Control of a Flywheel Assisted Driveline With Continuously Variable Transmission

This paper proposes a control solution for a vehicular driveline with an internal combustion engine, a continuously variable transmission and an additional flywheel unit. This unit plays a part only in transient situations. It compensates for the engine inertia, enabling optimal fuel economy in stationary situations without losing driveability during transients. For control design, a simple, nonlinear model is developed and used for feedback linearization. The proposed controller is evaluated by simulations, using an advanced simulation model. The compensation of the engine inertia by the additional flywheel is demonstrated by vehicle experiments. [DOI: 10.1115/1.1589033]

Keywords: Driveline Dynamics, Fuel Economy, Nonlinear Control, I/O Linearization

1 Introduction

Reduction of both fuel consumption and exhaust emissions of vehicles has been subject of much research in the recent past, e.g., [1–6]. The engine efficiency is maximal if the driveline management system (DMS) forces the engine to operate along its maximum efficiency line (E-line), see [7]. This requires a continuously variable transmission (CVT) with a sufficiently large ratio coverage. The main drawback of a DMS of this type is that it results in an unacceptably poor driveability if the engine displacement is small [8–10].

Generally, the efficiency is high if the power is delivered at low speed (large transmission ratio) and large torque (nearly wide open throttle). In this so-called E-line strategy, only a small increase of the engine power will be obtained by completely opening the throttle. A further increase of this power is possible only if the engine is speeded up by downshifting the CVT. A fast downshift will result in a large engine acceleration. The engine then may not have enough power to accelerate its own inertias and power will be withdrawn from the vehicle, resulting in a vehicle deceleration. This is seen as a serious drawback of the E-line strategy, and as such it is hardly applied in practice. Instead, lower torque and higher engine speed are—despite the lower combustion efficiency—chosen in order to guarantee more vehicle acceleration promptness.

To overcome the discrepancy between driveability and fuel consumption a power assist unit [11–13], embodied as a flywheel and a planetary gear set in parallel to a standard CVT driveline, is developed (see Fig.1). The sun gear of this set is connected to the flywheel whereas the annulus gear and the planet carrier are connected to, respectively, the engine shaft and the wheels. If the CVT is shifted down, the flywheel speed decreases at an increasing engine speed. The resulting decrease of the kinetic energy of the flywheel can be used to accelerate the engine. From a model point of view, it seems that the engine inertia is cancelled by the flywheel inertia. Therefore, the new driveline is called Zero Inertia or ZI driveline. For an upshifting CVT, the flywheel speed increases at a decreasing engine speed. The kinetic energy from the decelerating vehicle and engine inertias then is partly absorbed by the accelerating flywheel and this is beneficial for an instant coast down feeling at drive pedal back-outs.

The ZI driveline is designed, built and implemented in a mid-sized passenger car. The flywheel unit introduces additional nonlinearities in the driveline, making it a more challenging system to control. In this paper, a technique similar to feedback linearization is used to deal with these nonlinearities.

The remainder of this paper is organized as follows. In Section 2, the driveline, the control design model for this driveline and the ZI solution for driveability are described. After a discussion on the requirements for the driveline management system (DMS) in Section 3, the control law for the CVT rate of ratio change is presented in Section 4. The control law is evaluated by means of simulations, using a realistic simulation model of the driveline. Some simulation and experimental results are presented and discussed in Section 5. Finally, Section 6 gives the main conclusions and some suggestions for future research.

2 Model of the Flywheel Assisted Driveline

The considered front wheel driven vehicle moves along a straight line with angular wheel speed \( \omega_w \). The driveline (see Figs. 1 and 2) consists of an internal combustion engine (ICE), a torque converter, a drive-neutral-reverse (DNR) set, a transmatic CVT with a metal pushbelt, a final reduction, a differential, a planetary set and an extra flywheel. First the kinematics of the driveline are analyzed. After that, the equations of motion for the engine and the ZI driveline are given.

It is assumed that the torque converter is locked and that the DNR set is in drive mode, so the primary CVT pulley speed \( \omega_{p1} \) is equal to the engine speed \( \omega_e \). The secondary pulley speed \( \omega_{sc} \) is given by \( \omega_{sc} = r_f \omega_{p1} \omega_{pe} \omega_{sc} \) where the continuously variable CVT ratio \( r_{cv} \) is bounded by \( r_{low} \) and \( r_{up} \) with, typically, \( r_{low} = 0.4 \) and \( r_{up} = 2.5 \). The secondary pulley is connected to the wheels via the final reduction and the differential, so \( \omega_{w} = r_f \omega_{sc} \) with constant gear ratio \( r_f \).

The speeds \( \omega_n \) of the annulus gear, \( \omega_c \) of the carrier and \( \omega_s \) of the sun gear of the planetary set (with basic gear ratio \( z \)) are related by \( \omega_n = (1+z)\omega_s - z\omega_c \). The flywheel (with speed \( \omega_f \)) is rigidly connected to the sun gear, so \( \omega_f = \omega_s \). The annulus gear is connected to the primary pulley via a gearing with fixed gear ratio \( r_p \), so \( \omega_n = r_p \omega_{p1} \omega_{s} \). Finally, the carrier is connected to the secondary pulley via a gearing with fixed gear ratio \( r_c \), so \( \omega_{sc} = r_c \omega_{c} \). Combination of these relations results in

\[
\begin{align*}
\omega_n &= r_f \omega_{p1} \\
\omega_f &= (a_r - a_c) \omega_s \\
\omega_w &= r_f \omega_{c} \\
\omega_{sc} &= r_c \omega_{sc}
\end{align*}
\]

where \( \alpha_r = \frac{z}{r_p} \) and \( \alpha_c = \frac{(1+z)}{r_f r_c} \) are constant whereas \( r = r_f r_{cv} \) is the variable transmission ratio from engine to wheel.
This overall transmission ratio is bounded by \( r_{\min} = r_{f_{\text{low}}} \) and \( r_{\max} = r_{f_{\text{ad}}} \). The flywheel is at rest if \( r = r_{g_{\text{a}}} = \alpha_{e} / \alpha_{w} \), where \( r_{g_{\text{a}}} \) is the so-called geared neutral ratio.

The equation of motion for the engine side is given by

\[
J_{e} \frac{d\omega_{e}}{dt} = T_{e} - T_{\text{pri}} - r_{a} T_{a}
\]  

(2)

with moment of inertia \( J_{e} \) (engine, torque converter, DNR set, primary pulley, gearing \( r_{a} \) and annulus gear), induced engine torque \( T_{e} \), torque \( T_{\text{pri}} \) from the CVT belt on the primary pulley and torque \( T_{a} \) from the carrier on the annulus gear.

The engine speed \( \omega_{e} \) is bounded by \( \omega_{\min} = 100 \text{ [rad/s]} \) and \( \omega_{\max} = 580 \text{ [rad/s]} \). The stationary engine torque \( T_{e} \) depends on \( \omega_{e} \) and the throttle opening \( \varphi \). In the control design model it is assumed that this relation may also be used in instationary situations. For each \( \omega_{e} \), the engine torque is upper bounded by the wide open throttle \( \omega_{\text{wot}} \) torque \( T_{\text{wot}}(\omega_{e}) \), see Fig. 3. Any \( T_{e} \in [0, T_{\text{wot}}(\omega_{e})] \) can be realized by an appropriate setting of \( \varphi \). The considered driveline is equipped with an electronic throttle, so practically decoupling the relation between \( \varphi \) and the position of the drive pedal.

The set of maximal fuel economy operating points is called the E-line or maximum efficiency line (see Fig. 3). On the interesting part of this line, \( \omega_{e} \) and \( \varphi \) are functions of the engine braked output power \( P_{e} \), i.e.,

\[
\omega_{e} = \omega_{\text{mel}}(P_{e}); \quad \varphi = \varphi_{\text{mel}}(P_{e}); \quad P_{e} \in [0, P_{\max}]
\]  

(3)

The function \( \omega_{\text{mel}}(\cdot) \) is strictly increasing, implying that the engine power and the engine torque in operating points on the interesting part of the E-line are functions of \( \omega_{e} \), so

\[
P_{e} = P_{\text{mel}}(\omega_{e}); \quad T_{e} = \frac{P_{e}}{\omega_{e}} = T_{\text{mel}}(\omega_{e})
\]  

(4)

The equation of motion for the load side of the power train (carrier, gearing \( r_{c} \), secondary pulley, final reduction, differential and vehicle inertia) is given by

\[
J_{w} \frac{d\omega_{w}}{dt} = T_{w} - T_{\text{ext}}; \quad T_{w} = r_{c} T_{c} - T_{\text{sec}}
\]  

(5)

with moment of inertia \( J_{w} \), drive shaft torque \( T_{w} \), external torque \( T_{\text{ext}} \), torque \( T_{\text{sec}} \) from the CVT belt on the secondary pulley and torque \( T_{c} \) on the carrier in the planetary set. The external torque consists of the constant rolling resistance \( T_{\text{roll}} \), the aerodynamical drag \( c_{d} \omega_{w}^{2} \) (with constant \( c_{d} \)) and the disturbance \( T_{\text{dist}} \) (due to road slopes, wind gusts, etc.).

---

**Fig. 1** The flywheel assisted driveline

**Fig. 2** Torque scheme of the driveline

**Fig. 3** The engine map for a 1.6 l displacement Otto engine
\[
T_{\text{ext}} = T_{\text{roll}} + c_d \omega_r^2 + T_{\text{dist}}
\]

The equation of motion for the flywheel unit (flywheel plus sun gear) is given by
\[
J_f \frac{d\omega_f}{dt} = T_f
\]
with moment of inertia \(J_f\) and torque \(T_f\) from the carrier on the sun gear. Power losses in the planetary set are neglected, resulting in the torque relations \(T_u = -zT_f\) and \(T_c = -(1+z)T_f\). The power loss in the CVT is taken into account with a simple model, given by [14]
\[
T_{\text{sec}} = \frac{\eta}{\eta_{\text{ext}}} T_{\text{pri}}
\]
with CVT efficiency \(\eta\). In the model for control design (but not in the simulation model), it is assumed that \(\eta\) is constant and equal to 0.85. Quite rigorously, the CVT is assumed to behave as a pure integrator with input \(u_{\text{ext}}\) and the transmission ratio \(r\) as the output, so
\[
\frac{dr}{dt} = u_{\text{ext}}
\]
The equations of motion (2), (5), and (7) can be combined into one equation for \(\omega_e\), resulting in
\[
(J_{eq} + J_{wq} \omega_e^2) \frac{d\omega_e}{dt} = T_e(\omega_e, \varphi_e) \eta - r T_{\text{ext}} - J_{wq} \omega_e u_{\text{ext}}
\]
where \(J_{eq}\) and \(J_{wq}\) are the total moments of inertia of the load side, respectively, of the engine side, given by
\[
J_{eq}(r) = J_u + \left(1 - \frac{r \omega_e}{\eta} \right) \alpha_e^2 J_f; \quad J_{eq}(r) = J_e + \left(1 - \frac{r}{\eta \omega_e} \right) \alpha_e^2 J_f
\]
Obviously, \(J_{eq} = 0\) if \(r = r_{\text{ci}}\), where \(r_{\text{ci}}\) is the so-called zero inertia ratio:
\[
r_{\text{ci}} = \frac{\omega_f}{\omega_e} \left(1 + \frac{J_f}{\alpha_e^2 J_f \eta} \right) = \frac{\omega_e}{\omega_f} \left(1 - \frac{J_f}{\alpha_e^2 J_f \eta} \right)
\]
Furthermore, \(r_{\text{ci}} = \infty\) whenever the flywheel is not present, that is \(J_f = 0\). Finally, expression (12) shows that the original moment of inertia \(J_f\) is more than compensated by the moment of inertia \(J_f\) of the flywheel if \(r = r_{\text{ci}}\), that \(J_f\) is partly compensated by \(J_f\) if \(r \in [r_{\text{ci}}, r_{\text{f}}]\), and that \(J_f\) is enlarged by \(J_f\) if \(r \in [r_{\text{f}}, \eta \omega_e r_{\text{f}}]\). An ad hoc optimization of the design parameters \(\alpha_e\), \(\omega_e\), and \(J_f\) of the flywheel unit results in values for \(r_{\text{ci}}\) and \(r_{\text{f}}\) such that \(r_{\text{f}} < r_{\text{ci}} < r_{\text{max}}\) with \(r_{\text{max}}\) close to \(r_{\text{f}}\) and a value for \(J_f\) of the same order of magnitude as \(J_e\), see [15]. The contribution of \(J_f\) to \(J_{eq}(r)\) can be neglected.

3 Desired Behavior of the Driveline

The desired behavior of the driveline is commanded by the driver by means of the drive pedal. The actual, normalized pedal position \(\beta(t) \in [0,1]\) has to be translated by the DMS into a desired value for a yet to be chosen driveline quantity. Here, \(\beta(t)\) is translated into a desired stationary value \(P_{e,\text{d}}\) for the engine power, using a relation of the form
\[
P_{e,\text{d}} = f(\beta) P_{\text{max}}
\]
where \(f(.)\) is a strictly increasing function with \(f(0) = 0\) and \(f(1) = 1\). The corresponding desired stationary values \(\omega_{e,\text{d}}\) for the engine speed and \(\varphi_e\) for the throttle opening follow from the requirement that each stationary engine operating point has to lie on the maximum efficiency line. Hence, using Eq. (2), \(\omega_{e,\text{d}}\) and \(\varphi_e\) can be determined from
\[
\omega_{e,\text{d}} = \omega_{\text{med}}(P_{e,\text{d}}); \quad \varphi_e = \varphi_{\text{med}}(P_{e,\text{d}})
\]
In the given interpretation, the pedal position alone is not representative for a desired wheel speed. Suppose that the disturbance torque \(T_{\text{dist}}\) is constant, that the driver did not move the pedal for some time and that the state of the system has become stationary.

Without any restriction of generality, it may be assumed that this state is the desired state since otherwise the driver would move the drive pedal. Hence, Eq. (10) then reduces to:
\[
T_e(\omega_{e,\text{d}}, \varphi_e) \eta = r_d T_{\text{ext}}(\omega_{e,\text{d}})
\Rightarrow P_{e,\text{d}} \eta = r_d \omega_{e,\text{d}}(T_{\text{roll}} + c_d \omega_{e,\text{d}}^2 + T_{\text{dist}})
\]
where \(r_d\) is the stationary (desired) CVT ratio.

4 The Control Law for the ZI Driveline

The control design model of Section 2.1 represents a nonlinear system with two state variables (the engine speed \(\omega_e\) and the transmission ratio \(r\)) and two inputs (the rate of ratio change \(u_{\text{ext}}\) and the engine throttle opening \(\varphi\)). The state equations are given by Eqs. (9) and (10). With the engine speed as the output variable of interest, the relative degree is equal to 1.

Following the feedback linearization method [16,17], a new input \(v\) is introduced, such that
\[
J_{wq} r \omega_e u_{\text{ext}} = T_e(\omega_e, \varphi) \eta - r T_{\text{ext}} - (J_{eq} \eta + J_{wq} \omega_e^2)
\]
whereas the internal dynamics are described by
\[
J_{wq} r \omega_e \frac{dr}{dt} = T_e \eta - r T_{\text{ext}} - (J_{eq} \eta + J_{wq} \omega_e^2) v
\]
The proposed control law for \(v\) then is given by
\[
v = k_h (\omega_{e,\text{d}} - \omega_e); \quad k_h > 0
\]
where \(\omega_{e,\text{d}}\) is the desired engine speed as determined by the DMS. Combined with Eq. (16) this law results in one relation for the two inputs \(u_{\text{ext}}\) and \(\varphi\). The second relation follows from the requirement that the engine has to operate in points on the maximum efficiency line, meaning that the engine input \(\varphi\) has to be determined from \(\varphi = \varphi_{\text{med}}(\omega_e)\) and thus \(T_e = T_{\text{med}}(\omega_e)\) in Eq. (16).

The equilibrium point \(\omega_e = \omega_{e,\text{d}}\) of Eq. (17) with control law (19) is globally asymptotically stable. To prove stability of the controlled system it is, therefore, sufficient to show that the zero dynamics are stable.

Substitution of \(\omega_e = \omega_{e,\text{d}}\) and \(T_e = T_{\text{med}}(\omega_{e,\text{d}})\) in Eq. (18) yields the zero dynamics:
\[
J_{wq} r \omega_{e,\text{d}} \frac{dr}{dt} = T_e(\omega_{e,\text{d}}) \eta - r(T_{\text{roll}} + T_{\text{dist}} + c_d \omega_{e,\text{d}}^2)\omega_{e,\text{d}}^2
\]
With Eq. (15) the zero dynamics can also be written as
\[
\frac{dr}{dt} = (r_d - r) \frac{T_{\text{roll}} + T_{\text{dist}} + c_d (r_d^2 + r_d) \omega_{e,\text{d}}^2}{J_{wq} \omega_{e,\text{d}}}
\]
The fraction in the right hand side can be negative only for any \(r \in [r_{\text{min}}, r_{\text{max}}]\) if \(T_{\text{dist}}\) is negative large, e.g., in driving downhill on a very steep road. In such a situation commands for a transient to a higher engine power level will generally not be given by the driver. Hence, the mentioned fraction will be positive in all practically relevant transients to a higher engine power, so it may be concluded that the equilibrium point \(r = r_d\) of Eq. (21) is globally asymptotically stable and that the proposed control laws result in a stable behavior of the controlled system.
5 Results

To illustrate the impact of the flywheel unit on the driveability, simulations and experiments are performed for a conventional, CVT-based driveline and for the ZI driveline, i.e., the conventional driveline plus the flywheel unit. Detailed information on the advanced simulation model is given in [14] for the CVT and in [10] for the other components of the driveline. The moment of inertia $J_f = 0.4 \text{ kgm}^2$ of the flywheel is of the same order of magnitude as the moment of inertia $J_f = 0.22 \text{ kgm}^2$ of the engine side of the driveline. For simplicity it is assumed that the desired engine power is proportional to the pedal position, so Eq. (13) reduces to $P_e, d = P_{\text{max}} \beta$. In the simulations, a rather extreme situation is considered for the excursions of $\beta$ (see Fig. 4).

In the experiments a somewhat similar case is considered. There the driver is pushing the drive pedal once in and out for both the CVT and the ZI driveline. The results will be shown in Section 5.2.

5.1 Simulation Results. Simulations are performed with the control laws of Section 4 where $k_h = 5$ is applied. The path of the engine operating point for the considered driving cycle is given in Fig. 5 by the dashed line. The wide open throttle line and the maximum efficiency line are also depicted. Point A in this map corresponds to pedal position A in Fig. 4. The operating points move along the maximum efficiency line to point B (the maximum power point) after the pedal kick down at $t = 2 \text{ s}$. After the pedal release at $t = 7 \text{ s}$ the operating points move to point C (corresponding to pedal position C) along the previously introduced reference trajectory. Due to the CVT overdrive ratio limitation $r_{\text{max}}$, point C is not located on the maximum efficiency line. From point C the operating points move up to this line and then along this line to point D after the kick-down at $t = 15 \text{ s}$. Finally, after the pedal release at $t = 15 \text{ s}$, the operating points move along the reference trajectory to point E, corresponding to pedal position E.

Figure 6 shows the results for the power at wheels. The thin, solid line in Fig. 6a gives the desired power whereas the thick, solid line and the dashed line represent the realized power at wheels for the ZI driveline and the conventional CVT driveline, respectively.

It is noted that at $t = 7 \text{ s}$ (when the CVT is shifted up very fast) the ZI and the conventional driveline both show an inverse response although it is much less pronounced for the ZI driveline than for the conventional one.

The reason for this behavior of the ZI driveline is that $r < r_{\text{zi}}$ at $t = 7 \text{ s}$, and therefore, the engine inertia can not completely be compensated by the flywheel. At $t = 15 \text{ s}$ the CVT is shifted up very fast again. Then, however, $r > r_{\text{zi}}$ as can be seen from Fig. 6b.

As a result, the ZI driveline now does not show an inverse
behavior whereas the conventional driveline does. The ZI driveline, however shows a rather severe undershoot after the negative steps at $t = 7$ s and $t = 25$ s.

From Fig. 6a it follows that for $t = 18$ s to $t = 20$ s the error between the desired and the realized power is somewhat larger for the ZI driveline than for the conventional one. The reason is that in this time interval, part of the available engine power is used to accelerate the flywheel along with the vehicle. This energy flow to the flywheel also is the cause of the undershoot in the time intervals from $t = 11$ s to $t = 13$ s and from $t = 26$ s to $t = 33$ s.

Figures 7a and 7b present the vehicle acceleration for both drivelines after the first and the second kick-down. After the first kick-down, the acceleration of the conventional driveline shows both latency and a small inverse behavior whereas after the second kick-down, it shows latency only. The latency is caused by the fact that the vehicle is accelerated only after the engine is speeded up. The acceleration of the vehicle with the ZI driveline does not show an inverse behavior nor any significant latency.

Figure 8 shows the engine speed, flywheel speed and CVT secondary pulley speed. Apart from a constant factor, the vehicle speed is equal to the speed of the CVT secondary pulley. From this picture, it is seen that the engine speed and vehicle speed increase, whereas the flywheel speed decreases when the CVT is shifted down as is the case in the period from $t = 2$ s to $t = 7$ s. The corresponding decrease of the kinetic energy of the flywheel is mainly used for a fast acceleration of the engine and partly to accelerate the vehicle. During a CVT shift up (for instance in the period from $t = 7$ s to $t = 15$ s), the flywheel absorbs energy both from the engine and from the vehicle.

5.2 Experimental Results. The ZI driveline is implemented in a mid-sized passenger car, see Fig. 9a. This vehicle underwent moderate modifications in order to install the ZI driveline.
The control system comprises a dSPACE autobox system and an electronics cabinet. These systems are put in the trunk of the vehicle, see Fig. 9b.

The CVT ratio control strategy implemented in the control system is different from that presented by Eq. (16). Without further explanation—see therefore [10]—the control strategy implemented in the vehicle for desired power $P_{e,d}>0$ is given by:

$$\frac{d\omega_{e,d}}{dt} = \frac{T_{mot} - \omega_{e,d} P_{e,d}}{\min(-0.2J_{eq})}$$  \hspace{1cm} (22)

$$\int u_{cvt} dt = \frac{\omega_{e}}{\omega_{e,d}}$$  \hspace{1cm} (23)

This control law is used both for the CVT and the ZI driveline. With the vehicle a preliminary driveability studies are performed, [18] and [19].

Some experimental results from these studies are captured in Fig. 10. For both drivelines a subsequent pedal tip-in and back-out is performed. The pedal movements performed by the driver can be seen in Figs. 10a and 10b. In these same figures the vehicle acceleration is plotted without scale to stress the qualitative rather than the quantitative comparison.

In Figs. 10c and 10d, the engine speed and flywheel speed (for ZI driveline only) are plotted.

From these experiments the nonminimum phase behavior of the CVT driveline can be clearly observed. Especially for pedal back-out the acceleration of the vehicle occurs in the opposite direction than that of the pedal movements. This could also be observed in the power response of the simulation in Fig. 6(a) at $t=7$ s and $t=25$ s.

Another observation is the time-lag between pedal movements and the acceleration for both drivelines. This is particularly caused by the hydraulic and mechanical elasticity of the CVT actuators, [14]. This time-lag was not covered in the CVT simulation model.

The experiments as well as the simulations indeed show that the ZI driveline is able to circumvent the unwanted nonminimum phase behavior of a conventional CVT driveline. However, the mutual closed loop responses in the simulations and experiments of both the CVT and ZI driveline differ significantly. For the larger part this is caused by the different control strategies in the simulations and experiments. On the other hand, discrepancies between model and the actual driveline also cause such differences.

In the future the control strategy as well as the model need to be validated more profoundly by well-conditioned experiments with the test vehicle, for example using a roller bench. The assessment of the fuel economy of the test vehicle is performed on such a roller bench and is discussed extensively in [10] and [20]. At the European NEDC drive cycle, a reduction of the fuel consumption of up to 10% in comparison to a 4-gear automatic transmission was obtained using the E-line strategy.
6 Concluding Remarks
The paper presents a simple model for the design of control laws for a purely mechanical hybrid driveline. Simulations with the proposed, simple control laws, using a much more complex, realistic model of this driveline and some preliminary experiments with the ZI car showed that it is possible to operate the engine along the maximum efficiency line even in transient situations without compromising driveability.

The proposed control laws require that the disturbance torque $T_{\text{dist}}$ is available at each time $t$. In the simulations of Section 5 it is assumed that $T_{\text{dist}}$ is known a priori. In the implementation in the ZI car the disturbance torque is estimated, using Eqs. (5) and (6) with known estimates for the rolling resistance $T_{\text{roll}}$ and the drag constant $c_d$.

The wheel torque $T_w$ is measured with a strain gage based torque sensor in the drive shafts whereas the vehicle velocity and vehicle acceleration are estimated from encoder measurements at the wheels. It is not to be expected that commercially available cars will be equipped with torque sensors in the drive shafts. Therefore, an important topic of current and future research is the development of an observer for the disturbance torque, based on information that is available in current production cars, like engine torque and encoder measurements. Especially extended Kalman filters and disturbance observers like the one proposed in [21] are investigated and evaluated at the moment.

Robustness of the controlled system to system uncertainties and to external disturbances is not explicitly discussed here. It is noted that the proposed control laws are designed on the basis of a very simple model. The results of the application of these laws to the more complex, realistic driveline model do not indicate a significant influence of the differences between these models. Nevertheless, robustness will be a topic of future research.

Other topics are the investigation of alternative, more advanced control laws, e.g., sliding mode and $H_\infty$, and of alternative definitions for the reference trajectory of the engine power during transients to a lower power level.

Acknowledgments
This study is part of the EcoDrive project (1997–2001) which is a joint project of Van Doorne’s Transmissies (VDT), the Dutch Organization for Applied Scientific Research (TNO Automotive) and the Technische Universiteit Eindhoven (TU/e). The project is subsidized by the Dutch governmental program E.E.T. (Economy, Ecology and Technology).

References