System Dynamic Modeling and Optimal Torque Control Strategy for E.T.Driver based on AMT

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Abstract: Hybrid power system is one of the kernel technologies of hybrid electric vehicles (HEV). The performance of HEV is determined greatly by the hybrid power system. The integrative hybrid power system is a new and important research field now and in the future. The Electrical Transmission Driver (E.T.Driver) is a compositive hybrid power system. The special constitution of E.T.Driver, which is based on Automatic Mechanical Transmission (AMT), can greatly improve the shift quality and driving smoothness of HEV during the gear shift course. A non-linear multi-rigid-body system dynamic modeling is developed for HEV system in power transmission during clutch engagement. Two kinds of input torque optimal control strategies of E.T.Driver are introduced and validated based on the dynamic simulation model and experiment data. A minimum value principle is used to optimize the input torque of E.T.Driver and to achieve an optimal dynamic performance of the non-linear system compromised in friction wear and shock intensity. It is found the shock intensity and slipping friction work can be reduced to a very small value or can be controlled to a satisfying degree according to the optimal input torque control strategies from the experiment data results.

Key words: electrical transmission driver (E.T.Driver); parallel hybrid electric vehicles (PHEV); gear shift quality; non-linear system dynamics; driving smoothness; optimal control

1 Introduction

Hybrid power system is one of the kernel technologies of HEV. The capacity of hybrid power system will determine the whole vehicle performance of HEV directly. The hybrid power system has already developed from discrete structure into integrative structure over the past ten years. In general, there are two approaches that can be adopted to carry out the power system integration in HEV. The first approach is engine-motor integration. For example, the engine and integrated starter/generator (ISG) can be integrated to form a light-duty HEV. The second approach is gear shift transmission system and electrical drive system integration. Some companies have already made some efforts in this research field.

For example, the Insight HEV of Honda Company can be counted as an attempt of the first integration approach. The AHS2 transmission of Allison Company and THS system of Toyota Company can be categorized as the attempt of the second integration approach. This paper presents the idea of E.T.Driver. The E.T.Driver integrates the motor [1][2] [3]and transmission into a power unit assembly which can perform the drive, generate electricity, regenerative braking function and power transmission function. On the one hand, the

E.T.Driver can control motor and transmission system concentratively, consequently helps to improve work efficiency of hybrid power system. On the other hand, the E.T.Driver helps to improve the integrated level and reliability by way of reducing the component number of the whole vehicle, thus helps to enhance the maintenance and use of the vehicle. The match cycle between different components and difficulty degree can be reduced; the research and production cycle can also be shortened because there are of fewer components involved.

The motor usually couples with the transmission through the input shaft of transmission in most HEV. The velocity is to decrease because the friction clutch will be detached and the drive torque will be interrupted during the gear shift course. Furthermore, since the moment of inertia of motor is added to the input shaft of transmission, the synchronization time between flywheel and friction clutch will be longer. Most researches of AMT are focusing on how to control the transmission driving torque transmitted by friction clutch and adjust the friction torque of clutch to reduce the slipping friction work and shock intensity. The control strategy is a little complex correspondingly. The special structure of E.TDriver can greatly reduce the input driving torque of transmission during the gear shift course. Therefore, both the slipping friction work of friction clutch and shock intensity of vehicle can be reduced to a very small value or be controlled to a satisfying degree. At the same time, the E.T.Driver can maintain the velocity stable during the gear shift course and increase the riding comfort.

2 The structures and the characteristics of E.T.Driver

2.1 The constitution of E.T.Driver

E.T.Driver may have many structure types. The design proposal is presented as follows:

(1) Motor with AMT is integrated into the E.T.Driver.

(2) Motor with epicyclic gear transmission is integrated into the E.T.Driver.

(3) Motor with continuously variable transmission (CVT) or motor with double clutch transmission is integrated into the E.T.Driver.

The development of the third type of E.T.Driver is based on (1) and (2).

3 The specific structure type of E.T.Driver based on the output shaft of AMT

There are two structure types of E.T.Driver, which are based on AMT. The type depends on whether the motor is mounted in the front or at the rear of AMT. The first type is to mount the motor on the input shaft of AMT and the rotor of motor is coupled with the input shaft of AMT either directly or with a coupling device. The second type is to mount the motor on the output shaft of AMT and the rotor of motor is coupled with the output shaft of AMT either directly or with a coupling device. A gear device, either an epicyclic gear or a pair of reducing gears, is adopted here as the coupling device. The function of the gear device is to reduce the output rotational speed and increase the output torque of motor at the same time. The motor also can couple its power directly to the input shaft or output shaft of AMT by fixing the rotor of motor directly onto the input shaft or output shaft of AMT. This paper mainly discusses the second type of E.T.Driver which is based on the output shaft of AMT. An epicyclic gear or a pair of common reducing gears can be adopted to help the motor output its power to the output shaft of AMT. The picture of E.T.Driver based on AMT output shaft is shown in Fig.1 and Fig.2.

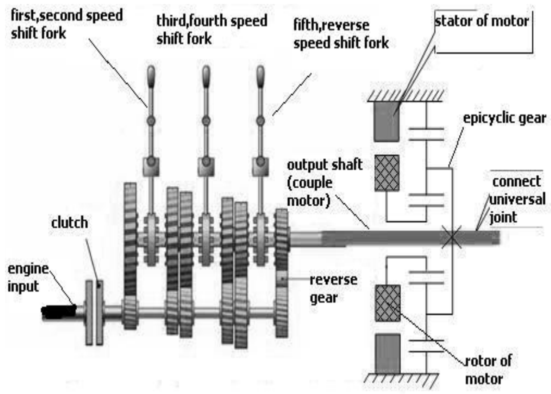


Fig.1 E.T.Driver for which the motor mounted on the output shaft of AMT, with its rotor coupling with the output shaft of AMT by epicyclic gear.

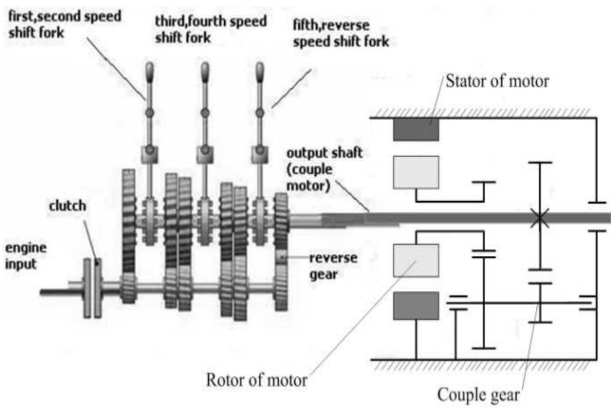


Fig.2 E.T.Driver for which the motor mounted on the output shaft of AMT, with its rotor of motor coupling with the output shaft of AMT by a pair of reducing gears

4 the configuration description of PHEV and dynamic modeling of PHEV

4.1 Configuration description of PHEV

The powertrain configuration of PHEV is illustrated in Fig.3. The diesel engine is connected with the input shaft of E.T.Driver through a single plate dry friction clutch.

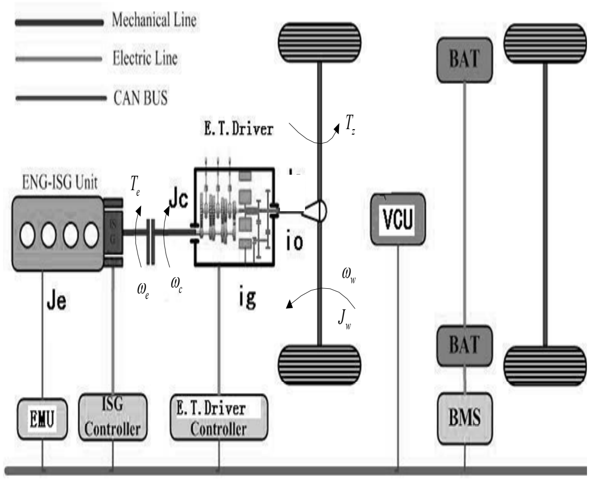


Fig.3 the powertrain configuration of PHEV

In Fig.3, the BMS is the abbreviation for battery management system; VCU is the abbreviation for vehicle control unit; BAT is the abbreviation for battery; EMU is the abbreviation for engine management unit.

4.2 The dynamic model of driveline

The major objectives of control system design of PHEV is to accomplish the aim below: to control the input torque of E.T.Driver and the engagement speed of clutch, so as to eliminate or reduce the abrasion of clutch and jerk of PHEV, and to enhance the driving smoothness of vehicle and riding comfort. The dynamic model has to be as simple as possible and can be used for the design and adjustment of the control system. Furthermore, the primary dynamics of powertrain system must be included in the dynamic model and the control strategy can be illuminated clearly and be accomplished in this dynamic model.

The PHEV powertrain driveline involved in clutch engagements can be treated as a non-linear multi-rigid-body system. For the purpose of simplification, the damping and elastic absorber of the clutch for reducing vibration and shock are ignored, so the driven plate of clutch and the input shaft of E.T.Driver are regarded as a rigid subassembly. The dynamic model shown in Fig.3 is established during the course of friction engagement in the process of power transmission.

The equation of motion during the friction clutch engagement can be denoted as [4] [5]:

$$J_e \dot{\omega}_e = T_e - T_c \qquad (1)$$

$$\bar{J}_v \dot{\omega}_c = T_c + T'_m / i_g - T_r \qquad (2)$$

Where J_e, T_e the total moment of inertia of flywheel, driving disk and crankshaft; Diesel engine output torque, the engine output torque is the function of engine angular velocity ω_e and acceleration pedal opening β, $T_e = f(\omega_e, \beta)$;

T_c, T'_m, T_r — friction clutch transmission torque,output torque of E.T.Driver and the resistance torque of PHEV applying on the driven plate;

ω_e, ω_c — engine angular velocity and driven-plate angular velocity, the dot above the angular velocity denotes the derivative with respect to time.

$\bar{J}_v$—the total moment of inertia of driven plate(J_c), wheels and vehicle body(J_w), driveline and E.T.Driver(J_d).

$$\bar{J}_v = J_c + J_w + J_d$$

The resistance torque T_r acting oppositely on the driven plate is composed of rolling resistance, wind resistance, ramp resistance and acceleration resistance as follows:

$$T_r = k(F_f + F_w + F_i + F_j) \qquad (3)$$

Where $k = r / i_o i_g$

i_o □ i_g —the ratio of differential and the gear ratio of E.T.Driver, r is the wheel rolling radius.

F_f, F_w, F_i, F_j,—the rolling resistance, wind resistance, ramp resistance and acceleration resistance respectively .

The output torque of E.T.Driver T'_m can be expressed in the following form:

$$T'_m = T_m \times i_m \qquad (4)$$

Where i_m — the gear ratio of couple gear which is shown in Fig2.

T_m — the output torque of motor used in E.T.Driver

5 The input torque control strategy of E.T.Driver under up shift condition

The engagement process of friction clutch comprises the following three stages [6-10]:

1) separated stage;
2) slipping stage;
3) engaged state ;

In the first stage, no torque is transmitted, so the driven plate should engage as quickly as possible. In the third stage, the engagement between the flywheel and driven plate is finished, and there is no slipping between the clutch plate and flywheel, thus there will be no abrasion for clutch. Therefore, the third states of engagement process need not to be taken into account. The first stage and the second stage of engagement process of friction clutch will be discussed in details in the next part of this section.

5.1 The function of proving auxiliary driving torque of E.T.Driver during the separated state of friction clutch in the gear up shift course

At the first stage of clutch engagement process, the clutch is separated, so the engine cannot provide drive torque for PHEV; therefore the driving smoothness will be affected. This problem can be solved in the PHEV which is equipped with E.T.Driver. As another power source, the E.T.Driver can provide auxiliary driving torque in PHEV. The special structure of E.T.Driver does not add additional moment of inertia to the gear shift course. As for a general PHEV, the induction motor will be mounted in the input shaft of transmission, which will add additional moment of inertia to the input shaft, therefore the synchronization time between input shaft and new gear will be increased and the abrasion of synchronizer sleeve will be accelerated.

In order to maintain the driving smoothness, firstly, the auxiliary driving torque provided by E.T.Driver is to be greater or equal to the resistance torque T_z.

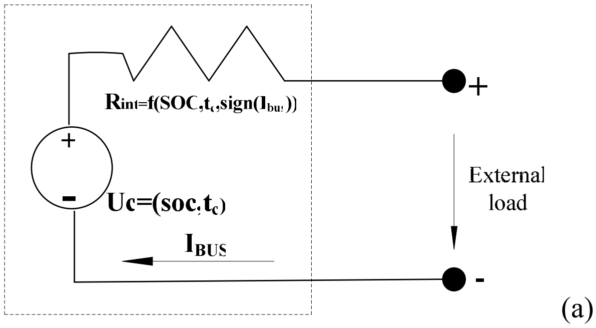
$$T_z = (F_f + F_w + F_i)r \quad (5)$$

Where T_z —resistance torque acting on the wheel.

Secondly, the output power of E.T.Driver cannot go beyond its power capacity and the power capacity of high voltage battery.

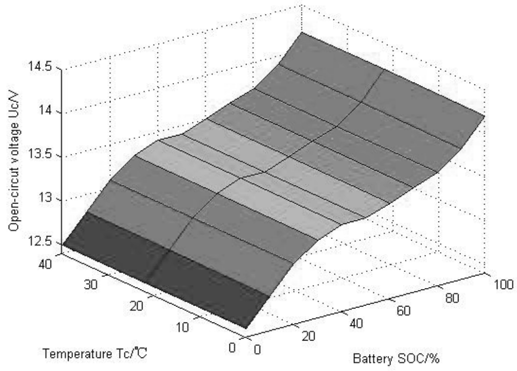
To calculate the maximum output power capacity of the high voltage battery pack, the model of high voltage battery pack, as shown in Fig.4, is constructed as follows.

The high voltage battery model consists of a perfect open circuit voltage (U_c) source in series with a resistor (internal resistance, R_{int}). The U_c is treated as a function of the battery SOC (state of charge), temperature (t_c) and the direction of current flow (I_{bus}).

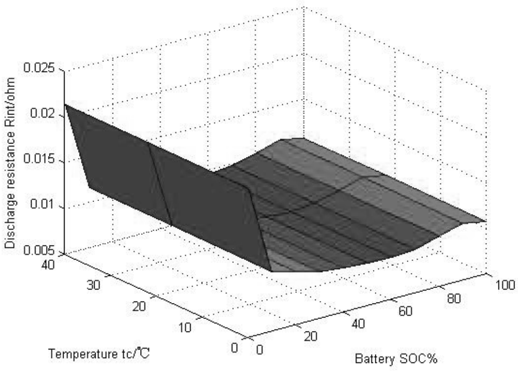


(a)

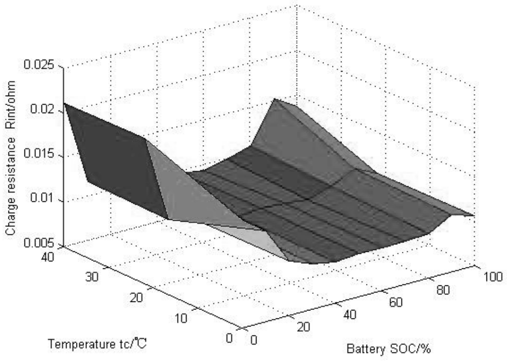
Inherent model of high power battery pack



(b) 12V battery module open-circuit voltage battery pack



(c) 12V module discharge resistance



(d) 12V module charge resistance

Fig.4 numerical model for 12V module of the NIMH

The battery is only based on the SOC and the losses during the course of charge and discharge are not to be considered. Inspired by the fact, G. Paganelli, et al. proposed the "penalty function f_P" [11] to correct the usage of high voltage battery for obtaining the higher overall energy efficiency. The "penalty function f_P" is shown in Fig.5.

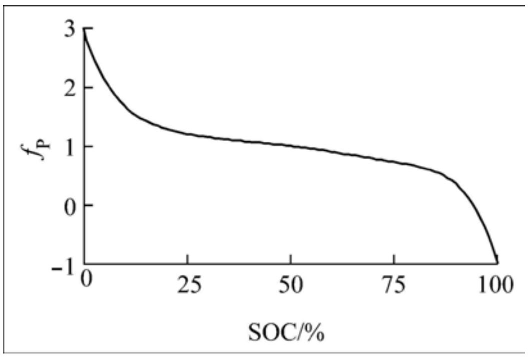


Fig.5 the penalty function f_P used for SOC correction factor

The allowable output power of high voltage is described as follows:

$$(U_c - \max(U_{b\min}, U_{m\min})) / R_{\text{int}} = I_{bus} \quad (6)$$

$$P_b = I_{bus} \times U_c = P_m \quad (7)$$

$$P_m = \frac{T_m \times \omega_m}{\eta_m} \quad (8)$$

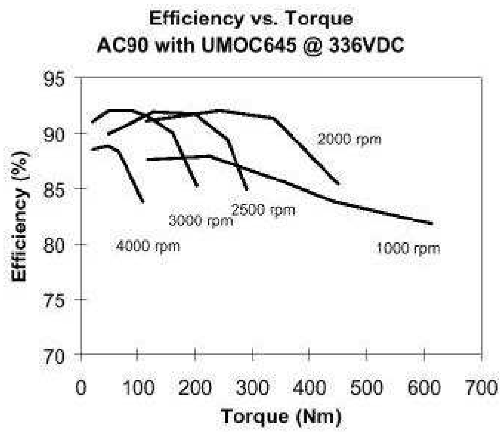
$$\omega_m = \frac{v \times i_o \times i_m}{3.6 \times r} \quad (9)$$

Where $U_{b\min}, U_{m\min}$—the normal lowest work voltage of high voltage battery, the normal lowest work voltage of motor used in E.T.Driver

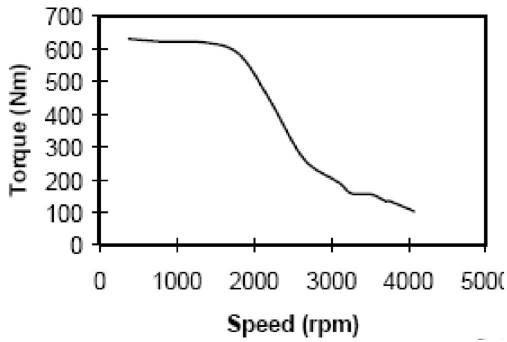
$P_b, P_m, \omega_m, \eta_m$—the allowable output power of high voltage battery; the output power of E.T.Driver; the angular velocity of motor used in E.T.Driver; the work efficiency of E.T.Driver;

v—vehicle velocity(km/h)

The numerical model of the motor used in E.T.Driver, including the effects of power losses in the E.T.Driver and its controller is depicted in Fig.6.



(a) Torque efficiency characteristic



(b) Torque vs. speed

Fig.6 the characteristic of motor used in E.T.Driver

So the E.T.Driver can provide its auxiliary driving torque during the separated state of friction clutch by calculating and judging the SOC of high voltage battery and the resistance of PHEV.

The allowable output power value of E.T.Driver can be calculated by equation (7), whereby the allowable output torque value of E.T.Driver can be calculated by equation (8), Fig.6 and current velocity.

If the allowable maximum output torque value of E.T.Driver is greater than or equal to the resistances torque T_z, the E.T.Driver only needs to provide the driving torque which is equal to the resistance torque T_z at the separated stage; if the allowable maximum output torque value of E.T.Driver is smaller than the resistance torque T_z, the E.T.Driver must output its allowable maximum driving torque at that stage.

Two kinds of optimal input torque control strategies are developed as follows according to the two kinds of output torque situation of E.T.Driver described above.

5.2 The optimal input torque control strategy for E.T.Driver when the output torque of E.T.Driver is greater than or equal to the resistance torque

If the output torque value of E.T.Driver is bigger than or equal to the resistance torque of vehicle, the friction clutch need not engage quickly in the separated state of friction clutch, because the E.T.Driver can provide the driving torque to satisfy the vehicle driving torque request. Under this condition, the vehicle velocity can be kept not to drop or be kept to increase continually. Because the clutch need not engage quickly, so the flywheel has enough time to track the rotational speed of the friction clutch plate. The target rotational angular velocity which flywheel is going to track is denoted in equation (10).

$$\omega_e^{aim} = \frac{v}{r} i_g i_o \qquad (10)$$

Where ω_e^{aim} —the target rotational angular speed of engine

When the rotational speed difference between the flywheel and clutch disk plate is smaller enough, that is to say, the rotational speed difference between the flywheel and clutch disk plate is smaller than what is set in advance, the flywheel and the clutch disk will engage as quickly as they can. During this course, the press force of driving plate is just big enough to ensure the flywheel and clutch disk plate can engage fully, after the engagement is finished, the press force will increase and the friction torque of clutch also will increase synchronously. The relationship between press force applied on the driven plate of clutch and the friction torque transmitted by driven plate of clutch can be described as follows:

$$T_{c} = n\mu_d \cdot F_n R_m sign(\omega_e - \omega_c) \qquad (11)$$

Where μ_d ——dynamic friction coefficient of clutch disk plate .

R_m ——Friction radius of clutch plates.

$$R_m = 2(R_0^{\ 3} - R_1^3)/3(R_0^{\ 2} - R_1^2)$$

R_0, R_1 ——Outer and inter radius of the clutch plate friction surface

n——Number of friction surfaces(n=2 for single plate clutch)

F_n ——Normal press force applied on the clutch plate

After the engagement is finished, the driving torque ratio between engine and E.T.Driver will be distributed anew according to the EMS (energy management strategy) of PHEV.

In this situation, the value of friction torque T_c transmitted by clutch plate is very small during the engagement course of clutch and the rotation speed difference between flywheel and clutch disk is also very small. The relationship between T_c and slipping friction works is shown in equation (12). From equation (12), we can see the slipping friction work produced by E.T.Driver is smaller than the one produced by primary AMT during the engagement course of clutch too.

$$\mathrm{W} = \int_{t_0}^{t_s} T_c(t)\left|\omega_e(t) - \omega_c(t)\right| dt \tag{12}$$

Where

W——slipping friction work (J) (it reflects how much mechanical energy is transferred to thermal energy and abrasion during the engagement course of clutch);

t_o——the time when the clutch begins to engage and transmits torque;

t_s——synchronization time when the clutch disk begins to synchronize with the flywheel;

The jerk mainly happens in the flywheel and clutch disk engagement course. The jerk intensity calculation equation is shown in equation (13).

$$\mathrm{J} = \frac{d^2 v}{dt^2} \tag{13}$$

The jerk intensity calculation equation also can be converted to equation (14) in real gear shift course.

$$\frac{dv}{dt} = \frac{T_c i_o i_g \eta_T + T_m i_m i_o \eta_T - T_Z}{mr\delta}$$

$$J = \frac{d^2 v}{dt^2} = \frac{i_0 i_g \eta_T}{\delta mr} \frac{d(T_c + T_m' / i_g)}{dt} \tag{14}$$

Where

J——jerk intensity(m/s^3);

η_T——efficiency of transmission;

m—vehicle mass(kg);

δ——vehicle rotational mass conversion factor;

Under this situation, since the rotational speed difference between the flywheel and clutch disk is very small, the jerk intensity is also very small during the engagement course of flywheel and clutch disk.

5.3 The output torque of E.T.Driver is smaller than the resistance torque

If the allowable maximum output torque of E.T.Driver is smaller than the resistance torque of vehicle, the E.T.Driver will provide the driving torque of vehicle as much as it can, and the rest requiring driving torque of vehicle is provided by engine. In this situation, the clutch needs to be engaged quickly to transmit the driving torque of engine. The flywheel maybe has not enough time to reach the same rotational speed of the clutch disk. Therefore, the rotational speed difference between the flywheel and the clutch disk will be produced .Under this condition, the slipping friction work can't be avoided. In order to speed up the engagement speed between clutch disk and flywheel and control the shock intensity under a satisfied level at the same time, the following state variables and transformation are defined:

$$x_1 = \omega_c \quad x_2 = \dot{\omega}_c \quad x_3 = \omega_e \tag{15}$$

$$u_1 = T_c + T_m' / i_g \quad u_2 = d(T_c + T_m' / i_g) / dt \tag{16}$$

Through substitution of the transformation in equation (15) and (16) into equation (1) and (2), the dynamic system of equation (1) and (2) can be rewritten as follows:

$$\dot{x}_1 = x_2 \tag{17}$$

$$\dot{x}_2 = \frac{u_2 - \dot{T}_r}{\bar{J}_v} \tag{18}$$

$$\dot{x}_3 = T_e / J_e - (u_1 - T_m' / i_g) / J_e \tag{19}$$

Equation (17) and (18) are related to the state variables x_1 and x_2; equation (19) is related with the state variable x_3.

In order to find a compromise between jerk and friction slipping work, an objective function is proposed as follows:

$$\int_0^{t_s}(J^2+Z)dt=\int_0^{t_c}(k^2\ddot{\omega}_c^2+Z)dt \qquad (20)$$

Where

J—jerk intensity;

t_s ——synchronization time when the clutch disk synchronizes with the flywheel;

Z—weighting coefficient for the shock intensity and tractive torque interrupt time, where the Z is defined as a function related with the opening and the time derivative of opening of brake pedal and acceleration pedal;

Z= $(\alpha,\dot{\alpha},\beta,\dot{\beta})$

Where α,β —the opening of brake pedal and acceleration pedal.

Obviously, the optimal control system of equation (15) to (20) is a typical minimum problem and can be solved by seeking the minimum of the function.

The Hamiltonian dynamic function which controls the engagement course of clutch can be written as follows:

$$H=k^2\dot{x}_2^2+Z+\lambda_1 x_2+\lambda_2\frac{u_2-\dot{T}_r}{\bar{J}_v} \qquad (21)$$

Where

λ_1,λ_2—the Lagrangian multipliers.

In equation (18), the gear shift time is very short, so the resistance torque T_r can be assumed as a constant. As a result the derivative of resistance torque $\dot{T}_r$ is equal to zero, which will help to simplify the dynamic system and control strategy in equation (17) to (21).

By solving the function and applying the initial, terminal and transversal condition according to the extremum value and the assumption, the optimal control of u^* in equation (17) to (19) can be derived and let:

$$u_1^*=\frac{\bar{J}_v\sqrt{Z}}{k}t+T_r \qquad (22)$$

$$u_2^*=\frac{\bar{J}_v\sqrt{Z}}{k} \qquad (23)$$

The friction clutch optimal control trajectory, including the angular speed and acceleration can be denoted as:

$$x_1^*=\frac{\sqrt{z}t^2}{2k}+v_0tk+v_0k \qquad (24)$$

$$x_2^*=\frac{\sqrt{z}t}{k}+\dot{v}_0\,k \qquad (25)$$

Where v_0 —the initial velocity when E.T.Driver begins to shift

$k=r/i_oi_g$

The jerk intensity of vehicle can be found in the flowing form:

$$|J|=\sqrt{z} \qquad (26)$$

We can see the jerk intensity is only determined by the Z from equation (26), so to limit the $|J|\le 10m/s^3$, the Z is limited to vary in the range of $0\le Z\le 100$.

From equation (22) and (23), we can see that the rate optimal control input torque is in direct proportion to the moment of inertia $\bar{J}_v$, resistance torque of vehicle, the gear ratio of differential, the transition ratio of E.T.Driver and the square root of the weighting coefficient but in inverse proportion to the rolling radius of wheel.

The gear ratio of differential and E.T.Driver can be gained before the gear shifting, so the control input driving torque is determined finally by the weighting coefficient Z. The upper limit of weighting coefficient Z can be set by the maximum value of jerk intensity and the lower limit of weighting coefficient Z can be set by preventing the friction clutch from over-abrasions. So the weighting coefficient Z can be computed by considering these restriction conditions, and then the weighting coefficient Z is provided to the E.T.Driver controller, the E.T.Driver controller will operate the actuator to finish a smooth and wearless engagement.

Furthermore, in order to avoid unintentional shut-off of the engine during the optimal input torque control course, the engine rotational speed must be maintained above a minimal value, so the target rotational speed of flywheel during the optimal control course is denoted as follows:

$$\omega_e^{aim} = \max\{\frac{v}{r} i_g i_o, \omega_e^{\min} - \dot{\omega}_e \Delta t\} \quad (27)$$

Where $\omega_e^{\min}$ —the minimal rotational speed of engine

Δt —time constant to compensate the lag of engine response

5.3.1 The special harmony function of E.T.Driver during the engagement course between flywheel and clutch disk

During traditional AMT engagement course, in order to control the jerk intensity under $10 m/s^3$, sometimes the detachment operation of clutch disk must be carried out to slow down the engagement speed between clutch disk and flywheel. But the jerk intensity of HEV equipped with E.T.Driver can be controlled under a satisfying level through regulating the output torque value of E.T.Driver instead of the detachment operation of clutch disk.

From equation (14), it can be found that the jerk intensity mainly determined by u_2^*, which is $d(T_c + T_m^{'}/i_g)/dt$.

The output torque of E.T.Driver can be kept unchanged or be reduced to keep the value of u_2^* under a satisfying level, so jerk intensity can be controlled under a satisfying level, the engagement speed between clutch disk and flywheel also can be speeded up on the condition of the engine not shut-off.

6 Input torque control strategy of E.T.Driver under the condition of gear down shift

6.1 Down shift course without braking

Because the driving torque request of vehicle is not very urgent under this condition, so the clutch can separate for a relative longer time to ensure the flywheel has enough time to track the rotation speed of clutch plate. When the rotational speed difference between the flywheel and clutch disk plate is smaller than the one what is set in advance, the flywheel and the clutch disk will engage as quickly as they can. During this course, the press force of driving plate is also just big enough to ensure the flywheel and clutch disk plate can engage fully, after the engagement is finished, the press force will increase and the friction torque transmitted by clutch also will increase at the same time.

If the VCU sends driving torque request signal before the flywheel synchronize with the clutch plate and if the output driving torque of E.T.Driver can satisfy the driving torque request of vehicle at the same time, the input torque control strategy of E.T.Driver under this condition will adopt the input torque control strategy described in section 5.2. If the output torque of E.T.Driver can not satisfy the driving torque request of vehicle, the clutch must engage quickly to transmit the friction torque provided by engine; the input torque control strategy of E.T.Driver under this condition will adopt the input torque control strategy described in section 5.3.

6.2 down shift course under braking condition

6.2.1 down shift course under common braking condition

Under common braking condition, this also is called foreseeable braking. For example, around the corner of road, swerve or passing each other. The driver will step on the brake pedal slightly and continuously or step on the brake pedal once and once and slightly. The clutch disk plate does not separate and the E.T.Driver does not begin to shift until the brake signal disappears. The input torque control strategy described in section 6.1 is adopted in succedent clutch engagement course.

During common braking down shift course, the E.T.Driver can provide reverse generation of electric torque to assist braking.

6.2.2 down shift course under Emergency braking condition

Under emergency braking down shift condition, the clutch should be separated in order to avoid engine flameout. The clutch disk plate will engage with flywheel after the braking signal disappears and new gear engagement finishes. If the VCU sends driving torque request during the engagement of clutch and if the output driving torque of E.T.Driver can satisfy the driving torque request of vehicle at the same time, the input torque control strategy of E.T.Driver under this condition will adopt the input torque control strategy described in section 5.2. If the output torque of E.T.Driver can not satisfy the driving torque request of vehicle, the input torque control strategy of E.T.Driver under this condition will adopt the input torque control strategy described in section 5.3.

If VCU does not send driving torque request during the engagement of clutch, the input torque control strategy of E.T.Driver under this condition will adopt the input torque control strategy described in section 6.1.

The input torque control strategy flow chart of E.T.Driver during the separated state of friction clutch is shown in Fig.7.

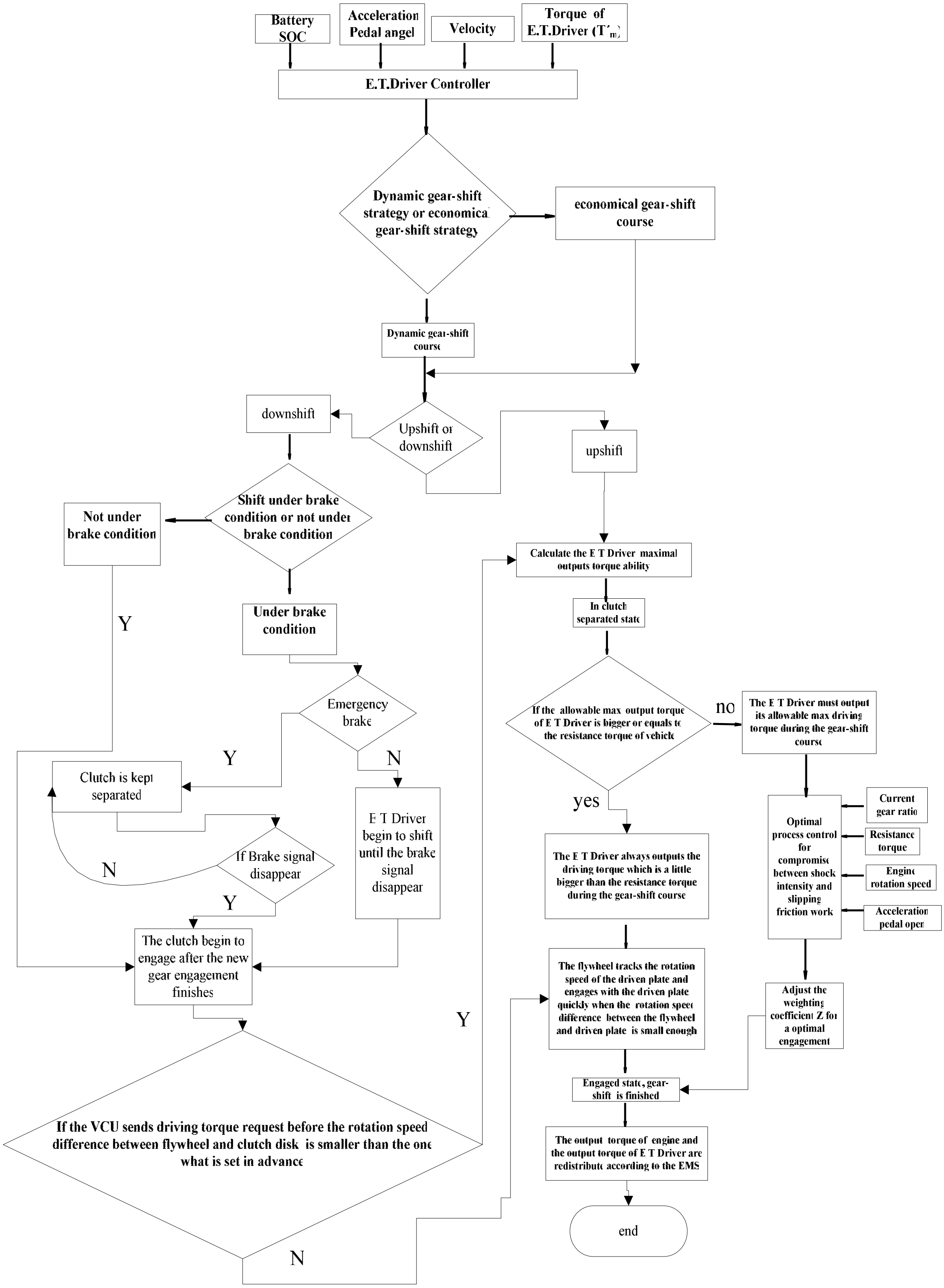


Fig.7 the input torque control strategy of E.T.Driver during the gear shift course

7 The experiment results

The parameters of E.T.Driver and the main components of hybrid electric buses are shown as follows.

1)engine——the numerical model of engine is shown in Fig.8.

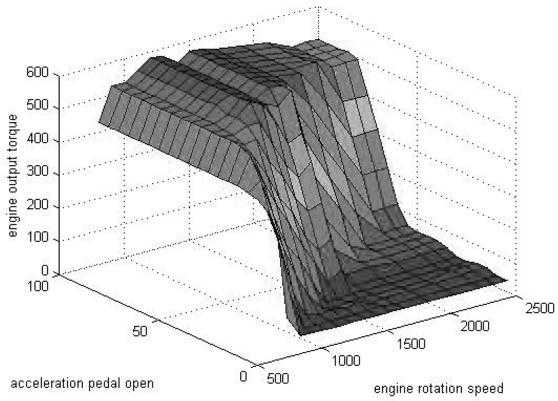


Fig.8 numerical model of engine

2) The parameters of friction clutch are shown in Table 1.The ratio of E.T.Driver is shown in Table 2.

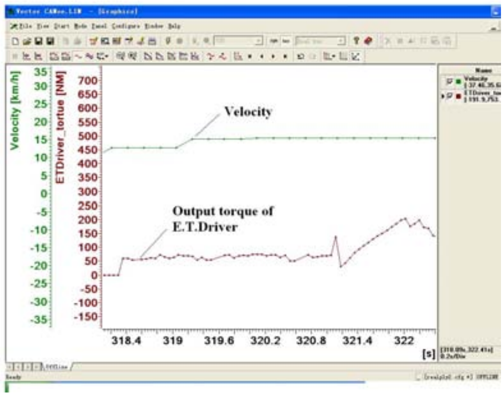
Table 1 the friction clutch parameters

title	value
H□release lever adjust height□	75±0.4(mm)
Mcmaximum□Nm□□maximum. friction torque of clutch□	19656±1645
m1□kg□□mass of pressure plate unit assembly□	44
m2□kg□(mass of clutch disk unit assembly)	8.3
J□kg.m^2□(inertia of pressure plate unit assembly)	1.369
n (rpm)(maximum. enable rotational speed of clutch)	2700
P(N) (pressing force of pressure plate)	2090
F(N) (maximumimal detachment force)	4644
S(mm)(detachment distance)	10^2

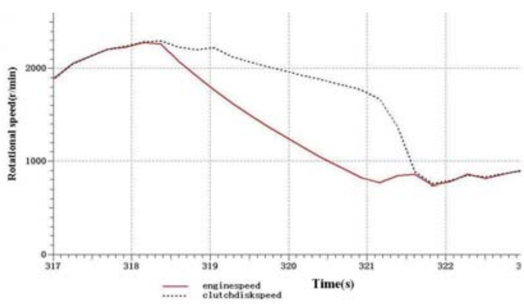
Table2 ratio of speed of E.T.Driver

Gear	Ratio	Number of input gear teeth	Number of output gear teeth
reverse	5.22	27	141
1st speed	6.11	18	110
2nd speed	3.389	59	200
3rd speed	2.05	20	41
4th speed	1.32	25	33
5th speed	1	10	10

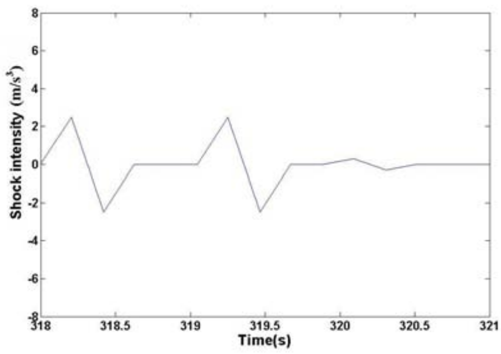
The experiment results are shown in Fig.9 and Fig.10. The data is a part of road experiment results; the road experiment has been carried out according to GB/T 19754-2005 □Test methods for energy consumption of heavy-duty hybrid electric vehicles□.



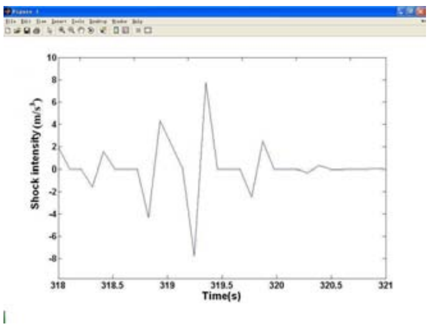
(a) The output torque of E.T.Driver and velocity change situation during the engagement course from first gear to third gear



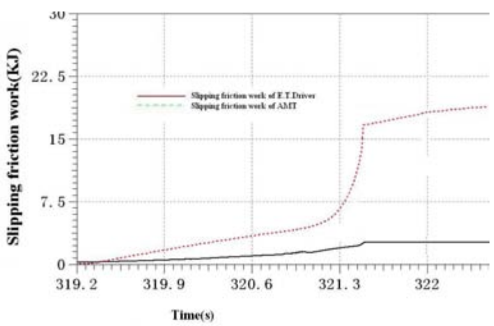
(b) synchrodrive response of the clutch plate from first gear to third gear, from 318s to 320.96s



(c) Shock intensity change situation during first gear to third gear course

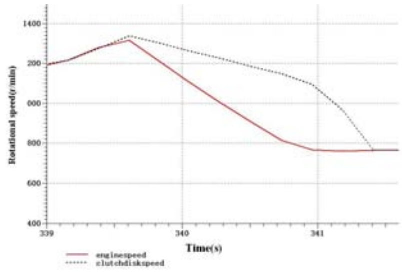


(d) Shock intensity changes situation of traditional AMT from first gear to third gear course

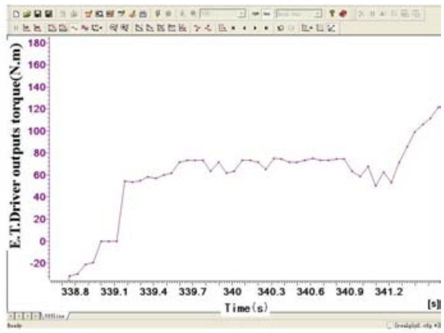


(e) Contrast of slipping friction work between E.T.Driver and AMT from first gear to third gear

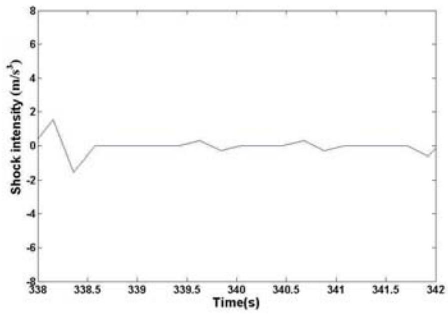
Fig.9 the shift course from first gear to third gear and gear shift quality situation



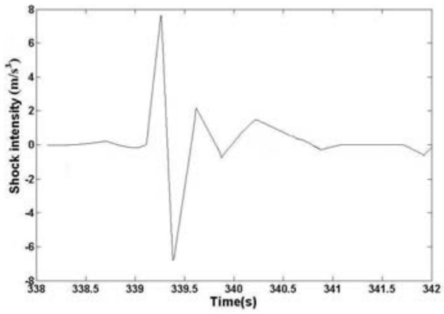
(a) Synchrodrive response of the clutch plate from third gear to fourth gear course



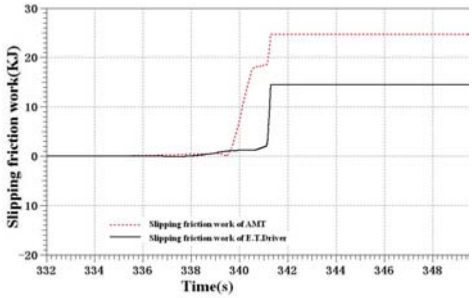
(b) Output torque change of E.T.Driver during clutch separate course



(c) Shock intensity change situation of E.T.Driver from third gear to fourth gear



(d) Shock intensity change situation of traditional AMT from third gear to fourth gear



(e) Contrast of slipping friction work between E.T.Driver and AMT from third gear to fourth gear course

Fig.10 the shift course from third gear to fourth gear and gear shift quality situation

From the Fig.9 and Fig.10, it can be found that the driving torque of vehicle do not be interrupted because the E.T.Driver can provide driving torque during the clutch separate course. The velocity can be kept steady or be kept continually increasing during the gear shift course. The shock intensity and slipping friction work of vehicle equipped with E.T.Driver are obviously much better than the vehicle equipped with AMT. The driving smoothness of the vehicle equipped with E.T.Driver is also enhanced greatly compared to the vehicle equipped with AMT [12].

8 Conclusions

The HEV compositive power system is the kernel technology of HEV and is also a new and important research field today and in the future.

In this study, the characteristics and the constitution of the E.T.Driver based on the output shaft of AMT are introduced. The non-linear system dynamics model is constructed. It is found that because of the special structure of E.T.Driver based on the output shaft of AMT and the optimal input torque control strategies[13], the gear shift quality, driving smoothness and riding comfort of HEV equipped with E.T.Driver can be greatly improved compared to the vehicle equipped with AMT during the gear shift course from the experiment results.

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