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A Control Strategy on Starting up of Vehicle with Automatic Manual Transmissions (AMT)*

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Abstract: This study puts forward a control strategy for the dry clutch engagement process, considered as a finite time optimal control problem. The main idea of this strategy is to obtain a linear quadratic state feedback controller by solving the optimal control problem. The performance obtained with the proposed controller are compared with those achievable by means of a classical open loop control strategy. Moreover, different starting operating conditions are used to test the controller performance. Numerical simulations shows the good performance obtained by the proposed controller, which guarantees fast engagement, minimize the slip losses, reduce the fuel consumption and improve in the shift quality.

Key words: Starting control, AMT, slipping friction work, slipping phase

INTRODUCTION

Automated Manual Transmission (AMT) has been widely developed due to its high transfer efficiency, low cost and easiness in manufactures. But for the reason such as: the antinomy between the starting performance indices, uncertainty of driver starting intent, complexity of the calculating process of the indices and the nonlinear of the control system etc.[1].The starting control becomes a key and difficult issue. The core of starting control is clutch control. The control targets is to make the clutch engagement smooth, minimize the time needed for the engagement, reduce the friction losses, extend clutch useful life period, reduce the engine speed fluctuation, preserve the driver comfort[2] and minimize the rate of fuel consumption. To this aims, several control strategies have been proposed. Franco and Luigi[2] developed a slip control technique which is formulated as a piecewise linear time invariant model for the dry clutch based on closed loop with feedback controller. Ercole and Mattiazzo[3] proposed clutch control model based on Fuzzy control strategy. Their dynamic model consists of sixth order state space of the powertrain. Szadkowski and Morford[4], proposed a model, avoid using the throttle during the engagement in diesel engine vehicles. The engagement of the clutch control in heavy trucks is considered[5]. In this study, clutch engagement process is considered as a finite time linear quadratic control problem. The solution of this problem is based on the dynamic model of the clutch during engagement phase. An optimal state feedback control law is formulated that minimizes a cost function, whose performance index weight is the speed difference between the clutch master and slave plates. This can be achieved by solving Riccati equation. A simulink/Stateflow simulation scheme has been built to simulate the proposed controller. Different starting up conditions are used to test the controller performance. Numerical simulations show the good performance obtained by the proposed controller.

STARTING UP EVALUATING INDICATORS

Slipping friction work (L_c): Slipping friction work is the work that generated from the sliding friction between the clutch drive and driven plates, its value can reflect the degree of clutch wear. The greater value of L_c is the heavier clutch wear. L_c can be denoted by[6]:

$$L_c = \int_0^{t_1} T_c\,\omega_e\,dt + \int_{t_1}^{t_2} T_c\,(\omega_e - \omega_v)\,dt$$

where, T_c: clutch transmitted torque
ω_e: engine rotating speed
ω_v: clutch driven plate rotating speed

The degree of jerk (j): Jerk degree is the rate of change of the vehicle acceleration. It can exclude the influence of bump acceleration due to the road conditions and really

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reflect the degree of starting jerk on human body. The expression of jerk (j) is [1]:

$$j = \frac{da}{dt}$$

Engine speed fluctuation: Engine speed fluctuate and the mainly reasons are: Firstly, the flywheel sometimes cannot do a perfect job in storing up rotation energy during the power impulses of the engine and releasing this energy between the power impulses. Secondly, the torsional vibration of the crankshaft that caused by the varying forces acting on it.

CLUTCH DYNAMIC MODEL

Clutch allows engine power to be applied gradually, when a vehicle is starting out and interrupt power to avoid crunching when shifting. The clutch disc is basically a steel plate, covered with a frictional material that goes between the flywheel and the pressure plate. When the clutch is engaged, the disc is squeezed between the flywheel and the pressure plate[7]. The dynamic model can be obtained by applying the equilibrium torque condition.

Slipping model: The dynamic model of the clutch engagement system during slipping phase conditions is formulated as a second order dynamic state space model, consists of the following two differential equations:

$$I_e \dot{\omega}_e = T_{en} - b_e \omega_e - T_{CL} \tag{1}$$

$$I_v \dot{\omega}_v = T_{CL} - b_v \omega_v - T_L \tag{2}$$

where, I_e is the engine inertia, I_v the equivalent vehicle moment of inertia (takes into account the presence of the clutch, the main shaft, the power train and the vehicle), ω_e the crankshaft rotor speed, T_{en} the engine torque, b_e the crankshaft friction coefficient, T_{CL} is the torque transmitted by the clutch, ω_v is the clutch disk rotor speed, b_v the corresponding friction coefficient and T_L the load torque. As the clutch friction torque operates as a load torque for the engine, it will act as a forcing torque for the driveline subsystem. Clutch transmitted torque can be expressed in the Coulomb friction region as a function of the normal clamping force, which can be defined as:

$$T_{CL} = k F_n \,\mathrm{sign}(\omega_e - \omega_v) \tag{3}$$

$$k = 2\,\mu_d R \tag{4}$$

where, R is the equivalent disk ratio, μ_4 is the dynamic friction coefficient. F_n is the normal clamping force.

Engage model: When the clutch is engaged, the engine speed ω_4 and the clutch disk speed ω_v are equals since the elastic forces lock the clutch disk to the crankshaft ($\omega_e = \omega_v = \omega$) and the clutch torque is smaller than the static friction torque, so slipping is avoided. In order to model this situation formula (1) can added to formula (2):

$$\left(I_e + I_v\right)\dot{\omega} = T_{en} - \left(b_e + b_v\right)\omega - T_L \tag{5}$$

The control strategy is based on the Eq. 1-2, which characterizes the engagement process as follows:

The state variables are chosen such that $x_1 = \omega_e$, $x_2 = \omega_e - \omega_v$ and $x_3 = F_n$ as a control variable. The dynamic Eq. 1-2 can be rewritten in terms of those variables as follows:

$$\left.\begin{aligned} \dot{x}_1 &= -\frac{b_e}{I_e}x_1 - \frac{k_1}{I_e}x_3 + \frac{T_{en}}{I_e} \\ \dot{x}_2 &= \left(-\frac{b_e}{I_e} + \frac{b_v}{I_v}\right)x_1 - \frac{b_v}{I_v}x_2 - \left(\frac{k_1}{I_e} - \frac{k_1}{I_v}\right)x_3 + \frac{T_{en}}{I_e} + \frac{T_L}{I_v} \\ \dot{x}_3 &= u \end{aligned}\right\} \tag{6}$$

LINEAR QUADRATIC CONTROLLER (LQC) DESIGN

In order to achieve a better dynamic performance for the clutch during the slipping process, a linear quadratic feedback control technique is used. This is done by linearizing the system of Eq. 6 in terms of linearized variables and state feedback matrix is derived that minimizes a cost function by solving Riccati equation. The performance index weight of this cost function is the slip speed (the speed difference between the clutch master and slave plates).

Equation 6 can be lumped as one steady state equation. The computation of the controllability matrix for this system show that the system is completely controllable.

$$\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\end{pmatrix} = \begin{pmatrix} -\frac{b_e}{I_e} & 0 & -\frac{k_1}{I_e} \\ -\frac{b_e}{I_e} + \frac{b_v}{I_v} & -\frac{b_v}{I_v} & -\left(\frac{k_1}{I_e} + \frac{k_1}{I_v}\right) \\ 0 & 0 & 0 \end{pmatrix}\begin{pmatrix}x_1\\ x_2\\ x_3\end{pmatrix} + \begin{pmatrix}0\\0\\1\end{pmatrix}u + \begin{pmatrix}\frac{1}{I_e} & 0\\ \frac{1}{I_e} & \frac{1}{I_v}\end{pmatrix}\begin{pmatrix}T_{en}\\ T_L\end{pmatrix} \tag{7}$$

This model is linearized in the neighborhood of a stationary point (x_0, u_0) and the linear model is expressed by:

$$\Delta\dot{x} = A\Delta x + B\Delta u \tag{8}$$

where, $x = x\text{-}x_0$, $\Delta u = u\text{-}u_0$ and this linear model is the same for all stationary points. The cost function to be minimize is:

$$J = \int_0^T x_2^2 + u^2 \tag{9}$$

x_2 can be express by:

$$x_2 = M\,x \tag{10}$$

where, M = (0 1 0).

The cost function is reformulate in terms of Δx, Δu and M, which is given by :

$$J = \int_0^T \left[M\left(x_o + \Delta x\right)\right]^2 + \eta\left(u_o + \Delta u\right)^2 \tag{11}$$

Equation 11 can be put in the form:

$$J = \int_0^T \left(M\,\Delta x + r_1\right)^2 + \eta\left(\Delta u + r_2\right)^2 \tag{12}$$

where $r_1 = Mx_0$, $r_2 = u_0$ and η is a constant chosen such that a feasible control signal is used.

In order to minimize Eq. 12, a Riccati equation is solved and this can be done by augmenting (A,B) of Eq. 8 with models of the constant r_1 and r_2. Since these models will not be controllable, they must be stable in order to solve Riccati equation. To achieve the stability condition, the following models are used.

$$\left.\begin{aligned} \dot{r}_1 &= -\alpha r_1 \\ \dot{r}_2 &= -\alpha r_2 \end{aligned}\right\} \tag{13}$$

where, α is constant value.
Hence, the new augmented model is given by :

$$B_r = \left(\,B \;\; 0 \;\; 0\right)^T = (0\;\;0\;\;1\;\;0\;\;0)^T \tag{14}$$

$$x_r = \left(\Delta x^T \;\; r_1 \;\; r_2\right)^T \tag{15}$$

$$A_r = \begin{pmatrix} & & & 0 & 0 \\ & A & & 0 & 0 \\ & & & 0 & 0 \\ & & & -\alpha & 0 \\ 0 & 0 & 0 & 0 & -\alpha \end{pmatrix}$$

$$= \begin{pmatrix} -\frac{b_e}{I_e} & 0 & -\frac{k_1}{I_e} & 0 & 0 \\ -\frac{b_e}{I_e}+\frac{b_v}{I_v} & -\frac{b_v}{I_v} & -\left(\frac{k_1}{I_e}+\frac{k_1}{I_v}\right) & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -\alpha & 0 \\ 0 & 0 & 0 & 0 & -\alpha \end{pmatrix} \tag{16}$$

Using Eq. 14-16, the cost function can be rewritten in the form of :

$$\int_0^T x_r^T\,Q\,x_r + R\,\Delta u^2 + 2x_r^T\,N\,\Delta u \tag{17}$$

with

$$Q = \left(\,M\;1\;0\,\right)^T\left(\,M\;1\;0\,\right) + \eta\left(\,0\;0\;0\;0\;1\,\right)^T\left(\,0\;0\;0\;0\;1\,\right)$$

$$= \begin{pmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & \eta \end{pmatrix}$$

$$N = \eta\left(\,0\;0\;0\;0\;1\,\right)^T$$

$$R = \eta$$

The optimal state feedback control law that minimizes the cost function (11) is given by :

$$\Delta u = -K_c\,\Delta x \tag{18}$$

with optimal gain

$$K_c = Q^{-1}\left(B_r^T P_c + N^T\right) \tag{19}$$

where, P_c is the solution of the Riccati equation, given by:

$$A_r^T P_c + P_c A_r + R - \left(P_c B_r + N\right)Q^{-1}\left(P_c B_r + N\right)^T = 0 \tag{20}$$

The control law in (18) becomes

$$\Delta u = -K_c\,x_r \tag{21}$$

$$u = -\left(k_{c1}\;k_{c2}\;k_{c3}\;\right)\left(x - x_o\right) - k_{c4}\,r_1 - k_{c5}\,r_2 + u_o \tag{22}$$

SIMULATION RESULTS

The following set of parameters have been used to simulate the proposed controller[8,9]: engine inertia I_e=0.25 kg m^2, crankshaft friction coefficient b_e=0.0185 N ms, friction plate mean effective radius=0.7433 m, vehicle mass=1190 kg, gear ratio for the first gear =3.416, final drive gear ratio=5.2, tire radius=0.274 m,η = 0.2 α = 0.0003. The diesel engine (JL474 Q1) characteristic performance map is used. The engine is powered by 1.3 L-4 cylinders, 63.0 kW with speed range 800-6000 rpm. For all cases studied, the stationary point is considered as the initial point for the engagement process.

The effectiveness of the proposed controller: The performance obtained with the proposed controller are compared with those achievable by means of a classical Open Loop Control strategy (OLC), during the starting process for constant throttle opening (10%). Figure 1a shows the variation of the engine and the clutch rotating speeds during engagement time. For the open loop controller speed characteristic curves, the engine revolution increases rapidly, so the gap between the driving plate and the driven plate is large i.e. relative high slip speed, which result in more friction energy dissipation at the end of the engagement phase (Fig. 1b).The LQC achieves reduction by 18.0% of the friction losses, moreover, shortens the slipping time by 18.9%, reduces the jerk as shown in Table 1 and improve the rate of fuel consumption by 5.7% as shown in Fig. 1c. So, the proposed controller that has a gain matrix (-0.5225, -0.2136, 1.6899, -0.4081 and -0.4081), grantees a smooth acceleration transition, improves the riding comfort and enhance the durability of friction elements.

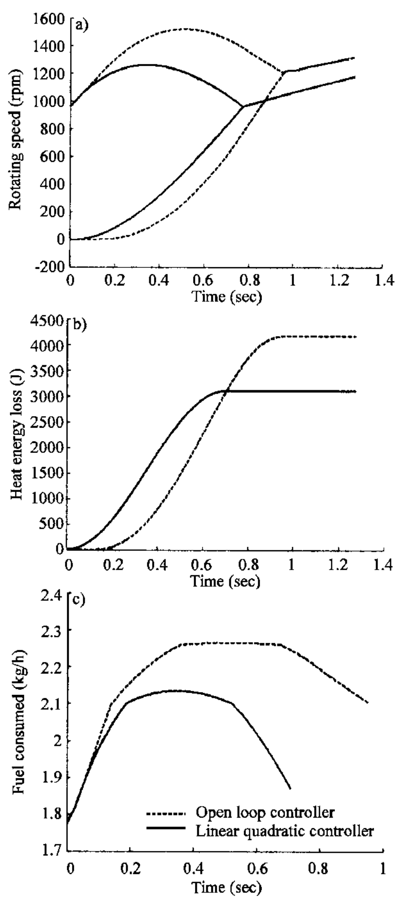

Fig. 1: Characteristic performance comparison between the proposed controller and the open loop controller during the slipping phase
a) Synchronization trajectories of engine clutch disk speed
b) Variation of the friction energy dissipation
c) Variation of the fuel consumption

Table 1: Comparison of the degree of jerk and slipping time for the two controllers

	Linear quadratic controller	Open loop controller
Slipping time (s)	0.78	0.96
Jerk (m/s^3)	-4.48	-11.80

Table 2: Slipping time for different friction plate material

	Leather on steel (μ=0.4)	Molden asbestos on steel (μ=0.25)	Cast iron on cast iron (μ=0.4)
Slipping time (s)	0.58	0.78	0.97
Maximum clutch torque (N m)	112.70	101.90	94.30

Table 3: Clutch slipping time and the transferable torque for driver intension

Driver intension	Crawl starting (12%)	Regular starting (17%)	Emergent starting (30%)
Slipping time (s)	0.76	0.74	0.69
Maximum clutch torque (N m)	104.70	114.90	131.30

LQC performance for different operating conditions: Different operating parameters such as loading, clutch friction plate materials and starting driver intension are used to investigate the dynamic performance of the LQC.

The effect of the load torque: Figure 2a-c show the performance of the LQC under different load torque conditions for constant throttle angle 10%. Different road grade 0, 3 and 5% is considered. Its clear that in Fig. 2a as the load torque increases, the lag time for the vehicle to start accelerating will increase. The reason is that the clutch driven disk starts rotating, when the load torque is equal to the clutch torque. Higher load torque causes lower vehicle acceleration during the slipping phase and elongate the slipping time (Fig. 2a). Moreover, the jerk is reduced as shown in Fig. 2b and a noticeable increase in the friction energy dissipation (Fig. 2c).

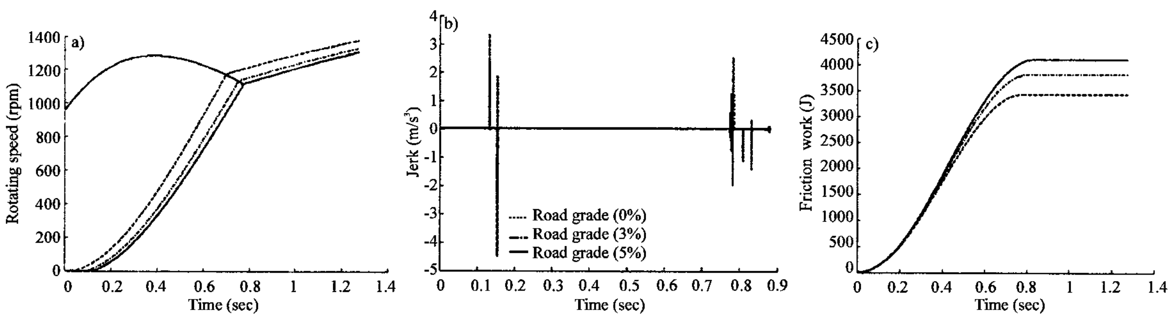

Fig. 2: Clutch dynamic response for different load torque conditions
a) Different synchronization trajectory of engine speed and clutch disk speed
b) Degree of jerk characteristic
c) Friction energy loss characteristic curves

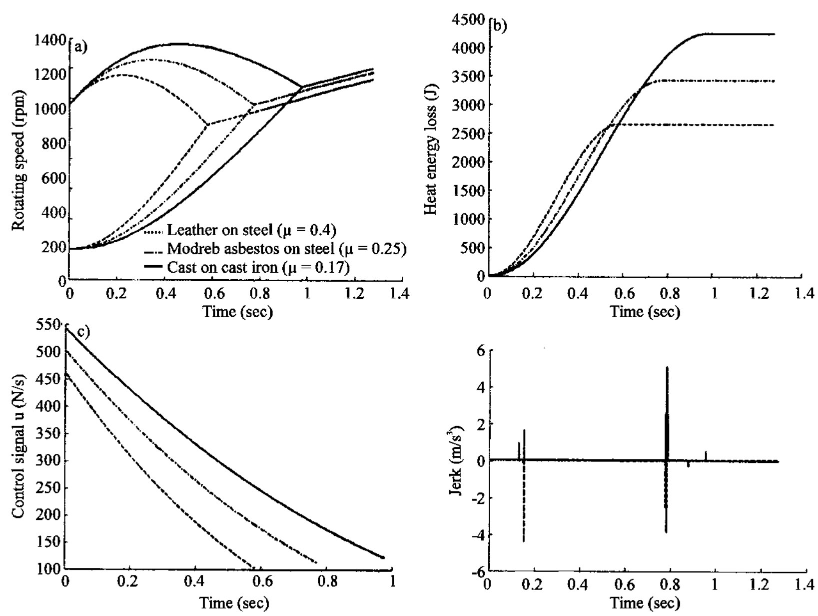

Fig. 3: Clutch dynamic response for different friction plate material
a) Synchronization trajectories of engine speed and clutch disk
b) Variation of friction energy dissipation
c) Variation of the control signal
d) Degree of jerk characteristic

The effect of the frictional plate material: As shown in Fig. 3a-d, different friction plate materials have influence on the clutch characteristic performance. Three different materials are used to evaluate the performance of the proposed controller: cast iron on cast iron, molden asbestos on steel and Leather on steel. Their dynamic coefficients of friction are 0.17, 0.25 and 0.4, respectively. As shown in Fig. 3a when the coefficient of friction increases, small slip speed, small slipping phase period and high torque capacity are provided (Table 2). This is due to the increase of the friction force and consequently, the friction torque. As shown in Fig 3b, in spite that the instantaneous friction energy dissipated is higher for higher coefficient of friction, the over all heat capacity is low when complete engagement is achieved. This is due to the small slip speed and the slipping time. Figure 3c shows the feedback control signal (the rate change of the clamping force). As the control signal increases for lower friction coefficient values, the engagement speed increases and the possibility for the acceleration instability will increase, providing relatively high jerk (Fig. 3d).

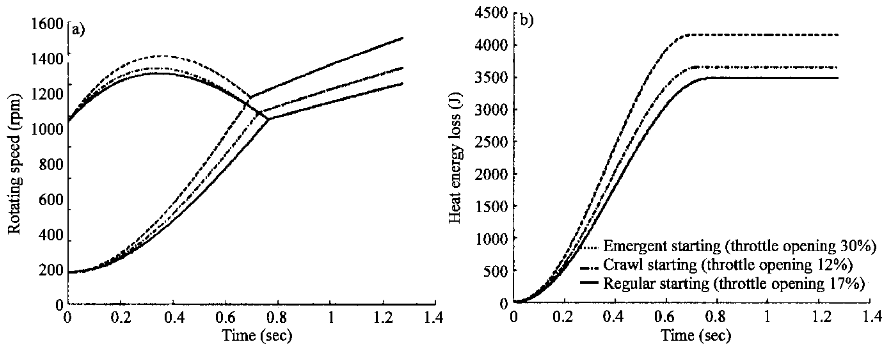

Fig. 4: Clutch dynamic response for different driver starting intension
a) Different synchronization trajectory of engine speed and clutch disk speed
b) Variation of the friction energy dissipation

The effect of the driver intension: Figure 4a and b show the characteristic performances of the LQC for different driver intension during the starting process. The driver transfers his or her starting operating intension through acceleration pedal in three different ways: crawl staring (opening<15%), regular starting (opening>15%) and emergent starting (opening>>15)[1]. The corresponding opening values used in the simulation model are 12, 17 and 30%. Figure 4a compares the slipping speed modes, we can see that for large throttle opening (emergent starting), these curves are shifting up and the engine rotating speed increases rapidly, generating higher vehicle acceleration, shorter engagement period and higher clutch transferable torque (Table 3) than the throttle opening for the other two types of driver intensions. In spite of these advantages, the thermal energy dissipation is high as shown in Fig. 4b.

CONCLUSIONS

In order to improve the starting performance of AMT vehicle, control of the dry clutch is considered. The control strategy is based on establishing a linear quadratic state feedback controller by solving finite time optimal control problem. The performance measure to be optimized comprises of the clutch slip speed as state variable and the rate change of the clamping force as control variable. Simulation is carried out by Simulink/Stateflow scheme, comparing the LQC characteristic performance with those achieved by open loop control strategy. Moreover, various clutch dynamic performance under different operating conditions were simulated. Numerical results show the good performance of proposed controller in minimizing the slipping time, reducing the friction energy loss, reducing the fuel consumption and enhancing the starting shift quality and the driving comfort. This control strategy can be extended to future work as an access for the engine torque feedback control to achieve the optimal transmission output torque during the clutch engagement for starting and shifting processes.

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