Nonlinear Control applied to Gearshifting in Automated Manual Transmissions

Jonas Fredriksson and Bo Egardt
Control and Automation Laboratory
Department of Signals and Systems
Chalmers University of Technology
SE - 412 96 Göteborg, Sweden
e-mail: jf,be@s2.chalmers.se

Abstract

In this paper the problem of automating the gearshift process of a manual transmission without synchronizers is investigated. The application is very interesting from an integrated powertrain control point of view, since it includes many different control tasks and encourages the use of the engine as an actuator to the rest of the powertrain. A model-based control law for the task of gearshifting is presented. The controller is designed based on the backstepping methodology. It includes control laws for transmission torque control as well as for engine speed control. Simulations have shown good results for the gearshift controller.

1 Introduction

Today, the controllers for the powertrain are divided into several different controllers, e.g. controllers for the engine and for the transmission. Even though the components of the powertrain are closely coupled, the communication between the controller for the engine and the controller for the transmission is very limited or negligible. By introducing the concept of integrated powertrain control, which means that the engine and the transmission together with the rest of the driveline is treated as a single control object, the functionality of the vehicle can be improved. With such a controller, in combination with modern control theory, it may be possible to improve fuel economy, driveability, gearshift quality etc. Integrated powertrain control is not aimed at centralizing all controllers to one single controller unit, rather to increase and exploit the communication between the involved controllers.

The integrated powertrain control philosophy used in this paper can be described as using the engine, not only as a power generating source, but also use it actively as part of the control system, so-called active engine control. The application which forms the background for this investigation is gearshifting of an automated manual transmission. The powertrain in focus consists of a diesel engine with a variable geometry turbine (VGT), a fixed ratio gearbox without synchronizers and the rest of the components to form a complete powertrain.

2 Gearshifting

A typical gearshift sequence consists of five phases, *1-5*, as illustrated in Figure 1. There are actually six steps, but the sixth step is clutch control for take-offs. Take-offs are not considered so the sixth step can be neglected. In Figure 1 normal cruising corresponds to *0*.

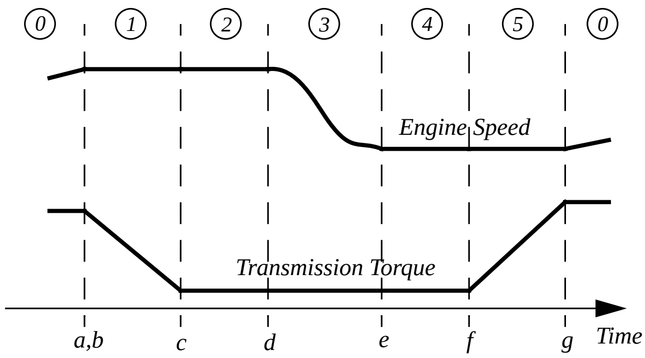

Figure 1: Gearshift sequence for an upshift. 1-5 are different states in the gearshift sequence, normal cruising is represented by 0. The events a-g represent switching conditions for the gearshift controller.

During the first phase (*1* in Figure 1) of the gearshift sequence, the transmission torque is controlled to zero in order to make the transmission torque free. The reason for making the transmission torque free is to be able to disengage the dog clutch. When the transmission is torque free, the dog clutch is disengaged and the transmission is put into neutral gear, *(2)*. The third phase is to synchronize the engine speed with the transmission speed for the new gear. This phase is the most critical in the gearshift sequence, since the power from the engine to the road driven wheels is cut off. It is therefore important to make the speed synchronization fast. During the synchronization phase, the engine speed needs to be retarded or accelerated fast, depending on upshift or downshift. Since the engine speed can not be controlled equally fast in both directions, the engine brake must be used during retardation of engine speed. When the engine speed and the new transmission speed is synchronized, the corresponding dog clutch is engaged. The power is no longer cut off. The last phase (*5*) in the gearshift sequence is to apply torque to the transmission again. This state might not be necessary, depending on the algorithms used in normal cruising. If the algorithms include a control law for reduction of driveline oscillations, as described in [1] and [8], then this state is not necessary, it only prolongs the shifting time. On the other hand, if such a control law is not included, then the fifth state is necessary in order to apply torque without causing major driveline oscillations.

From the brief description above it can easily be seen that the engine plays an essential role when performing a gearshift. Not only for controlling the engine speed during the synchronization state, but also for making the transmission torque free. Integrated powertrain control is necessary for solving this task.

However, there are some fundamental drawbacks when performing a gearshift with such a transmission.

- Loss of tractive effort
- Exhaust emissions (particularly smoke)
- Oscillation induction
- Noise and wear on powertrain components

The task for the control system is to minimize the influence of the drawbacks and to perform a smooth gearshift.

3 Powertrain Modeling

The transmission has three working states; engaged, disengaged and during engagement/disengagement.

When the transmission is engaged it is transferring power as a fixed, gear ratio. The powertrain model becomes

$$\begin{aligned} J_{fly*}\dot{x}_1 &= f(u_1, x_2) - M_f - \frac{1}{i}\left(k_s x_4 + \frac{d_s}{i}x_1 - d_s x_5\right) \\ \dot{x}_2 &= \theta_1 \frac{x_3}{g_p(x_2)} - k_2 x_1 x_2 \\ \dot{x}_3 &= \frac{1}{\tau_{tb}}(-x_3 + \theta_2 u_2) \\ \dot{x}_4 &= \frac{x_1}{i} - x_5 \\ J_{chassis}\dot{x}_5 &= k_s x_4 + \frac{d_s}{i}x_1 - d_s x_5 - M_{load} \end{aligned} \tag{1}$$

The powertrain model (1) describes a simple nonlinear engine model connected with a simple model of the driveline. The engine model is similar to the engine model presented in [5]. In [7] it was shown that a third order model is sufficient and captures the main characteristics of the driveline. The model includes two lumped masses representing the engine flywheel (including the transmission inertia) and the vehicle respectively. The masses are connected via the transmission and the driveshafts. The driveshafts are modelled with a spring and a damper, which represent the flexibility in the driveshafts and the wheels. Figure 2 visualizes the powertrain model.

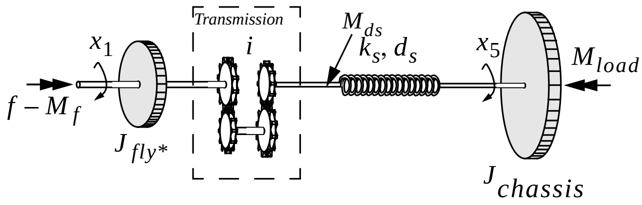

Figure 2: The powertrain model

At the left side of the powertrain shown in Figure 2 the engine torque is applied. The engine torque is approximated with a function of the fuel amount and the boost pressure, $f(u_1, x_2)$. M_f is the engine friction.

In (1) the first three states represent the engine, x_1 is the engine speed, x_2 corresponds to the boost pressure and x_3 is the compressor power. The states x_4 and x_5 represent the torsion in the driveshafts and the wheel speed respectively. The control signals u_1 and u_2 correspond to the fuel amount and the position of the inlet vanes in the turbine, respectively.

When the transmission is disengaged the powertrain works as two separate systems, see Figure 3.

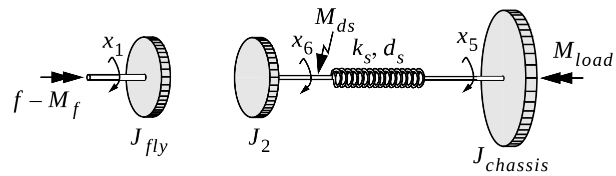

Figure 3: Powertrain when the transmission is disengaged.

To be able to handle this case, an additional inertia in the transmission needs to be introduced. The total system can be described as

$$\begin{aligned} J_{fly}\dot{x}_1 &= f(u_1, x_2) - M_f \\ \dot{x}_2 &= \theta_1 \frac{x_3}{g_p(x_2)} - k_2 x_1 x_2 \\ \dot{x}_3 &= \frac{1}{\tau_{tb}}(-x_3 + \theta_2 u_2) \end{aligned} \tag{2}$$

and

$$\begin{aligned} \dot{x}_4 &= x_6 - x_5 \\ J_{chassis}\dot{x}_5 &= k_s x_4 + d_s x_6 - d_s x_5 - M_{load} \\ J_2\dot{x}_6 &= -(k_s x_4 + d_s x_6 - d_s x_5) \end{aligned} \tag{3}$$

where (2) represents the engine, the left part in Figure 3 and (3) describes the disconnected driveline, the right part in Figure 3. The additional inertia introduced in (3) results in an additional state variable, the transmission speed. When the transmission is engaged the inertia, J_2, is incorporated in the flywheel.

If the disengagement is performed with zero torque transmitted through the dog clutch and the engagement is performed with no or small relative speed then the dog clutch can be modelled as a simple time delay, corresponding to the axial movement of one of the halves of the dog clutch.

4 Gearshift Control Design

The purpose of the gearshift controller is to minimize the effects of the drawbacks when performing a gearshift. By making the gearshift process fast, the loss of tractive effort is minimized. The second fundamental drawback is exhaust emissions or smoke. If the combustion efficiency is high the smoke level is reduced, so by using the VGT to control the amount of air to a sufficient level, the smoke emissions can be reduced. The third drawback can be limited by including algorithms for control of driveline oscillations, as shown in e.g. [8]. Noise and wear can be reduced by performing the different gearshift control tasks accurately.

There are mainly two different control objectives in the gearshift process. The two objectives correspond to transmission torque control (or driveline torque control) and speed synchronization. Normal cruising is not included as part of the actual gearshift controller.

The controller design is based on the backstepping methodology, see e.g. [6]. The reason for choosing a nonlinear method instead of linear methods is the major nonlinearities present in the diesel engine system; this is also stated in [3].

4.1 Transmission Torque Control

To be able to decouple the transmission, i.e. to disengage the dog clutch, the torque transmitted through the dog clutch should be as small as possible, preferably zero. By controlling the torque in the driveshafts to zero, the transmission will be torque free. If the transmission torque is nonzero, it means that when the dog clutch is disengaged oscillations in the driveshaft are introduced. These oscillations make it more difficult to synchronize the engine speed with the new transmission speed.

The quantity to be controlled is the driveline torque, which can be expressed as

$$M_{ds} = k_s x_4 + d_s\left(\frac{x_1}{i} - x_5\right) \tag{4}$$

Introduce z_1 as the error in driveline torque

$$z_1 = M_{ds} - M_{dsd} \tag{5}$$

Compute the derivative of the error

$$\begin{aligned} \dot{z}_1 &= \dot{M}_{ds} - \dot{M}_{dsd} = k_s\dot{x}_4 + d_s\left(\frac{\dot{x}_1}{i} - \dot{x}_5\right) \\ &= k_s\left(\frac{x_1}{i} - x_5\right) + d_s\left(\frac{\dot{x}_1}{i} - \dot{x}_5\right) \end{aligned} \tag{6}$$

During the unload state of the gearshifting process, the desired driveline torque and its derivatives are zero. By inserting (1) into (6), the differentiated error becomes

$$\dot{z}_1 = k_s\left(\frac{x_1}{i} - x_5\right) + d_s\left(\frac{f(u_1, x_2) - M_f - \frac{1}{i}M_{ds}}{J_{fly*}i} - \dot{x}_5\right) \tag{7}$$

The actual control signal u_1 appears in (7) as part of the engine torque function. A control law for u_1 can be determined such that the resulting error equation becomes

$$\dot{z}_1 = -c_1 z_1 \tag{8}$$

where c_1 is a positive design constant.

One major drawback with fast engine transients, which is necessary for the gearshift controller in order to minimize the loss of tractive effort, is exhaust emissions, and especially smoke emissions. The additional control input introduced in the turbine offers the potential to control the amount of air available for combustion and thereby the combustion efficiency and the smoke emission level. It is preferable to have an air-to-fuel ratio between 20-70 according to [4]. By keeping it sufficiently high the combustion efficiency will be high and the smoke emission level relatively low. Ideally, it is desirable to maintain a certain air-to-fuel ratio at all time, $u_1 = c_{AFR}x_2$, which means that if the boost pressure is fast, u_1 can be replaced with $c_{AFR}x_2$ in the engine torque function. Equation (9) is the control law to accomplish (8) under the assumption that the boost pressure is tightly controlled.

$$\begin{aligned} f(c_{AFR}x_2, x_2) &= \frac{J_{fly*}i}{d_s}\left(\frac{d_s}{J_{fly*}i}M_f + \frac{d_s}{J_{fly*}i^2}\left(k_s x_4\right.\right. \\ &\left.\left. + d_s\left(\frac{x_1}{i} - x_5\right)\right) + d_s\dot{x}_5 - k_s\left(\frac{x_1}{i} - x_5\right) - c_1 z_1\right) \end{aligned} \tag{9}$$

Unfortunately the boost pressure is slow compared to the fuel injection, so the air-to-fuel ratio can not be fulfilled at all time. Instead, the fuel amount will be used to compensate for the lack of air. The fuel amount can be calculated as the solution to (9) but with the true boost pressure, and the desired boost pressure is calculated by replacing x_2 with x_{2d} in (9).

Introduce z_2 as the error between the measured boost pressure and the desired boost pressure calculated from (9),

$$z_2 = x_2 - x_{2d} \tag{10}$$

and design a backstepping controller for the boost pressure.

The derivative of z_2 is computed as

$$\begin{aligned} \dot{z}_2 &= \dot{x}_2 - \dot{x}_{2d} \\ &= \theta_1 \frac{x_3}{g_p(x_2)} - k_2 x_1 x_2 - \dot{x}_{2d} \end{aligned} \tag{11}$$

Since the actual control signal u_2 does not appear in (11), x_3 is chosen as virtual control signal, for which the following stabilizing function is chosen

$$\alpha_1(\boldsymbol{x}) = \frac{g_p(x_2)}{\theta_1}(k_2 x_1 x_2 + \dot{x}_{2d} - c_2 z_2) \tag{12}$$

where $c_2 > 0$. The corresponding error variable is $z_3 = x_3 - \alpha_1(\boldsymbol{x})$, and the resulting error becomes

$$\dot{z}_2 = -c_2 z_2 + \theta_1 \frac{z_3}{g_p(x_2)} \tag{13}$$

where z_3 is the difference between the compressor power and the stabilizing function, i.e. the same as before. The derivative of z_3 with respect to time is

$$\begin{aligned} \dot{z}_3 &= \dot{x}_3 - \dot{\alpha}_1(\boldsymbol{x}) \\ &= -\frac{1}{\tau_{tb}} x_3 + \frac{\theta_2}{\tau_{tb}}(u_2 - \dot{\alpha}_1(\boldsymbol{x})) \end{aligned} \tag{14}$$

Since the control signal, u_2, appears in (14) the control law can be chosen as

$$u_2 = \frac{\tau_{tb}}{\theta_2}\left(\frac{x_3}{\tau_{tb}} + \frac{\theta_2}{\tau_{tb}}\dot{\alpha}_1(\boldsymbol{x}) - \theta_1 \frac{z_2}{g_p(x_2)} - c_3 z_3\right) \tag{15}$$

where c_3 is a positive design constant and $\dot{\alpha}_1(\boldsymbol{x})$ is given from differentiation of (12). With this choice of control law the error equation becomes

$$\dot{z}_3 = -c_3 z_3 - \theta_1 \frac{z_2}{g_p(x_2)} \tag{16}$$

Proposition 1 *Given the system (1), the controller determined by (9) and (15) makes the closed-loop system locally asymptotically stable. Furthermore, it drives the driveline torque to the desired driveline torque and maintains the steady state air-to-fuel ratio.*

Proof. The following Lyapunov function is proposed

$$V = \frac{1}{2}z_1^2 + \frac{1}{2}z_2^2 + \frac{1}{2}z_3^2 \tag{17}$$

Differentiating (17), along the solutions of the system (8), (13), and (16) yields

$$\begin{aligned} \dot{V} &= z_1\dot{z}_1 + z_2\dot{z}_2 + z_3\dot{z}_3 \\ &= -c_1 z_1^2 - c_2 z_2^2 + \theta_1 \frac{z_2 z_3}{g_p(x_2)} - c_3 z_3^2 - \theta_1 \frac{z_2 z_3}{g_p(x_2)} \\ &= -c_1 z_1^2 - c_2 z_2^2 - c_3 z_3^2 \le 0 \end{aligned} \tag{18}$$

the derivative of the Lyapunov function becomes negative semidefinite. ❑

Unfortunately, the model is not valid globally, so the closed-loop system is not globally asymptotically stable. The region of attraction is considerable, being determined essentially by the "largest" level curve of V contained in the region where the model is valid. This results in conditions for the controller, the desired engine speed should be larger than zero, $x_{1d} > 0$, and that the ratio between the injected fuel and the air-to-fuel ratio constant should be larger than the surrounding pressure, $u_1 / c_{AFR} > p_{surr}$.

4.2 Speed Synchronization

The second important part in the gearshifting sequence is the synchronization of the engine speed with the new transmission speed. This control task is the most time critical part in the gearshift process, because the power transfer through the transmission is cut off. The problem is a special case of the previously described problem. During this part, only the engine is considered, (2).

The quantity to be controlled is the error between the engine speed and the desired engine speed,

$$z_4 = x_1 - x_{1d} \tag{19}$$

Compute the derivative of the error

$$\begin{aligned} \dot{z}_4 &= \dot{x}_1 - \dot{x}_{1d} \\ &= \frac{1}{J_{fly}}(f(u_1, x_2) - M_f) - \dot{x}_{1d} \end{aligned} \tag{20}$$

Notice the similarity to the first step in the backstepping design for the previous case. By choosing the control signal, u_1, as the solution to

$$f(u_1, x_2) = -J_{fly} c_4 z_4 + M_f + J_{fly}\dot{x}_{1d} \tag{21}$$

and calculate the desired boost pressure as

$$f(c_{AFR} x_{2d}, x_{2d}) = -J_{fly} c_4 z_4 + M_f + J_{fly}\dot{x}_{1d} \tag{22}$$

the resulting error equation becomes

$$\dot{z}_4 = -c_4 z_4 \tag{23}$$

where c_4 is a positive design constant. The rest of the controller design becomes equal to the controller design for the previous case. The final controller can be summarized as

Proposition 2. *Given the system (2), the controller determined by (21) and (15) makes the closed-loop system locally asymptotically stable. Furthermore, it drives the engine speed to the desired engine speed (the new transmission speed) and maintains the steady state air-to-fuel ratio.*

The proof is easily derived in the same manner as for Proposition 1, resulting in a locally asymptotically stable closed-loop system.

5 Simulations

The control laws presented in Proposition 1 and Proposition 2 have been implemented in Dymola for verification and evaluation. The normal cruising control laws include algorithms for control of driveline oscillations, so the fifth state in the gearshift sequence can be neglected.

The design parameters for the different controllers are not chosen to minimize the gearshift time, but chosen such that it reveals the characteristical behavior of the controllers.

To be able to perform a gearshift sequence, several switching conditions must be defined. During the simulation the following switching conditions are used. A downshift (event *a*, see Figure 1) occurs at an engine speed lower than 110 rad/s and an upshift (event *b*) at 190 rad/s. Event *c* occurs when the transmission torque is zero. As described earlier, the disengagement and the engagement of the dog clutches are modelled as a time delay, so the switching condition (event *d* and event *f*) occurs when the time elapsed since entering the state is larger than 0.1 s. Event *e* occurs when the speed difference is less than 0.1 rad/s. The last event *g* is not used.

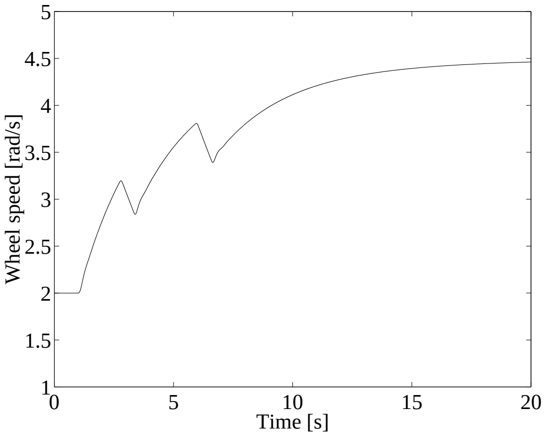

Figure 4: Wheel speed as function of time.

Figure 4 shows a drive cycle corresponding to a change in wheel speed from 2 rad/s to 4.5 rad/s. During this drive cycle, two gearshifts take place. It shows a smooth acceleration to the new wheel speed. It clearly reveals the two gearshifts resulting in a loss of tractive effort. The slow acceleration between the gearshift events is a result from the normal cruising control algorithms, since the gearshifting is in focus in this paper.

Figure 5 presents the engine speed as function of time during the drive cycle. Simulations clearly reveal the active use of the engine.

By zooming in on the engine speed and the engine torque during the first gearshift, Figure 6, it is possible to see more clearly the use of the engine. When the engine speed reaches 190 rad/s the gearshifting begins, and this happens at approximately 2.7 s. At this time the control law is switched to the driveline torque controller in Proposition 1. From 2.7-2.9 s the driveline torque is controlled to zero. At approximately 2.9 s and during 0.1 s the dog clutch is disengaged and the driveline becomes decoupled, as visualized in Figure 3.

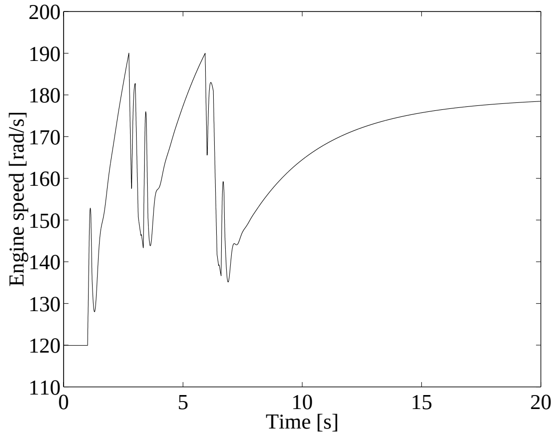

Figure 5: Engine speed as function of time.

When the disengagement is finished, the engine speed is synchronized with the new transmission speed by the control law in Proposition 2. When the synchronization is finished, the dog clutch is engaged and at approximately 3.3 s the gearshift is over and it goes back to normal cruising again. The last peak in engine speed is due to the normal cruising control law, and its characteristic behavior can be found in many papers, e.g. [2, 7, 8].

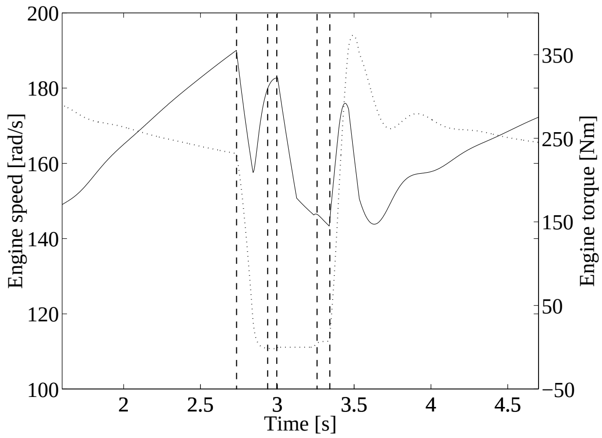

Figure 6: Engine speed (solid line) and driveline torque (dotted line) as function of time.

During the gearshift it is preferable to maintain the air-to-fuel ratio in order to avoid major exhaust emissions, especially smoke emissions, as described in the previous section. Figure 7 shows the boost pressure. The air-to-fuel ratio during the simulation is between 40-50. The oscillatory behavior of the boost pressure is due to the 'aggressive' control law for the boost pressure. The desired boost pressure, x_{2d}, is calculated as u_1/c_{AFR}, which means that the boost pressure controller tries to follow the fuel amount. One way to avoid this can be to filter the desired boost pressure in order to get a smoother desired boost pressure.

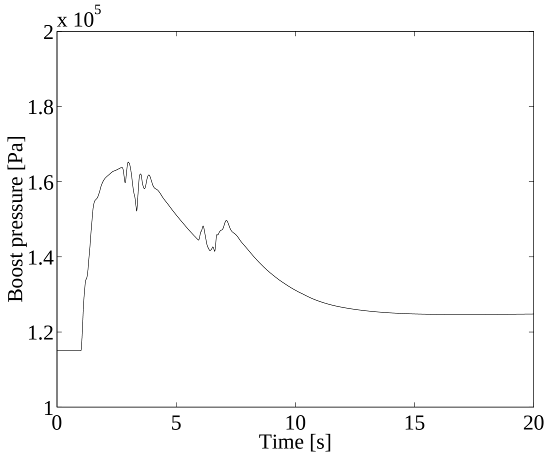

Figure 7: Boost pressure as function of time

The corresponding control signals, u_1 and u_2, are shown in Figure 8. The fuel amount, u_1, is between 0-220 mg/stroke and the VGT position, u_2, is between 0-100%. The limits correspond to the physical limits of the engine. The huge variation in fuel amount is necessary in order to perform fast gearshifts and minimize the effects of the drawbacks.

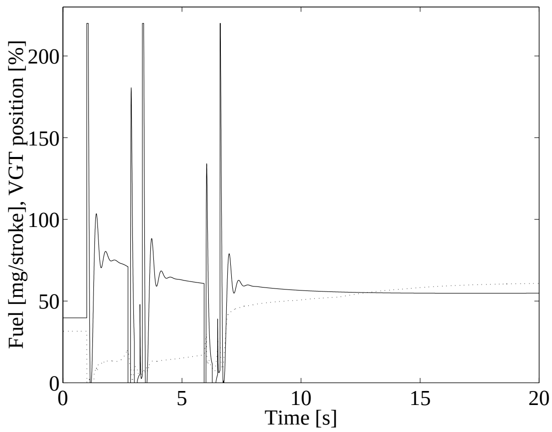

Figure 8: Control signals. The solid line correspond to u_1 and the dotted line correspond to u_2.

In the above simulations, all variables were assumed to be known or measurable. Unfortunately, this is not the case in a real vehicle. All state variables except for x_4 are measurable or computable, x_4 can be estimated with e.g. a Kalman filter as shown in [8].

The control law (9) includes the derivative of the wheel speed and this can for instance be accomplished by a sliding mode observer, see [9]. All constants can be assumed to be known from material data or identification experiments. The parameters θ_1 and θ_2 varies substantially with operating point and a further investigation is necessary to study how to handle these parameter variations.

6 Conclusions

In this paper a model-based control law for the task of gearshifting is presented. The controller is designed based on the backstepping methodology. It includes control laws for transmission torque control as well as control laws for engine speed control.

Simulations have shown good results for the gearshifting of an automated manual transmission. The shifting is smooth and does not introduce any major oscillations in the driveline. The total time for one shifting is approximately 0.6 s, which is good. The shifting is performed without any optimization of design parameters, which will improve the performance a little bit further.

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