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Engagement Control for Automotive Dry Clutch

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Abstract. Control of the dry clutch engagement process for automotive application is considered. A linear quadratic state feedback controller is obtained by solving a finite time horizon optimal control problem. The engine torque and the load torque are assumed as known disturbances. The control signal consists of a feedforward and a feedback component and it guarantees fast engagement, minimum slipping losses and comfortable lock-up. The critical standing start operating conditions are considered. Numerical results show the good performance of the closed loop system.

1. Introduction

The engagement of dry clutches is a very important process both to ensure small facing wear and good powertrain performance [1]-[3]. The engagement must be controlled in order to satisfy different and sometimes conflicting objectives: small friction losses, minimum time needed for the engagement, preservation of driver comfort. These goals must be reached by applying a suitable normal force to the clutch driven disk. To this aim, several control strategies have been proposed in the literature [4]-[8]. In this paper, by using a classical dynamic model of the clutch engagement system, an optimal control state feedback is proposed whose performance index weights the difference between the two speeds (so as requested in order to preserve the passengers comfort) and the energy dissipated during the engagement. Moreover, the controller parameters are chosen so that the clutch force does not exceed its maximum value. The control design problem is then formulated as a finite time optimal control problem with initial time constraints (zero initial force) and final time constraints (the two speeds must be equal at the end of the engagement process) [9]. The solution consists in the sum of two control signals: a time-varying state feedback controller, whose gain depends only on the desired duration of the engagement, and a feedforward component which compensates for the unknown initial conditions and the applied engine torque. Numerical simulations show the good performance obtained by the optimal control for different load torques.

2. Dynamic model

The dynamic model of the clutch engagement system during slipping conditions consists of the following two differential equations:

$$I_e\dot{\omega}_e = T_{in} - b_e\omega_e - T_{cl}, \tag{1}$$

$$I_v\dot{\omega}_v = T_{cl} - b_v\omega_v - T_l, \tag{2}$$

where I_e is the engine inertia, ω_e the crankshaft rotor speed, T_{in} the engine torque, b_e the crankshaft friction coefficient, T_{cl} the torque transmitted by the clutch, I_v the equivalent vehicle moment of inertia (it takes into account the presence of the clutch, the mainshaft, the powertrain and the vehicle) - which will also depend on the gear ratio - , ω_v the clutch disk rotor speed, b_v the corresponding friction coefficient and T_l the equivalent load torque. Equation (1) models the rotation of the crankshaft, whereas (2) models the rotation of the so called clutch disk. The remaining part of the powertrain transmission system is simply modeled through the equivalent vehicle inertia I_v and the load torque T_l. Though equations (1)-(2) do not model in detail the whole powertrain, this model captures the main dynamics of the system under investigation and is simple enough to design a controller through analytical procedures. The clutch torque T_{cl} can be expressed as a function of the normal force applied to the clutch disk as follows:

$$T_{cl} = kF_n sign(\omega_e - \omega_v), \tag{3}$$

where $k = 4R\mu_d/3$, R is the equivalent disk ratio and μ_d is the dynamic friction coefficient (see [10] for further details). When the clutch is engaged the dynamic model becomes

$$(I_e + I_v)\dot{\omega} = T_{in} - (b_e + b_v)\omega - T_l, \tag{4}$$

where $\omega=\omega_e=\omega_v$. The commutation from the slipping model (1)-(2) to the engaged model (4) is determined by the equality condition $\omega_e=\omega_v$ with the constraint that the clutch torque is smaller than the static friction torque, so that slipping is avoided.

The control strategy will be based on the model (1)-(2), which characterizes the engagement process, where $x_1=\omega_e$ and $x_2=\omega_e-\omega_v$ are chosen as state variables and F_n is the control variable.

Since the control variable F_n is zero at the initial time, i.e. $F_n(0)=0$, it cannot vary discontinuously and cannot exceed a maximum value F_{nMAX} , we introduce as a further state variable $x_3=F_n$ with the constraints $x_3(0)=0$, $x_3(t) \le F_{nMAX}$ for all t, and use as input to the system the time derivative u of the force rather than the force itself. Thus, the dynamic model (1)-(2) can be rewritten as follows:

$$\dot{x}_1 = -\frac{b_e}{I_e}x_1 - \frac{k}{I_e}x_3 + \frac{T_{in}}{I_e}, \tag{6}$$

$$\dot{x}_2 = \left(-\frac{b_e}{I_e} + \frac{b_v}{I_v}\right)x_1 - \frac{b_v}{I_v}x_2 - \left(\frac{k}{I_e} + \frac{k}{I_v}\right)x_3 + \frac{T_{in}}{I_e} + \frac{T_l}{I_v}, \tag{7}$$

$$\dot{x}_3 = u \quad , \tag{8}$$

with the constraint $u \in [u_{MIN}, u_{MAX}]$.

Direct computation of the controllability matrix of the system (6)-(8) shows that the system is completely controllable. The input torque and the load torque will be considered as known disturbances.

3. Control strategy

The Maximum Pontryagin Principle can be applied to the system (6)-(8) in order to design a controller that minimizes the duration of the engagement process. This allows one to conclude that, along the optimal trajectory, the control variable can assume only two different values: the maximum value u_{MAX}, corresponding to the maximum time derivative of the normal force, and the minimum value u_{MIN} which is zero since it is assumed that the force does not decrease. Though the minimum time controller does not allow to control losses and comfort related to the engagement process, this controller is useful to determine lower bounds for the engagement duration, depending on the maximum allowable derivative of the control force, a typical technological constraint.

To consider both losses minimization and comfort, a state feedback controller can be used. The controller is obtained by applying the Linear Quadratic control approach to the system under investigation. The objective function to be minimized is the following:

$$J = \int_0^{t^*} (q x_2^2(t) + r u^2(t)) dt , \qquad (9)$$

where t^* is an *a priori* chosen time instant, and q and r are weight constants; the engagement must be forced through the final time constraint $x_2(t^*)=0$.

The control coming out by the application of this technique has the following expression (see [10] for details):

$$u(t) = k(t)^T x(t) + \rho(t) , \qquad (10)$$

where $k(t)$ is a vector function obtained by the integration of a differential Riccati equation and $\rho(t)$ is a scalar function depending both on the initial conditions and on the values of the input and load torques. In real applications it is obviously unrealistic to assume the availability of on-line computational power sufficient to solve the matrix differential Riccati equation. To overcome this problem it is possible to compute off-line the feedback gains and to find the coefficients of the polynomials which approximate their time evolution. This coefficients can than be stored in a look-up-table whose entries are the input torque, the load torque and the initial conditions.

It is important to stress that u is the time derivative of the force F_n. In order to obtain the force to be applied to the clutch disk an integration is still needed. Coherently with this, the value of the control u is zero when the lock-up is reached, i.e. $u(t^*)=0$, so that the force is kept constant to its maximum value for $t>t^*$.

4. Simulations

The vehicle moment of inertia has been obtained by imposing that the linear and rotating kinetic energies are equal, which leads to $I_v=mR_r^2/(\tau_g^2\eta)$ where m is the vehicle mass, R_r the tires ratio, τ_g the gear ratio and η the efficiency. The system parameters are: I_e =0.2 kgm^2, m=1600kg, b_e=b_v=0.03 Nms, η=0.88, μ_d=0.4, F_{nMAX}=5000N, k=0.098 m, R_r=0.318 m. Moreover, different values of the input torque and load torque have been considered.

The engagement process during standing starts has been considered. Fig. 1 shows the crankshaft and vehicle rotor speeds under the LQ feedback control. The two groups of plots are obtained by applying constant input torque steps of 100 Nm and 150 Nm at t=0. Within each group, the different behaviors are determined by different values of the load torque (from 5 Nm to 30 Nm approximately).

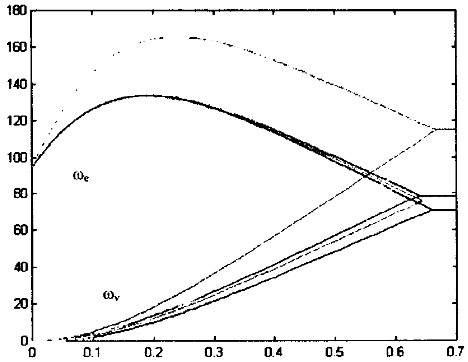

Fig. 1. Crankshaft and vehicle rotor speeds (measured in rad/sec) as a function of time (measured in sec) during the LQ controlled engagement process at standing starts for q=1000 and r=1.

References

[1] B. K. Powell, K. E. Bailey, S. R. Cikanek, "Dynamic Modeling and Control of Hybrid Electric Vehicle Powertrain Systems", *IEEE Control Systems Magazine*, October 1998, p. 17-33.

[2] C. Bader, "Powertrain Dynamics – Progress on the Use and Development of the Computer Aided Gearshift Systems", *Proc. of Transportation Electronics*, 1990, p. 375-385 (paper #901160).

[3] M. Schwab, "Electronically-Controlled Transmission Systems – Current Position and future Developments", *Proc. of Transportation Electronics*, 1990, p. 335-342 (paper #901156).

[4] A. Szadkowski, R. B. Morford, "Clutch Engagement Simulation: Engagement Without Throttle", *SAE Technical Paper Series 920766*, p. 103-117, 1992.

[5] G. Ercole, G. Mattiazzo, S. Mauro, M. Velardocchia, F. Amisano, R. Flora, "Co-operating Clutch and Engine Control for Servoactuated Shifting through Fuzzy Supervisor", *SAE Technical Paper 1999-01-0746*, p. 101-109, 1999.

[6] R. Hibino, M. Osawa, M. Yamada, K. Kono, M. Tanaka, "H^∞ Control Design for Torque-Converter-Clutch Slip System", *Proc. 35th IEEE CDC*, Kobe, Japan, 1996, p. 1797-1802.

[7] H. D. Lee, S. K. Sul, H. S. Cho, J. M. Lee, "Advanced Gear Shifting and Clutching Strategy for Parallel Hybrid Vehicle with Automated Manual Transmission", *Proc. of 33rd IEEE Industry Applications Conference*, 1998, p. 1709-1714.

[8] J. Slicker, R. N. K. Loh, "Design of Robust Vehicle Launch Control System", *IEEE Trans. on Control System Technology*, vol. 4, no. 4, 1996, p. 326-335.

[9] A. E. Bryson, Y. C. Ho, *Applied Optimal Control*, Hemisphere Publ. Corporation.

[10] L. Glielmo, F. Vasca, "Optimum Control of Dry Clutch Engagement", *SAE Technical Paper 2000-01-0837*, 2000.