel) are present. There are other components in the drive train that can make a significant contribution to the total inertia.  

In this chapter, it is explained how the chain gears inertia are added to the drive line model. Two options are possible:  

- adding the chain wheels inertia to the engine inertia (and thus translating the chain wheel inertia from the primary shaft to the secondary one; take into account the chain ratio) and thereby omitting an extra element in the drive train  

- adding a physical element to the drive train; this option is explained in the section below.  

The chain gears are represented by two inertia, $J_{c1}$ and $J_{c2}$, with estimated values of respectively 0.0075 and 0.0312 $[kgm^2]$. The chain itself is represented as a rigid body. Therefore, the total inertia acting on the primary axis is:  

\[ J_c = J_{c1} + F^2 J_{c2} = 0.0228 [kgm^2] \]  

In figure 4, a schematic view of the augmented CVT is shown. It includes the engine inertia, primary pulley, spring and damper, primary chain inertia, fixed ratio, secondary chain inertia. Due to torsional effects between two inertia on a shaft, additional damping and stiffness are modeled between engine inertia and chain gears inertia:  

\[ T_{shaft} = k \left( \int \omega_{engine} dt - \int \omega_{chain} dt \right) + d (\omega_{engine} - \omega_{chain}) \]
The differential equation describing the chain gears inertia \( J_c \) is:

\[
J_c \dot{\omega}_c = T_{shaft} - F T_{planet} \tag{8}
\]

Here, \( T_{planet} \) comes from the planetary gears. Equation 7 is substituted in equation 8 and the latter is implemented in the drive line model.

A simulation is done with the drive line, including the chain gears inertia. An output torque step of 400[Nm] is used as setpoint. The initial conditions are:

\[
\omega_{e,DV} = 3000\text{[rpm]}, \quad \xi = 2.5\text{[\%]}, \\
T_{out,DV} = 400\text{[Nm]}, \quad J_c = 0.0228\text{[kgm}^2]\]

The same simulation is done for the drive line model, but this time without chain gears inertia. From the results, the torques acting on and the angular velocities of the drive line components are included in this report. This comparison is made to show how the extra inertia effects the behavior of the drive line.

In figure 5 the effect of the extra inertia is obvious. The initial velocity of the engine is 3000 [rpm]. It takes approximately half a second for \( \omega_{planet} \) to settle. Then, it decelerates because the engine inertia is decelerating. The same holds for the sun gear, because \( \omega_{sun} \) depends on \( \omega_{planet} \). This is clear from equation 11 in the next section.

In figure 6 the cause of the deceleration of the engine inertia is depicted. The input torque \( T_{in} \) dominates the engine torque \( T_e \). This means by definition that the engine inertia decelerates. The high value for \( T_{in} \) can be explained by the chain gears inertia that needs to be accelerated. After three seconds, \( T_e \) dominates \( T_{in} \) again, and the engine inertia starts accelerating (see figure ??).
Figure 7 and 8 show the same simulation, except without chain gears inertia. So, only the engine inertia is left over. Again, the initial speed of the engine inertia is 3000 [rpm], which is equal to the desired value. Thus, on the start already, all angular velocities are settled. However, due to delays caused by the hydraulic unit, it takes some time for the output torque to settle. Because of the proportional action, the torque values of $T_{\text{planet}}$ and $T_{\text{sun}}$ show some overshoot. This causes $T_{\text{in}}$ to exceed $T_e$ and thereby slowing down the engine inertia from 0 to 1.7 [s].

5.3 Testrig model

The algebraic equations for the drive line components are (except the chain gears that were already mentioned):

**Engine** The engine is represented by a standard differential equation:

$$J_e \dot{\omega}_e = T_e - T_{\text{sec}} - T_{\text{shaft}}$$ (9)

**Variator** The variator is described by a mechanical equation for the sun gear torque.

$$T_{\text{sun}}(\xi) = \frac{2F_{\text{sec}}R_{\text{pri}}\mu(\xi)}{\cos(\lambda)}$$ (10)

Furthermore, the angular velocity ratio and torque ratio are computed.
Figure 6: Torque of drive line components with extra inertia

Figure 7: Angular velocity of drive line components without extra inertia
The planetary gear constitutive block consists of three relations:

\[
\begin{align*}
\omega_{\text{sun}} &= P(\omega_{\text{planet}} - \omega_{\text{out}}) \\
T_{\text{out}} &= -PT_{\text{sun}} \\
T_{\text{planet}} &= -(1 + P)T_{\text{sun}}
\end{align*}
\]  

Output inertia This component is also described by a standard differential equation:

\[J_f w_\omega \dot{\omega}_{\text{out}} + P_{\text{brake}} \omega = T_{\text{out}}\]  

Here, \(P_{\text{brake}}\) represents the gain of what can be seen as a viscous brake. This brake is activated during full stall.

Slip controller Here, the original slip controller is mentioned:

\[F_{\text{shift}} = K_P(\xi_{DV} \text{sign}(T_{sec,DV}) - \xi)\]  

Engine speed controller The engine controller is a proportional controller:

\[T_e = K_P(\omega_{e,DV} - \omega_e), \quad 0 \leq T_e \leq 36[Nm]\]
For the hydraulics of the cvt, a model is available from the AES lab. This model is reverse engineered because no data were available from the manufacturer. The model is rather complicated, and explaining it entirely is beyond the scope of this section. However, following sections require some knowledge of this model. Therefore the four main parts will be explained.

6 Hydraulic model of Jatco CK2

6.1 Introduction

For the hydraulics of the cvt, a model is available from the AES lab. This model is reverse engineered because no data were available from the manufacturer. The model is rather complicated, and explaining it entirely is beyond the scope of this section. However, following sections require some knowledge of this model. Therefore the four main parts will be explained.

6.2 Four main parts

The hydraulic model describes how the primary and secondary pressure as well as the primary valve behave and variator shifting occurs. Simplified, the model uses the stepper position, primary torque and angular velocity to compute the primary and secondary pressure, the geometric ratio and primary radius. The model consists of four main parts, two hydraulic submodels with their respective controllers, see figure 9. The four submodels will be discussed separately in the next sections.

6.2.1 Line pressure hydraulics and controller

As was explained in section 3.2.1, the secondary or line pressure is controlled by a solenoid valve. The line pressure controller (LPC) computes a duty cycle for this solenoid valve, depending on the secondary pressure and setpoint. The LPC consists of a feedforward and an integral action for error reduction. The solenoid valve is represented by equation 17, a third order low pass filter with a bandwidth of 31 [rad/s]. The input \( u \) of the transfer function is duty cycle, and the output is the line pressure. A frequency response function (FRF) calculated in Simulink (for Matlab listing of a FRF measurement, see section B.1)
Figure 10: Bode diagram of the line pressure hydraulics

is shown in figure 10 and shows a bandwidth of 25 [rad/s]. The difference in bandwidth can be explained by additional effects such as the centrifugal force that is included in the FRF measurement.

\[ P = H_{\text{line}} u, \quad \text{where}: \quad H_{\text{line}} = \frac{3.101 \cdot 10^4}{s^3 + 62.83s^2 + 1974s + 3.101 \cdot 10^4} \] (17)

Oil is present in the secondary pulley. This causes additional centrifugal forces that are also modeled and added to the line pressure:

\[ P_{\text{line}} = H_{\text{line}} u + f_{c,\text{sec}} \omega_s^2 \] (18)

6.2.2 Ratio hydraulics and controller

Another part of the variator is described in the ratio hydraulics submodel. It contains the lever, a highly non linear primary valve model (see B.2), cylinder pressurization and the pulley motion represented by Shafai’s model (see 7.3). The geometric ratio and primary running radius are computed with empiric data. Furthermore, the primary pressure is computed as a function of the valve position \( s_p \):

\[ P_p = \int \left( \frac{(q_{\text{net}} - \dot{s}_p A_{pri}) \beta}{s_p A_{pri}} \right) \] (19)

For this part a frequency response can be calculated also, but therefore the primary non linear valve should be linearized first. An attempt is made in appendix B.2.

The ratio controller consists of PID-action and uses the stepper motor as actuating mechanism.
7 Extension of the drive line model

7.1 Introduction

In the previous section the hydraulic model is discussed. As is explained in section 1.1, the existing model TESTRIG is augmented with the hydraulic model, which is described in this chapter. First, the hydraulic model is viewed as a black box. It is checked if the input signals are available from TESTRIG and the output signals from this black box are usable to TESTRIG. Then, some changes have to be made. This is described in section 7.3. Also, the control law has to be changed, see section 9.

7.2 Black Box

Instead of discussing all signals and steps done in Matlab, schematic representations are used to explain how the combination is fulfilled.

In figure 11 the hydraulic unit is depicted schematically. It is implemented in the drive line model (figure 12) as in figure 13. Below, the signals between the submodels are discussed.

The slip control unit in TESTRIG computes a setpoint for the secondary pulley clamp force, depending on the slip error and the desired output torque. By dividing with the piston area, the clamp force is converted into pressure:

\[ P_{\text{sec},DV} = \frac{1}{A_{\text{sec}}} \left( \frac{|T_{\text{out}}| \cos(\varphi)}{2R_{\text{sec}} \mu} - f_{c,\text{sec}} \omega_s^2 \right) \]  

(20)

The secondary pressure is a setpoint for the LPC, as depicted in figure 13. The LPC in the hydraulic model computes a duty cycle for the secondary solenoid. In the line pressure hydraulics, the duty cycle is transferred into secondary pressure, and this is fed back to the LPC.

Parallel to this hydraulic loop is the ratio computation. A setpoint for the slip percentage is available from a parameter file. The desired geometric ratio can be computed from this slip percentage setpoint:

\[ \xi = 100 \left( \frac{r_\omega}{r_g} - 1 \right) \Rightarrow r_g = \frac{r_\omega}{0.01 \xi + 1} \]  

(21)

The ratio controller subtracts the desired geometric ratio from the existing ratio value and computes an actuating signal for the stepper motor. Both the secondary pressure as well as the stepper motor position are inputs for the ratio hydraulics submodel. The output of this model, the geometric ratio and primary radius, are send to the drive line block of TESTRIG.
Figure 11: Black Box representation of the hydraulic model

Figure 12: Original lay out of TESTRIG

Figure 13: Schematic representation of final model
7.3 Transient Variator Model

7.3.1 Introduction

In the mechanical model as well as in the hydraulic model, a transient variator model is present. In the combined model, only one transient variator model can be used. Because of the large impact the transient variator model has on the total dynamics, it is important that both models are accurately compared and a choice is made. In the following section, the term “transient variator model” is explained and two models are outlined. Finally, one model is chosen.

7.3.2 Definition

One can model a variator as a mechanical system, which consists of three rotational elements, namely a push belt and two pulleys. In this case the pulleys are modeled as two inertia and the push belt as an element with damping and stiffness. For the steady state situation this can be satisfactory. In the transient situation however, changing ratio of the variator will take a finite amount of time, caused by damping in the variator. Therefore, it is important that a transient model is also included.

Two transient variator models are outlined that compute the time derivative of the geometric ratio:

- Ide’s model

\[
\dot{r}_g = k_r \left( r_g \right) \left( F_p - F_p^* \right) \omega_p
\]  

- Shafai’s model

\[
m_p \ddot{s}_p = F_p - F_p^* - b_p \dot{s}_p
\]

In TESTRING.MDL, Ide’s model is used to calculate the geometric ratio, while HYDRAULICS.MDL uses Shafai’s model to compute the primary pulley position. [1] mentions also Guebeli’s model.

A choice has to be made between Ide’s and Shafai’s model. An advantage of Ide’s model is, that it is dependent of the primary angular velocity, in such a way that shifting is not possible with zero angular velocity. But for the geared neutral test rig, this advantage probably has no significant influence, for the secondary angular velocity will be of order 3000 [rpm]. Shafai’s model has the form of a differential equation, which includes terms representing mass and damping. From a dynamic point of view, shafai’s model is favorable due to the differential equation. Therefore, shafai’s model is chosen. When the test rig is ready to be operated maybe some experiments can be done to compare different transient variator models in the geared neutral case, especially for the steady state situation.
Part III
Control

8 Dynamics

When the hydraulic model is inserted, it would be favorable to evaluate the model from a dynamic point of view. The current hydraulic model consists of two models (see section 6.2):

a. Line pressure hydraulics model
b. Ratio hydraulics model

The first model is linear and single input single output (SISO). The latter is non linear and multi variable. Therefore some assumptions are made to simplify the ratio hydraulics model:

a. Only Shafai’s model is used for the approximation
b. The primary valve is linearized
c. Three inputs are modeled: Primary pressure $P_p$, $F^*_p$ and $\omega_p^2$
d. The third input $\omega_p^2$ is linearized at $3000 \ [rpm] = 314.2 \ [rads^{-1}]$.

Shafai’s model (23) on page 23 is augmented with the primary centrifugal force:

$$m_p \ddot{s}_p = F_p - F^*_p - b_p \dot{s}_p + 2f_{c,pri}\omega_p^2$$

and transferred into state space form:

$$x = \begin{pmatrix} s_p \\ \dot{s}_p \end{pmatrix} \Rightarrow \dot{x} = \begin{pmatrix} \dot{s}_p \\ \ddot{s}_p \end{pmatrix}, \ u = \begin{pmatrix} P_p \\ F^*_p \\ \omega_p \end{pmatrix}$$

Then

$$\ddot{x} = \begin{pmatrix} 0 & 1 \\ 0 & -\frac{b_p}{m_p} \end{pmatrix} \ddot{x} + \frac{1}{m_p} \begin{pmatrix} 0 & 0 & 0 \\ 10^5 A_{pri} & -1 & 2f_{c,p} \end{pmatrix} \ddot{u}$$

and the output is defined as follows:

$$\ddot{y} = \begin{pmatrix} 0 & 0 \end{pmatrix} \ddot{x} + \begin{pmatrix} 0 & 0 & 0 \\ 0 & 0 \end{pmatrix}$$

The large number $10^5$ is used to convert [Pa] into [bar]. The number 1000 in the output equation is used to convert from [m] to [mm]. A transfer function is computed from input 1 ($P_p$) to output $y$.

It is important to know how fast ratio changes can take place. Therefore the bandwidth of the hydraulic model is computed. Because the hydraulic model
is very augmented, some steps have to be made to simplify the model. It consists of various hydraulic sub models, controllers and kinematic relations and each hydraulic sub model has several inputs and outputs. For the drive line part, only two variables from the hydraulic part are of importance, namely the secondary pressure and the geometrical ratio (see figure 9). Therefore, the line pressure hydraulics and ratio hydraulics are modeled as SISO-systems, with duty cycle and secondary pressure respectively, as inputs, and secondary pressure and geometrical ratio respectively, as outputs. The transfer from duty cycle to secondary pressure that represents the line pressure hydraulics (LPH), is simply a third order low-pass butterworth filter.

The transfer from secondary pressure to geometric ratio that represents the ratio hydraulics (RH), is more complicated. This transfer consists of the primary valve, shafai’s model etc. The valve is modeled in a continuous way, described by B.2 on page 41. In this way the transfer $H_2$ becomes a linear function, and a frequency response function (FRF) can be computed. This can be done by applying white noise for the input and measuring the output. From the input and output vectors, an transfer can be computed using “tfe” (transfer function estimate). For the file listing, see appendix B.1.

The hydraulic models are replaced with their transfer functions LPH and RH. From these functions the bandwidth value can be estimated, by multiplying LPH and RH to obtain the open loop transfer.
9 Control

The secondary or line pressure is generated by the oil pump and controlled directly by the line pressure controller. The primary pressure depends on the position of the primary valve and the secondary pressure and can’t be controlled due to the construction of the primary valve. Shortly spoken:

\[ F_p = F_p(s_v, p_s), F_s = F_s(\text{DC}) \]  

(25)

However, in the original drive line model, it is suggested that both primary and secondary clamp forces can be computed and controlled separately:

- If the amount of slip is larger than the desired value (too much slip), the primary pulley clamp force is raised.
- If the amount of slip is smaller than the desired value (too little slip), the secondary pulley clamp force is raised.

In theory this control law works well, but it is difficult to implement in practice. It suggests that both clamp forces can be controlled independently. In reality, the primary pressure is dependent on the secondary pressure.

As can be seen in picture 9 there is one input that can be actuated (the secondary solenoid valve) and there are two outputs (the primary and secondary pressure):

This causes the TESTRIG system to be underactuated (i.e. the number of to be controlled quantities exceeds the number of actuators). So, in TESTRIG, the computed primary and secondary clamp forces, should be regarded as desired values, or setpoints, and the hydraulic part should take care that the true values of the clamp forces converge towards these setpoints. Slip is computed with the following function:

\[ \xi = 100 \left( 1 - \frac{r_w}{r_g} \right) \]  

(26)

\( r_w \) is almost constant because of the rigid connection between the sun gear and engine axis, therefore controlling slip basically comes down to controlling \( r_g \), which results from the numerical integral of Shafai’s model. Rewriting equation 26:

\[ r_g = \frac{100r_w}{100 - \xi} \]  

(27)
In the geared neutral application, slip control is possible in two different ways: either the primary or secondary clamp force can be controlled.

9.1 Control of the primary clamp force

A setpoint for the primary clamp force is computed to control the amount of slip, while the secondary clamp force prevents the push belt from slipping. This primary pressure is controlled by the secondary pressure. Hence no secondary pressure setpoint is needed.

Also, the output torque has to be controlled. This output torque is determined by the transmitted torque in the variator, which is a function of the lowest clamp force:

\[ T_{sun}(\xi) = \frac{2F_{min,R_{pri}}\mu(\xi)}{\cos(\lambda)} \]  

Thus by controlling the lowest clamp force, the desired output torque is guaranteed.

Control of the primary pressure can be done as follows:

A setpoint for the primary clamp force is generated by the force interchanger. This setpoint consists of two parts:

**Basic clamp force** For the steady state situation, a basic clamp force is computed that equals the desired output torque and computed by the following function:

\[ F_{basic}(\xi) = -\frac{T_{out,DV}\cos(\lambda)}{2P_{\mu}(\xi)R_{pri}} \]  

**Shift force** On top of the basic clamp force, a shift force is added that is a function of the slip error. The shift force is computed by multiplying the slip error by the sum of a proportional and integral action:

\[ F_{shift} = K_P\varepsilon_\xi + K_I\int(\varepsilon_\xi)dt \]  

Where:

\[ \varepsilon_\xi = r_\xi \text{sign}(T_{pri,DV}) - y_\xi \]  

In this situation, the secondary pressure is lowered or raised until the primary pressure equals the desired primary pressure:

- If the amount of slip is larger than the desired value (too much slip), the primary pulley clamp force is lowered.
• If the amount of slip is smaller than the desired value (too little slip), the primary pulley clamp force is raised.

This method seems most straightforward, because slip occurs at the primary pulley. However, it has some disadvantages. As said before, the primary pressure is derived from the secondary pressure, or:

\[ p_p \leq p_s \]  

(32)

So, for large primary setpoints, the secondary pressure has to be large also. Simulations point out that, for large setpoints, the engine can’t handle the transmitted torque in the variator which causes the inertia to slow down. The differential equation for the engine inertia:

\[ J_e \ddot{\theta} = T_e - (T_{sec} + T_{chain}) = T_e - (T_{sec} - (1 + P)FT_{pri}) \]  

(33)

shows that the engine slows down if the expression at the right side is less than zero. This is caused by the large values for \( T_{pri} \) and \( T_{sec} \), both depending on the smallest clamp force:

\[ T_{pri} = T_{pri}(\min(F_{pri}, F_{sec})) \]  

(34)

\[ T_{sec} = T_{sec}(\min(F_{pri}, F_{sec})) \]  

(35)

A second disadvantage is the unpredictable behavior of the primary valve. This valve is used to control the angular speed ratio but has a great unpredictable effect on the primary pressure too.

9.2 Control of the secondary clamp force

Control of the secondary clamp force would be the most ideal situation, because the pressure on the secondary pulley is directly measured and controlled. In this situation, the secondary pressure is changed until \( r_g \) equals \( r_{g,DV} \). This is an indirect method because slip occurs at the opposite (primary) pulley. For the push belt is modeled rigidly, this method should be possible.

• if the amount of slip is larger than the desired value (too much slip), the secondary pulley clamp force is lowered.

• if the amount of slip is smaller than the desired value (too little slip), the secondary pulley clamp force is raised.

Simulations of this method point out that the percentage of slip converges towards 16%. For different control parameters, such as raising the proportional action, the same percentage of slip is obtained.

From previous experiments it is known however, that the stepper motor of the primary pulley has a significant influence on the geometric ratio, of which slip is derived. Thus by adjusting the stroke of the stepper motor, for example
by means of the ratio controller, the desired slip percentage can be obtained. When this is done, the slip percentage does converge towards the desired value. Therefore, the control strategy consists of controlling both the secondary clamp force and the stepper motor.

9.3 Implementation of controllers

9.3.1 Stepper motor controller

As explained in the previous section, slip is controlled by controlling both the stepper motor and the secondary pressure. A controller for the stepper motor is already included in the hydraulic model. The difference is that instead of the angular velocity ratio \( r_\omega \), the geometric ratio \( r_G \) is used as feedback signal.

As depicted in figure 14, the core of this controller consists of feedforward (red) combined with an anti-windup PI-controller (green):

\[
pos_d = K_P (r_{DV} - r) + K_I (\int r_{DV} - \int r) + K_F (r_{DV})
\]

(36)

The feedforward is computed out of the geometric ratio setpoint \( (r_d) \) and the feedback depends on the error signal \( (r_d - r) \). Both efforts are summed, leading to the total control signal which is a input for the stepper motor \( (pos_d) \). Three saturation functions (cyan) are included also. They constrain the domain of each parameter so they can’t take on unrealistic values. The saturation function at the right side constrains the stepper motor excitation and is critical to the anti windup controller. When an excitation limit is reached, the anti windup is activated.

The original controller of the drive line model computes a clamp force in order to obtain a desired output torque. Second, an additional clamp force is added for slip control purposes. In the new control strategy, the former has remained unchanged. The latter is substituted by the ratio controller of the hydraulic unit (figure 14).

9.3.2 Secondary pressure controller

The controller for the secondary or line pressure (LPC) was already mentioned in section 6.2.1. It is also a PI-controller with feed forward, see figure 15. Here, the output of the feed forward (red) and the PI-controller (green) are summed and form the duty cycle for the secondary solenoid, at the right of the picture:

\[
DC_{LP} = K_P (P_{sec,DV} - P_{sec}) + K_I (\int P_{sec,DV} - \int P_{sec}) + K_F (P_{sec,DV})
\]

(37)
Figure 14: Ratio controller of the hydraulic model

Figure 15: Controller of the secondary or line pressure
9.4 Conclusion

From the above mentioned ways of control, the following can be concluded. In a normal CVT, overclamping prevents the push belt from slipping. In the GN setup, slip is necessary to control the output torque and prevent overclamping. This means that for zero output speed, the angular velocity ratio $r_\omega$ should equal $r_{GN}$ and the variator has to operate in overdrive. The smallest radius occurs at the primary pulley (at the output shaft). In this situation it would be logical to control the primary pulley, but this has some disadvantages. It is decided to control the secondary pressure because this can be done in a straightforward way. Controlling both pressures isn’t favorable because of the interaction between the pressure values. Besides control of the secondary pressure, the stepper motor is used to control the geometric ratio making it possible for the percentage of slip to converge towards the desired value.
Part IV

Simulation

10 Simulation

After finishing the GN setup, some experiments will be done. One of these experiments will be transferring power from the input inertia (engine) to the output inertia (flywheel), to simulate a car accelerating from stand still. The goal of this experiment is to see how the slip controller behaves during this acceleration. The focus will be on the slip percentage during the simulation, as well as on the output torque and angular velocity. For a more detailed discussion of how to apply a road load envelope, see appendix A.

10.1 Preconditions

The engine that is represented by this input inertia can deliver up to 36 Nm torque. A larger value can be obtained by slowing down the engine inertia. This causes an extra torque generation and makes it possible to transfer more power to the output inertia. The experiment follows the next sequence:

a. \( t=0 \) The engine is started, while maintaining the ratio in the GN position \([1+P]F\)

b. \( t=2 \) When engine inertia reaches maximum rotational velocity, full stall is applied (400 Nm)

c. \( t=4 \) When the output torque is steady, the brake is released, causing the output inertia to accelerate

d. \( t=5.2 \) When output inertia reaches particular rotational velocity, the brake is applied again, to arrive in the original situation

e. The experiment can be repeated in reverse order

In picture 16, the above sequence is represented schematically. The first action \((a-d)\) simulates a stall experiment. The reverse action \((e)\) simulates braking with regeneration of power, which is saved in the engine inertia and available for accelerating afterwards. The experiment described in this chapter covers the first action only.
The results are plotted in three figures. Each figure is discussed in a separate section and unexpected behavior is illuminated.

10.2 Results

The results are plotted in three figures. Each figure is discussed in a separate section and unexpected behavior is illuminated.

10.2.1 Torque

In figure 17 the results with respect to the torque values are shown. In the first two seconds a minimum amount of torque is applied that is limited by the centrifugal forces and a minimal pressure in the secondary circuit. After about two seconds the situation has become steady state and the clamp force is raised, leading to a torque step of 400 [Nm] on the output inertia (blue). Due to delays in the hydraulic circuit, it takes about four seconds before the output torque reaches it’s maximum value (green). From four seconds, the engine torque (red) is lower than the reaction torque on the engine inertia (cyan). As will be seen in the next graph, this inequality causes the engine inertia to slow down. The output torque equals zero from 4.5 seconds until 5.2 seconds. Then, the output torque reaches the maximum value again and the engine and reaction torques are equal.
The angular velocity of two components plotted in figure 18 is discussed. The first, the output inertia, is accelerating from four seconds. At 4.5 [s], the output torque maintains zero, causing the acceleration to diminish. When the brake is applied again at 5.2 [s], the output inertia slows down. Second, as mentioned in the simulation conditions, the engine inertia delivers torque from 4 [s], causing the inertia to slow down.

Figure 20: Torque value for various components

10.2.2 Angular Velocity

The angular velocity of two components plotted in figure 18 is discussed. The first, the output inertia, is accelerating from four seconds. At 4.5 [s], the output torque maintains zero, causing the acceleration to diminish. When the brake is applied again at 5.2 [s], the output inertia slows down. Second, as mentioned in the simulation conditions, the engine inertia delivers torque from 4 [s], causing the inertia to slow down.

10.2.3 Ratio & Slip

Due to a negative output torque, the slip sign is positive, as depicted in figure 20. The slip setpoint is 2.5 [%]. First, this is asymptotically approached. This is caused by damping in the variator that prevents a fast convergence. Then, in the section without any brake torque (4-5.2 [s]), the slip value goes to zero. In this section, by definition, the angular velocity ratio equals the geometric ratio as can be seen in figure 19, while the actuator is saturated. The latter means that the stepper motor is excited 100[%]. Maybe this behavior can be explained by the large amount of damping in the variator, so the variator can’t keep up with the ratio change velocity (the derivative of the ratio).
Figure 18: Angular velocities for various components

Figure 19: Ratio values
From the experiment, some conclusions can be drawn. First, slowing down the engine inertia and thereby delivering torque to the drive line is a realistic option. The angular velocity drops from 1000 [rpm] to about 800 [rpm] in 0.5 [s], so theoretically, the experiment can last < 2.5 [s]. Second, the slip value converges towards zero when the brake is released.

10.3 Conclusion

From the experiment, some conclusions can be drawn. First, slowing down the engine inertia and thereby delivering torque to the drive line is a realistic option. The angular velocity drops from 1000 [rpm] to about 800 [rpm] in 0.5 [s], so theoretically, the experiment can last < 2.5 [s]. Second, the slip value converges towards zero when the brake is released.
11 Conclusion

A number of problems had to be dealt with after augmenting the mechanical model with the hydraulic unit. To successfully connect both models, changes had to be made to the transient variator model, the control strategy, the computation of geometric ratio, angular velocity ratio and secondary clamp force setpoint.

After simulating both Shafai’s and Ide’s model for a transient variator, it was shown that the differences were small. An advantage of Ide’s model is the angular velocity term that makes shifting at zero speed impossible. An advantage of Shafai’s model is the differential notation. Because the angular velocity remains constant at a speed of 3000 [rpm], the advantage of Ide’s model cancels and therefore Shafai’s model is chosen.

With respect to the control strategy it is shown that the strategy of the drive line model is difficult to implement when taking the hydraulic unit into account. In the hydraulic unit, only the secondary pressure (line pressure) is controlled. Therefore, it is most straightforward to control the secondary clamp force.

The ratio controller of the hydraulic unit was first used to control the angular velocity ratio. Simulations point out that the geometric ratio (and thus the slip percentage) depends on the ratio controller. Therefore, the ratio controller is used to control the geometric ratio instead of the angular velocity ratio.

Some final simulations point out that:

- while accelerating, the ratio controller is incapable of maintaining the desired geometric ratio (slip percentage)
- the chain gears inertia can have a significant influence on the total inertia
- the engine flywheel is capable of delivering a sufficient torque if an acceleration (section 10) is simulated

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Part V
Appendices

A Road Load
The GN technique in combination with a push belt can in the future be operated in road vehicles. Thus, it would be logical to simulate a road load profile, to see how the GN transmission responds to resistance torques.

A.1 Output Inertia
At the output ring wheel an output inertia is placed that represents the vehicle’s wheels. When no braking nor damping is present, setting the output torque to zero the output inertia should maintain zero angular velocity. In the transient situation there is unfortunately some fluctuation of the output torque. This causes the output inertia to speed up. In steady state the output torque becomes zero and the inertia stays rotating at a constant velocity. In normal situations, in bearings, damping is present. This damping results in a damping torque unless the angular velocity is zero. When the damping torque is larger than the output torque, the output angular velocity will go to zero.

A.2 Road load envelope
In this section the road load envelope is constructed. The envelope is a representation of the resistance the vehicle will meet when operated on a horizontal road. “Horizontal” means that no gradient resistance is taken into account. Hence, only three types of resistance are left:

- Air resistance
  \[ T_L = \left( \frac{1}{2}c_w \rho L A_v \omega^2 \right) R_w^3 \]  
  (38)

- Wheel resistance
  \[ T_R = (f_R m_v g \cos(\alpha)) R_w \]  
  (39)

- Acceleration resistance
  \[ T_a = J_v \dot{\omega} \]  
  (40)

Which leads to a total resistance:

\[ T_{total} = T_L + T_R + T_a = \left( \left( \frac{1}{2}c_w \rho L A_v \omega^2 \right) R_w^2 + (f_R m_v g \cos(\alpha)) \right) R_w + J_v \dot{\omega} \]  
(41)

For a graphic representation, see fig. 21.
To simulate a road load profile, a brake device is needed. The test rig already contains a disc brake. Unfortunately, a disc brake can’t be used for this purpose because actuating a disc brake continuously will cause a lot of heat. Furthermore, a disc brake is hard to control due to highly nonlinear behavior. So, another device is considered: An Eddy-current brake (Du: Wervelstroomrem) can be used. This is an electric mechanism, consisting of a rotor attached to the output axis axially and a stationary coil mounted to a fixed housing.

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
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<td>10</td>
<td>Coil</td>
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<td>11</td>
<td>Rotation sensor</td>
</tr>
<tr>
<td>12</td>
<td>Support springs</td>
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</table>

Figure 21: Road load envelope (cumulative)

A.3 Eddy-current brake

To simulate a road load profile, a brake device is needed. The test rig already contains a disc brake. Unfortunately, a disc brake can’t be used for this purpose because actuating a disc brake continuously will cause a lot of heat. Furthermore, a disc brake is hard to control due to highly nonlinear behavior. So, another device is considered: An Eddy-current brake (Du: Wervelstroomrem) can be used. This is an electric mechanism, consisting of a rotor attached to the output axis axially and a stationary coil mounted to a fixed housing.
Figure 22: Eddy-current brake lay out
B  Matlab listings

B.1  FRF measurements

In section 8 an estimate is made for the transfer of the ratio hydraulics. In this section, the listing is included. The same sort of measurement is done in section 6.2.1.

If a frequency response function (FRF) has to be made of object A, white noise is applied at the input signal \(X\) of object A while the output signal \(Y\) is measured.

Then, the input and output vectors \(X\) and \(Y\) are loaded into the workspace:

\[
\text{load } X \\
\text{load } Y
\]

A power spectral density is computed and plotted to investigate the coherence between \(X\) and \(Y\):

\[
[P,F]=\text{specturm}(X,Y); \\
\text{specplot}(P)
\]

"tfe" computes an estimate for the transfer function:

\[
[FRF,W]=\text{tfe}(X,Y);
\]

"frfit2" is a graphical programme to fit the experimental data. The order of the numerator and denominator and the number of integrators are defined by the third term \([2 1 0]\), respectively:

\[
[\text{num,den}]=\text{frfit2}(FRF,W,[2 1 0]);
\]

From the obtained estimate, a transfer function is assembled:

\[
H = \text{tf}([\text{num}], [\text{den}]); \\
\text{bode}(H)
\]

B.2  Valve fit

The primary circuit consists of a valve that gives access to both drain and primary circuit. Excitation of this valve causes a flow, which leads to a primary pressure. In the hydraulic part, the shape of the valve is modeled accurately. Thereby the valve is divided into eight discontinuous parts (23). This discontinuous representation of the valve produces two disadvantages:

- Simulating with Matlab takes a long time
- Modeling the valve in a dynamic way is impossible

To make a continuous model of the valve, different methods are available. There are some constraints, that make some parts of the fit more important
than other parts:

- For an excitation of 0 [mm], the flow should be zero (except leakage). This is the steady state position of the valve. The primary pressure is partly the integral of the volume flow. When the volume flow is non-zero in steady state condition, the pressure keeps rising.

- At the lower and upper bounds, the derivative of the volume flow should be zero.

This is difficult to accomplish with the least squares approach. Therefore, another fit method should be used.

Figure 23: Valve position
C  Recommendations

Continue of the geared neutral project
Unlike the hydraulic unit, the combined model as described in this report hasn’t been validated. The results in this report are purely based on computer simulations. It is important to validate the model in practise to see if the implemented control law functions correctly and if the assumptions are justifiable. For this experiment, knowledge with respect to both automotive science as well as control science is necessary.

The validation of the model can be done on the geared neutral set up. Situations that can be test are:

- The geared neutral point
- Acceleration out of the geared neutral point

During this traineeship, some shortcomings in the current models and points of further investigation were noted and are listed below.

- Due to the geared neutral operation, the angular velocity of the primary pulley, and therefore the centrifugal force, is much higher than in the original CVT. If $\omega_{pri}$ exceeds a certain value (depending on $\omega_{engine}$ and $r_w$) the clamp force causes overclamping. When this happens, slip control is impossible. The maximum primary angular velocity $\omega_{pri}$ should experimentally be obtained by operating the variator in overdrive and accelerating the engine until the slip value increases. Then, this value can be added to the model as a constraint for $\omega_{pri}$.

- In this report the inertia of the chain gears is added to obtain a more complete dynamic model of the drive line. However, only a estimated value is used. Furthermore, the inertia of the pulley’s as well as the planetary gear is not included. For an accurate dynamic model, all inertia should be measured and included in the model.

- In section 7.3 on page 23 two transient variator models are discussed, from which only Shafai’s model is used in the final GN setup model. In the future also Ide’s and Guebli’s model can be tested.

- In section 8 an attempt to a dynamic evaluation is made. The goal was to linearize the models around a certain state. However, for the primary valve, which is a discrete function of the valve position, linearization is difficult. This has to be finished, in order to compute robustness and bandwidth of the controller.
## D Nomenclature

<table>
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<th>Symbol</th>
<th>Name</th>
<th>Unit</th>
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</tr>
<tr>
<td>A</td>
<td>surface</td>
<td>m²</td>
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<td>$A_v$</td>
<td>vehicle frontal surface</td>
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<td>$b_p$</td>
<td>primary pulley damping coefficient</td>
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<td>Viscous brake gain</td>
<td>$N s rad^{-1}$</td>
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<tr>
<td>$pb_{cd}$</td>
<td>push belt center distance</td>
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<td>$pb_L$</td>
<td>push belt length</td>
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<tr>
<td>$q_{net}$</td>
<td>Volume flow through primary valve</td>
<td>$m^3 s^{-1}$</td>
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<td>$r$</td>
<td>reference signal</td>
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<tr>
<td>$r_\omega$</td>
<td>ratio of angular velocity</td>
<td>-</td>
<td>0.438 - 2.28</td>
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<tr>
<td>$r_g$</td>
<td>geometric ratio</td>
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<td>$r_{GN}$</td>
<td>geared neutral ratio</td>
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<tr>
<td>$R_i$</td>
<td>pulley running radius</td>
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<td>$R_w$</td>
<td>Wheel radius</td>
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<td>$s_p$</td>
<td>axial primary pulley position</td>
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<tr>
<td>$s_v$</td>
<td>primary valve position</td>
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<tr>
<td>$T_e$</td>
<td>engine torque</td>
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<tr>
<td>$T_{sun}$</td>
<td>sun wheel torque</td>
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<td>chain torque (same axis as engine)</td>
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<tr>
<td>$T_{sec}$</td>
<td>secondary pulley torque</td>
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<tr>
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<td>$\beta$</td>
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<td>friction coefficient</td>
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<tr>
<td>$\xi$</td>
<td>slip percentage</td>
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<td>$\rho_l$</td>
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<td>$\omega$</td>
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<td>r a d$^{-1}$</td>
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References


