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FLATNESS–BASED CLUTCH CONTROL FOR AUTOMATED MANUAL TRANSMISSIONS

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Abstract: Control of the clutch position is a crucial point for automating manual transmissions. Here, an electrohydraulic clutch position control system is considered. Based on the flatness approach, a nonlinear feedforward control is designed, which is combined with a linear feedback control and implemented on a standard transmission control unit. Experiments with a Mercedes–Benz Sprinter and a Mercedes–Benz CLK prove that this new control system is significantly superior to conventional control concepts, and provides an accurate trajectory tracking of the clutch position.

Keywords: Automotive Control, Nonlinear Control, Inverse Dynamics Control, Hydraulic Actuators, Position Control.

1. INTRODUCTION

An automated manual transmission (AMT) is typically built by a standard gearbox and a standard dry clutch, where both gearbox and clutch are operated by either electromechanic (Ottenbruch and Gaubitz, 1996), electropneumatic (Tanaka and Wada, 1995), or electrohydraulic actuators. The development of automated manual transmissions and their integration into the vehicle powertrain lead to a number of challenging control tasks, since the system dynamics are fast and highly nonlinear.

As an example of integrated powertrain control, backstepping control (Krstić *et al.*, 1995) of a turbocharged diesel engine during gearshifting is presented in (Fredriksson and Egardt, 2000).

Here, trajectory tracking of the clutch position using an electrohydraulic actuator is discussed. In section 2, the model of the clutch position system is presented. In section 3, a nonlinear feedforward control is derived using the flatness (Fliess *et al.*, 1995) of the system. In section 4, experimental results with a Mercedes–Benz Sprinter and a Mercedes–Benz CLK are presented.

2. MODEL OF THE HYDRAULIC CLUTCH POSITIONING SYSTEM

The model of the hydraulic clutch positioning system consists of three sets of differential equations: The first set describes the motion of the valve piston, the second set the pressure inside the hydraulic cylinder that actuates the clutch, and the third set the motion of the cylinder piston.

Let x_v and v_v denote position and velocity of the valve piston. The motion of the valve piston is described by

$$\dot{x}_v = v_v \quad , \tag{1}$$
$$\dot{v}_v = \frac{1}{m_v}\left(F_{magnet}\left(I, x_v\right) + F^v_{spring}\left(x_v\right) + F_{flow}\left(\Delta p, Q\left(\Delta p, x_v\right), x_v\right) + F^v_{friction}\left(v_v\right)\right) \quad , \tag{2}$$

where F_{magnet}, F^v_{spring}, F_{flow}, and $F^v_{friction}$ denote the forces caused by the solenoid current, valve spring, stationary oil flow, and friction, respectively. Further notations are: mass of the valve piston m_v, solenoid current I, and oil flow Q. Δp of the three–port valve is given by

$$\Delta p = \begin{cases} p_P - p & \text{for P–C opened} \\ p - p_T & \text{for C–T opened} \end{cases} \quad , \tag{3}$$

where p_P denotes the system pressure (port P), p_T the tank pressure (port T), and p the pressure of the port C, which is assumed to be equal to the pressure of the hydraulic servocylinder.

The differential equation for p is given by

$$\dot{p} = \frac{E\left(p\right)}{V\left(x_c\right)}\left(Q\left(\Delta p, x_v\right) - A_c\, v_c\right) \tag{4}$$

with the elasticity module E, total volume of tube and cylinder V, and area A_c of the cylinder piston.

Finally, the motion of the cylinder piston is given by

$$\dot{x}_c = v_c \quad , \tag{5}$$
$$\dot{v}_c = \frac{1}{m_c}\left(F^c_{spring}\left(x_c\right) + A_c p + F^c_{friction}\left(v_c\right)\right) \quad , \tag{6}$$

where x_c denotes the position of the cylinder piston, v_c its velocity, m_c the moving mass, and F^c_{spring} and $F^c_{friction}$ the forces caused by the clutch spring and friction, respectively.

3. DESIGN OF FLATNESS–BASED FEEDFORWARD CONTROL

For control design, a simplified model is used, neglecting the dynamics of the valve piston. Thus, the piston position is given by the algebraic equation

$$x^s_v = x^s_v\left(I, \Delta p\right) \quad , \tag{7}$$

which is derived by solving $\dot{x}_v = 0$, $\dot{v}_v = 0$. The stationary oil flow is now expressed as a function of the solenoid current I and the pressure difference Δp,

$$Q = Q_s\left(I, \Delta p\right) \quad . \tag{8}$$

Note that the stationary oil flow $Q_s\left(I, \Delta p\right)$ of the valve can be measured directly, instead of calculating x^s_v and inserting it into $Q = Q\left(\Delta p, x_v\right)$.

The remaining differential equations are

$$\dot{x_c} = v_c \quad , \tag{9}$$
$$\dot{v_c} = \frac{1}{m_c}\left(F^c_{spring}\left(x_c\right) + A_c p + F^c_{friction}\left(v_c\right)\right) \quad , \tag{10}$$
$$\dot{p} = \frac{E\left(p\right)}{V\left(x_c\right)}\left(Q_s\left(I, \Delta p\right) - A_c v_c\right) \quad . \tag{11}$$

For simplification, the index c is omitted in the sequel, which yields

$$\dot{x} = v \quad , \tag{12}$$
$$\dot{v} = \frac{1}{m}\left(F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right)\right) \quad , \tag{13}$$
$$\dot{p} = \frac{E\left(p\right)}{V\left(x\right)}\left(Q_s\left(I, \Delta p\right) - Av\right) \quad . \tag{14}$$

For derivation of the feedforward control law, it is shown that

$$y = x \tag{15}$$

is a *flat* output, i.e., the state variables x, v, and p and the system input I can be expressed as functions of the system output y and a finite number of its time derivatives (Fliess *et al.*, 1995). Differentiating equation (15) and inserting the state equations gives

$$\dot{y} = \dot{x} = v \quad , \tag{16}$$
$$\ddot{y} = \dot{v} = \frac{1}{m}\left(F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right)\right) \quad , \tag{17}$$
$$\overset{(3)}{y} = \frac{1}{m}\left(\frac{\partial F_{spring}\left(x\right)}{\partial x}\dot{x} + A\dot{p} + \frac{\partial F_{friction}\left(v\right)}{\partial v}\dot{v}\right) = \frac{1}{m}\left(\frac{\partial F_{spring}\left(x\right)}{\partial x}v + A\frac{E\left(p\right)}{V\left(x\right)}\left(Q_s\left(I, \Delta p\right) - Av\right) + \frac{\partial F_{friction}\left(v\right)}{\partial v}\frac{1}{m}\left(F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right)\right)\right) \quad . \tag{18}$$

The relative degree r of the system equals the system order n. Thus, the system is input–state linearizable (Slotine and Li, 1991) for invertible $Q_s\left(I, \Delta p\right)$, which implies that the system is flat.

Solving (15), (16), and (17), the state variables are given by

$$x = y \quad , \tag{19}$$

$$v = \dot{y} \quad , \tag{20}$$

$$p = \frac{1}{A}\left(m\ddot{y} - F_{spring}\left(y\right) - F_{friction}\left(\dot{y}\right)\right) \quad . \tag{21}$$

Equation (18) yields

$$\begin{aligned} &Q_s\left(I, \Delta p\right) = \\ &\quad A\,v + \frac{1}{A}\frac{V\left(x\right)}{E\left(p\right)} \\ &\quad \left(m \overset{(3)}{y} - \frac{\partial F_{spring}\left(x\right)}{\partial x} v \right. \\ &\quad - \frac{\partial F_{friction}\left(v\right)}{\partial v}\frac{1}{m} \\ &\quad \left. \left(F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right)\right)\right) \quad . \end{aligned} \tag{22}$$

Since $Q = Q_s\left(I, \Delta p\right)$ is continuous and monotonically increasing with respect to I, i.e. $\partial Q_s\left(I, \Delta p\right)/\partial I \geq 0$, an appropriate 'inverse' function $I = Q_s^{-1}\left(Q, \Delta p\right)$ can be defined. Thus, equation (22) yields

$$I = Q_s^{-1}\left(Q_f, \Delta p\right) \tag{23}$$

with

$$\begin{aligned} &Q_f = A\,v + \frac{1}{A}\frac{V\left(x\right)}{E\left(p\right)} \\ &\quad \left(m \overset{(3)}{y} - \frac{\partial F_{spring}\left(x\right)}{\partial x} v \right. \\ &\quad - \frac{\partial F_{friction}\left(v\right)}{\partial v}\frac{1}{m} \\ &\quad \left. \left(F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right)\right)\right) \quad . \end{aligned} \tag{24}$$

Inserting the state variables according to (19), (20), and (21), the system input I can be expressed as a function of the system output y and its time derivatives $\dot{y}$, $\ddot{y}$, and $\overset{(3)}{y}$,

$$I = I\left(y, \dot{y}, \ddot{y}, \overset{(3)}{y}\right) \quad , \tag{25}$$

according to

$$I = Q_s^{-1}\left(Q_f, \Delta p\right) \tag{26}$$

with

$$\begin{aligned} &Q_f = \\ &A\,\dot{y} + \frac{1}{A}\frac{V\left(y\right)}{E\left(\frac{1}{A}\left(m\ddot{y} - F_{spring}\left(y\right) - F_{friction}\left(\dot{y}\right)\right)\right)} \\ &\quad \left(m \overset{(3)}{y} - \frac{\partial F_{spring}\left(y\right)}{\partial y}\dot{y} - \frac{\partial F_{friction}\left(\dot{y}\right)}{\partial \dot{y}}\ddot{y}\right) \end{aligned} \tag{27}$$

and

$$\Delta p = \begin{cases} p_P - \frac{1}{A}\left(m\ddot{y} - F_{spring}\left(y\right) - F_{friction}\left(\dot{y}\right)\right) \\ \qquad \text{for } Q_f \geq 0 \\ \frac{1}{A}\left(m\ddot{y} - F_{spring}\left(y\right) - F_{friction}\left(\dot{y}\right)\right) - p_T \\ \qquad \text{for } Q_f < 0 \end{cases} \tag{28}$$

Inserting the desired system output y_d and its time derivatives yields the nonlinear feedforward control

$$I_d = I\left(y_d, \dot{y}_d, \ddot{y}_d, \overset{(3)}{y}_d\right) \quad , \tag{29}$$

conveniently written as

$$I_d = Q_s^{-1}\left(Q_d, \Delta p_d\right) \tag{30}$$

with the desired oil flow

$$\begin{aligned} &Q_d = A\,\dot{y}_d + \frac{1}{A}\frac{V\left(y_d\right)}{E\left(p_d\right)} \\ &\left(m \overset{(3)}{y}_d - \frac{\partial F_{spring}\left(y_d\right)}{\partial y_d}\dot{y}_d - \frac{\partial F_{friction}\left(\dot{y}_d\right)}{\partial \dot{y}_d}\ddot{y}_d\right) \quad , \end{aligned} \tag{31}$$

the desired pressure difference

$$\Delta p_d = \begin{cases} p_P - p_d & \text{for } Q_d \geq 0 \\ p_d - p_T & \text{for } Q_d < 0 \end{cases} \quad , \tag{32}$$

and the desired cylinder pressure

$$p_d = \frac{1}{A}\left(m\ddot{y}_d - F_{spring}\left(y_d\right) - F_{friction}\left(\dot{y}_d\right)\right) \; . \tag{33}$$

By neglecting higher derivatives, e.g. $\overset{(3)}{y}_d$ in equation (31), the feedforward control law can be simplified. Using the assumption that the dynamics of v is much faster than the dynamics of the other state variables, the differential equation (13) can be replaced by the algebraic equation

$$F_{spring}\left(x\right) + Ap + F_{friction}\left(v\right) = 0 \quad . \tag{34}$$

This yields the simplified feedforward control law

$$I_d = Q_s^{-1}\left(Q_d, \Delta p_d\right) \tag{35}$$

with the desired oil flow

$$\begin{aligned} &Q_d = A\,\dot{y}_d + \frac{1}{A}\frac{V\left(y_d\right)}{E\left(p_d\right)} \\ &\quad \left(-\frac{\partial F_{spring}\left(y_d\right)}{\partial y_d}\dot{y}_d - \frac{\partial F_{friction}\left(\dot{y}_d\right)}{\partial \dot{y}_d}\ddot{y}_d\right) \quad , \end{aligned} \tag{36}$$

the desired pressure difference

$$\Delta p_d = \begin{cases} p_P - p_d & \text{for } Q_d \geq 0 \\ p_d - p_T & \text{for } Q_d < 0 \end{cases} \quad , \tag{37}$$

and the desired cylinder pressure

$$p_d = \frac{1}{A}\left(-F_{spring}\left(y_d\right) - F_{friction}\left(\dot{y}_d\right)\right) \quad . \tag{38}$$

4. EXPERIMENTAL RESULTS

The flatness–based feedforward control, combined with a linear feedback control, was implemented on a standard Siemens transmission control unit, equipped with a 16 bit 80C167 processor. The control algorithm was calculated using pure integer arithmetic with a sample time of 4 msec. The experiments were conducted with a Mercedes–Benz Sprinter, see figure 1, and a Mercedes–Benz CLK, see figure 4.

4.1 *Mercedes–Benz Sprinter*

Starting off the vehicle is the most crucial situation for clutch position control. Therefore, experimental results for starting off using a conventional control approach and the flatness–based approach are presented in figure 2 and figure 3, respectively. Figure 2 shows the desired and the actual clutch position in artificial units as well as engine speed and gearbox input speed using a conventional controller. The touch point of the clutch is about 400 artificial units, and at 1000 artificial units, the clutch is fully engaged. Thus, the most relevant part of the trajectory is between 400 and 800 artificial units. The desired clutch position is calculated by the so called AMT manager, the controller of an cascaded outer control loop, taking into account the position of the accelerator pedal as well as engine speed and vehicle velocity. The measurement of the actual position is disturbed by high–frequent oscillations caused by the diesel engine. Furthermore, there is a large tracking error of the clutch position. This error in the inner control loop causes oscillations in the engine speed and thus in the desired clutch position, the controller output of the outer loop, and results in a poor starting off performance of the vehicle.

Figure 3 shows the desired and the actual clutch position in artificial units as well as engine speed and gearbox input speed using the flatness–based controller. Again, the measurement of the actual position is disturbed by high–frequent oscillations caused by the diesel engine, but the tracking error has been reduced significantly. Therefore, the engine speed and the desired clutch position are much smoother, and the starting off performance of the vehicle is improved significantly.

4.2 *Mercedes–Benz CLK*

Due to these results, only the flatness–based control had been implemented for the Mercedes–Benz CLK. Figure 5 shows the desired and the actual clutch position in artificial units as well as engine speed and gearbox input speed for starting off and consecutive 1–2, 2–3, and 3–4 gear shifts. Note that the controller is switched off about three seconds after the clutch has been fully engaged, and switched on again when a gear shift is initiated. Switching off the controller results in a steady–state error at $t = 135$ sec. When the controller is switched on, the trajectory tracking of the clutch position is almost perfect.

5. SUMMARY AND CONCLUSIONS

Automated manual transmissions embrace the gearbox, the clutch, and the actuators operating gearbox and clutch. Clutch position control is a crucial task, especially for starting off the vehicle. Here, a electrohydraulic clutch position control system was considered. Based on the flatness approach, a nonlinear feedforward control was derived. This nonlinear feedforward control, combined with a linear feedback control, was implemented on a standard transmission control unit, and experiments with a Mercedes–Benz Sprinter and a Mercedes–Benz CLK were conducted. The system performance using the flatness–based control was significantly improved compared to a conventional control approach, and provides an accurate trajectory tracking of the clutch position. This result is even more impressive considering the limited computing resources of the standard transmission control unit, especially the low sampling rate and the integer arithmetic.

6. REFERENCES

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Fig. 1. Mercedes–Benz Sprinter

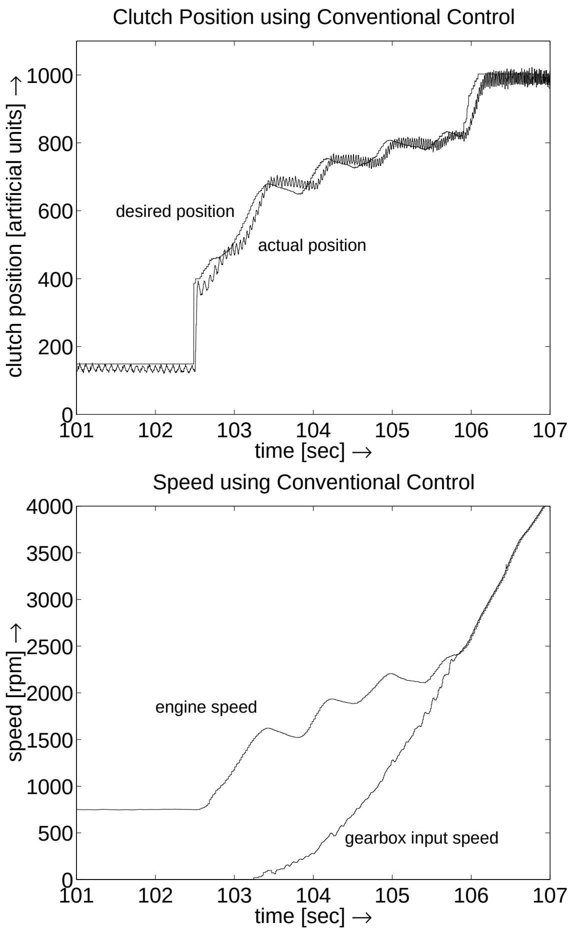

Fig. 2. Starting off using a conventional controller

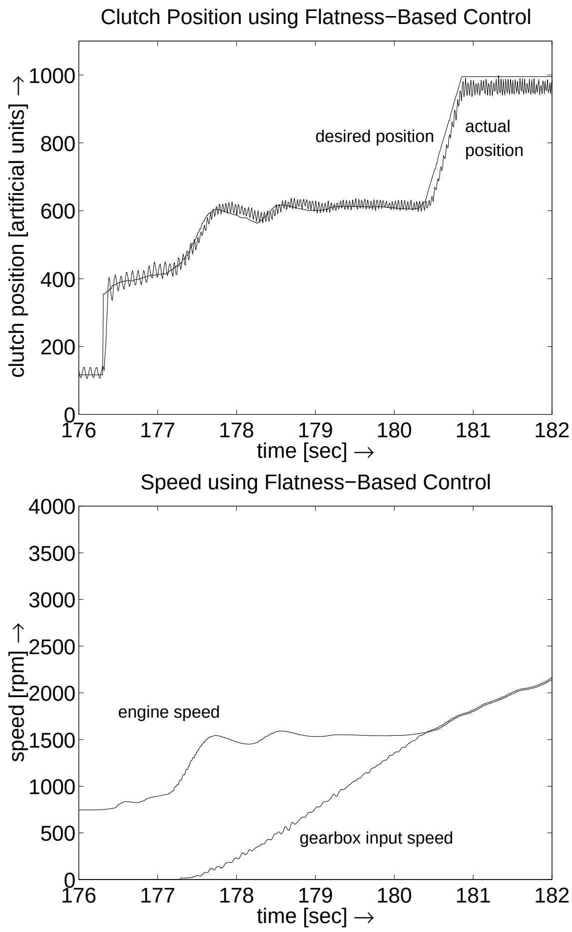

Fig. 3. Starting off using the flatness–based controller

Fig. 4. Mercedes–Benz CLK

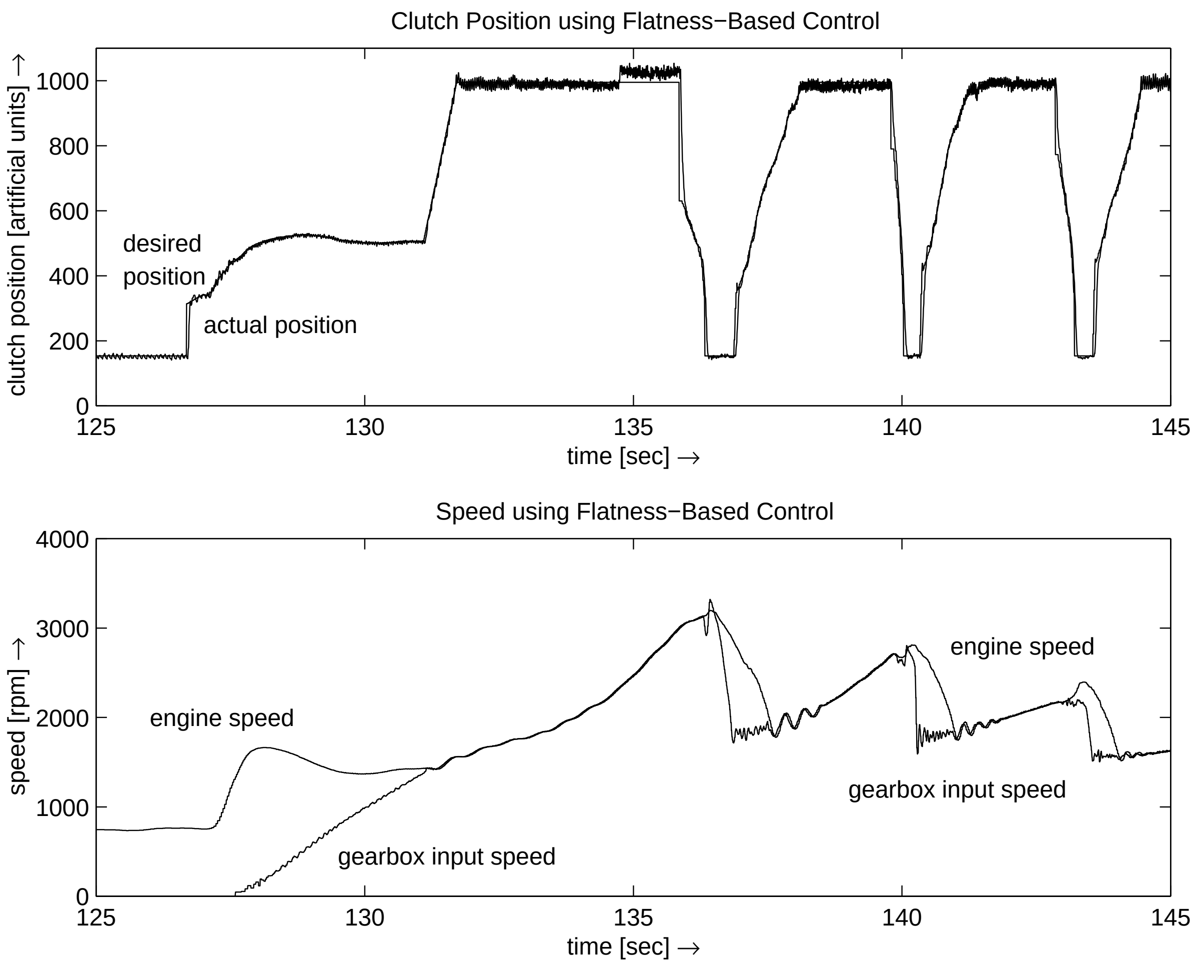

Fig. 5. Starting off and 1–2, 2–3, and 3–4 gear shifts using the flatness–based controller