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(54) **METHOD OF CONTROLLING THE CLOSING PHASE OF A CLUTCH OF AN AUTOMATED AUTOMOBILE TRANSMISSION SYSTEM**

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(57)　　　　　　**ABSTRACT**

A method of controlling an automobile clutch in an automated transmission system with a CMPC control is disclosed having application to vehicle clutch control in an AMT system. The driver's request is translated in terms of sliding velocity ω_{sl}. Constraints on the engine and clutch actuators are defined to respect their operating limits, and driving quality constraints are defined to guarantee comfort during the clutch engagement phase. In order to meet these quality constraints, a reference trajectory is defined for ω_{sl} as a function of the clutch engagement time. An analytical expression allowing real-time calculation of a set of control trajectories with a CMPC control law is then defined from the expression of this reference trajectory. The trajectory respecting the constraints on the actuators is selected from among all these control trajectories. Finally, the clutch is controlled with the selected control trajectory.

24 Claims, 2 Drawing Sheets
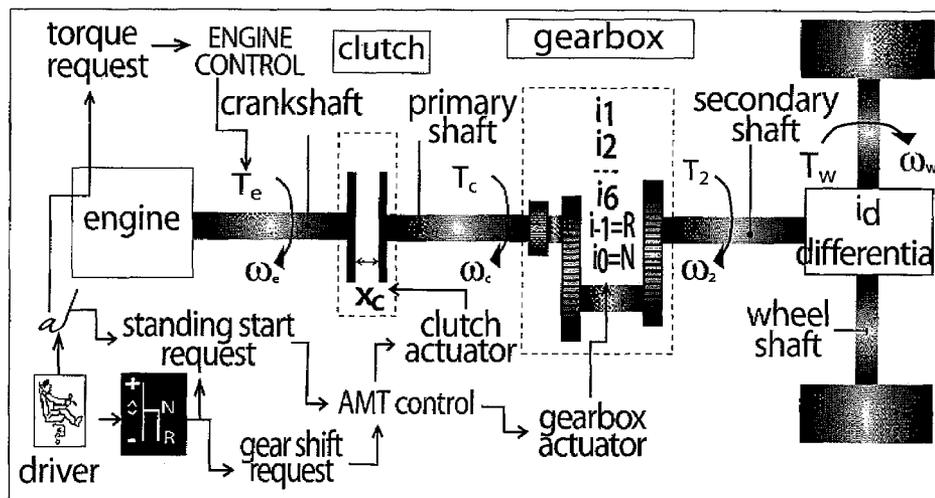
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Fig. 1

Fig. 2

Engine speed

Torque request

Engine torque set value generation → Engine torque set value → Engine torque control → Engine → Engine torque → ⊙ → Crankshaft

Clutch torque set value generation → Clutch torque set value (position) → Clutch actuator → Clutch Position → Transmissibility curve → Clutch torque transmitted → Transmission and vehicle

Load torque

Primary/secondary/wheel (acceleration) speed

Fig. 3

$\omega_{si}(K)$ [rad/s]

120
100
80
60
40
20
0

← N_0
← NN
← N
← N_{max}
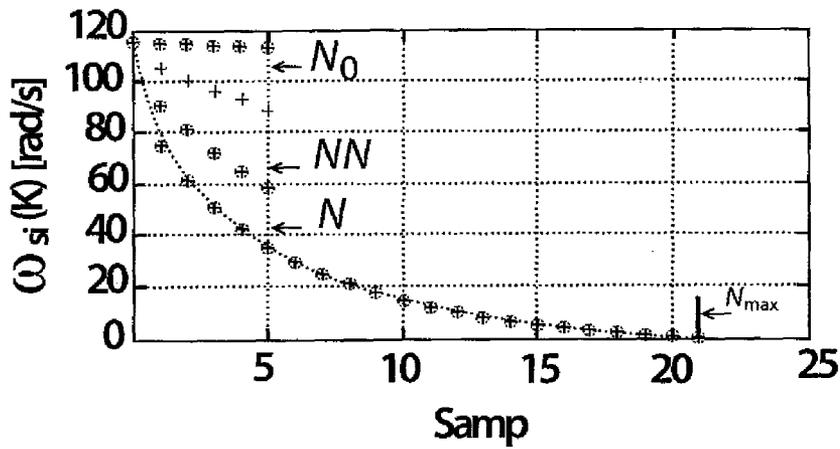
5   10   15   20   25

Samp

Fig. 4

METHOD OF CONTROLLING THE CLOSING PHASE OF A CLUTCH OF AN AUTOMATED AUTOMOBILE TRANSMISSION SYSTEM

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to automatic control systems for vehicle engines. In particular, the invention relates to a method for controlling the closing phase of a clutch in an automated automobile transmission system.

2. Description of the Prior Art

Automated transmission systems, or automated manual transmission (AMT) systems, are an intermediate solution between conventional manual transmissions and automatic transmissions. Unlike the latter, characterized by complex mechanical components such as torque converters and epicyclic trains, automated transmissions use devices (clutch, gearbox) having conventional mechanics controlled not by the driver but by one or more on-board control unit(s). FIG. 1 shows a generic diagram of an automated transmission. FIG. 2 shows a conceptual diagram of an automated transmission.

Any automated transmission control system, having on-board one or more ECUs (Electronic Control Units), has to generate set points for:

(electric, pneumatic or hydraulic) control of the position of the clutch (clutch actuator);

(electric, pneumatic or hydraulic) control of the gears of the gearbox (gearbox actuator(s)), and expressing the gear change selected by the driver.

Thus, in an automated transmission, it is the ECUs dedicated to transmission control that allow responding to the driver's requests (standing start and gear shift), through the aforementioned actuators, in collaboration with the engine control ECU that determines the torque produced by the engine.

Standing start is the maneuver of setting a stationary vehicle in motion by transmitting an increasingly large fraction of the engine torque to the primary transmission shaft, therefore to the wheels, through progressive closing of the clutch. The clutch (progressively, then fully) couples the engine flywheel to the primary shaft, thus transmitting the torque produced in the engine (minus the friction of the engine crankshaft assembly) to the primary shaft. In this scheme, the clutch can therefore be:

completely open (disengaged, out of gear), with a zero torque transmitted to the primary shaft,

completely closed (engaged, geared), with the engine torque fully transmitted to the primary shaft,

sliding, closing or opening. It is in the closure sliding phase that progressive transmission of the engine torque to the primary shaft takes place.

In the case of a conventional manual transmission, the driver achieves progressive closing of the clutch by adjusting simultaneously the pressure exerted on the clutch pedal and the pressure exerted on the accelerator pedal. His or her experience as a driver will determine the successful outcome of the maneuver or its failure (engine stalling, over-revving, strong oscillations). In the case of an automated transmission, the clutch pedal is absent and it is the transmission control system that adjusts the action of the clutch and coordinates it with the engine torque production so as to ensure smooth progress of the standing start maneuver.

Progressive closing of the clutch and its coordination with the engine torque production thus is the key phase of the automated transmission control. To understand the operation of the powertrain in this phase, the torque transmitted by the

clutch (engine torque fraction) has to be considered as a negative torque on the crankshaft side (thus decreasing the net torque provided by the engine) and as a positive torque on the transmission side downstream from the clutch, minus the transient and static losses due to elasticities, friction and the efficiency of each mechanical element. It is this torque that, geared down by the gear ratio, is transmitted to the wheels. The state of the powertrain is then defined by variables measured upstream and downstream from the clutch. Typically, the engine speed upstream, is always available on a vehicle, plus (at least) a speed measurement downstream from the clutch which is primary shaft speed, secondary shaft speed or wheel speed. These variables are not systematically measured on a standard vehicle, but they have to be measured for a vehicle equipped with an automated transmission.

In order to fulfill the main two functions of an automated transmission system, standing start and gear shift, the control systems of standard vehicles generally calculate the set point of the clutch actuator from pre-filled charts (mapping) as a function of the torque required by the driver (accelerator pedal position), the engine speed, the primary shaft speed (or other speed on the transmission side) and other parameters such as the gear ratio (in the case of gear shifting).

Regarding the engine control, the engine speed must be ensured to remain compatible with the maneuver being considered, despite the negative torque acting on the crankshaft (the engine control therefore has to increase the engine torque accordingly). From a systemic point of view, this type of control structure is but a particular case of the general structure of FIG. 3. This diagram shows that, in the control of the clutch sliding phase, there are two "levers" which are the engine torque that acts only upon the crankshaft and the torque transmitted by the clutch, which acts both on the crankshaft (as a negative torque) and on the downstream of the transmission up to the wheels. There are, for the control system, and using the Automatic terminology, two actuators that act at the input of the powertrain system. In order to know the state of the system and to act accordingly, it is necessary to measure at least two outputs, the engine speed and one of the clutch downstream speed which, for example, may be the primary shaft speed. The inputs of the powertrain system, that is the engine and clutch torques, are never measured in a standard vehicle and are only assessed, with much imprecision.

This mapping control, typical of standard vehicles, does not allow readily translating the specifications to be met during clutch closing which are compliance with the driver's requests, comfort and maintenance of powertrain smooth running. Besides, a long calibration time to fill in the maps.

In order to do without methods based on mapping, there are known solutions based on control laws inspired by the Automatic principles. These laws are no longer based only on charts that have been previously filled in. These solutions are based on algorithms that calculate the input data to be sent to the powertrain system, engine torque and clutch torque, from measurements of the state of this system (typically engine speed and primary shaft speed), designed using the feedback principle.

For these feedback control laws to be usable within the context of the engine control of a vehicle, they must allow meeting a number of specifications which respect constraints guaranteeing smooth running of the thermal engine, constraints guaranteeing comfort upon clutch engagement (no oscillation), and the driver's will.

Now, these techniques cannot explicitly manage these constraints to ensure that one or more variables of the system to

be controlled (inputs, outputs or state), or their derivatives, do not exceed certain limits set as specifications.

SUMMARY OF THE INVENTION

Thus, the invention is an alternative method for real-time control of an automobile clutch in an automated transmission system which by use of a control law allows overcoming the drawbacks of prior techniques. The method therefore translates the driver's request in terms of sliding velocity ω_{s1} and of engine speed ω_e, and it uses a CMPC control law to control this sliding velocity and the engine speed.

The invention is a method of controlling an automobile clutch in an automated transmission system, within a vehicle comprising an engine connected to a primary shaft via the clutch. The method comprises the following:

translating a driver's request on the accelerator pedal in terms of sliding velocity ω_{sl}, defined by the difference between an engine speed ω_e and a primary shaft speed ω_c;

defining constraints on actuators of the engine and of the clutch so as to respect their operating limits;

defining driving quality constraints to guarantee comfort during a clutch engagement phase;

in order to respect the quality constraints, defining a reference trajectory for the sliding velocity by defining, for each control interval, a set value ω_{sl}^{ref} for the sliding velocity as a function of a clutch engagement time N^*_f;

from the expression of the reference trajectory, defining an analytical expression allowing real-time calculation of a set of control trajectories for the sliding velocity, by a constrained model predictive control law (referred to as CMPC control);

selecting, from the set of the control trajectories, the trajectory that respects the constraints on the actuators by a dichotomic search technique; and controlling the clutch with the selected control trajectory.

According to the invention, the constraints on the actuators can be defined by imposing minimum and maximum values on the engine torque and on the clutch torque, as well as minimum and maximum values on the derivatives of these torques.

The quality constraints can be defined by requiring a zero sliding velocity and sliding velocity derivative during the clutch engagement phase.

According to the invention, set value ω_{sl}^{ref} can be defined for each control interval k by the following relation:

$$\forall\, i \in \{1, \ldots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N_f^*}{(1 + \lambda \cdot i/N_f^*)^2}\, \omega_{sl}^k$$

with:

N_p being an integer defining a prediction horizon; and
λ being a parameter to be calibrated.

Parameter λ is advantageously selected in such a way that engagement occurs with a quasi-zero sliding velocity derivative.

Finally, a second set value ω_{sl}^{ref} can be generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;
T is a maximum torque depending on speed ω_e;
T_e^d is an engine torque required by the driver; and
X_{Pedal} is a position of the accelerator pedal.

BRIEF DESCRIPTION OF THE DRAWINGS

Other features and advantages of the invention will be clear from reading the description hereafter of non limitative embodiment examples, with reference to the accompanying figures wherein:

FIG. 1 shows an automated transmission generic diagram;

FIG. 2 shows a conceptual automated transmission diagram;

FIG. 3 illustrates a general structure of a control for the clutch sliding phase; and

FIG. 4 illustrates the principle of a dichotomic search for the predictions for the sliding velocity respecting the constraints.

DETAILED DESCRIPTION OF THE INVENTION

Progressive closing of a clutch and its coordination with the production of a given engine torque is a key phase of automated transmission control. The following points have to be ensured during this phase:

maintenance of the powertrain (the system) smooth running: no stalling, no engine over-revving;

driver's comfort: oscillation minimization upon clutch closing;

respect for the driver's request: for example, shorter standing start if one presses the accelerator pedal down harder, standing start abandoned if one releases the pedal.

According to the invention, in order to take into account these specifications in a transparent manner (for the driver), an approach based on a control law that can optimize a criterion while managing constraints is used. The technique referred to as constrained model predictive control (CMPC) described, among others, in D. Q. Mayne, J. B. Rawlings, C. V. Rao, and P. O. Scokaert, "Constrained Model Predictive Control: Stability and Optimality," Automatica, 36:789-814, 2000, is therefore used. The possibility of integrating constraints on the inputs, the outputs and the states of the system (for example minimum and maximum engine speed, clutch torque variation rate) makes this type of control particularly well suited for the technical problem of the invention.

However, the CMPC control technique, in its standard formulation, is not applicable to powertrains that have fast dynamics. In fact, to be efficient, a control system for the engine-transmission assembly must be able to calculate a new control at relatively short time intervals (some ten milliseconds maximum). The standard optimization formulation for the CMPC control does not allow carrying out this calculation in such a short time. It is not possible to use a conventional CMPC control law to perform an on-line (real-time) optimization.

According to the invention, a new approach is constructed to optimize calculation of the CMPC control within time limits compatible with the requirements of an automobile control system. It is thus possible, while integrating the specifications required for smooth running of a standing start as regards driving quality and consumption

to calculate the control very rapidly in real time, with periods compatible with the calculation of power of on-board series vehicle-mounted control units; and

translate the driver's will in a transparent manner.

This method of controlling in real time an automobile clutch in an automated transmission system comprises the following:

translating the driver's request in terms of sliding velocity ω_{sl}, defined by the difference between engine speed ω_e and primary shaft speed ω_c;

defining constraints on engine and clutch actuators so as to respect their operating limits;

defining quality constraints to guarantee comfort during a clutch engagement phase;

defining a reference trajectory for the sliding velocity;

calculating a set of control trajectories with a CMPC control law; and

selecting, from all these control trajectories, the trajectory that respects the constraints on the actuators using a dichotomic search technique.

Thus, according to the invention, the specifications are met as follows:

maintenance of smooth powertrain running by applying constraints to measured variables, such as the engine speed engine speed above a minimum value, to prevent stalling and engine speed below a maximum value, to prevent over-revving;

respect for the driver's will by translating the driver's request in terms of sliding velocity ω_{sl}, defined by the difference between engine speed ω_e and primary shaft speed ω_c;

driver's comfort upon clutch engagement, by applying constraints to sliding velocity ω_{sl} (also referred to as sliding speed); and

determination of a control law for the clutch respecting these specifications in real time, using sliding velocity ω_{sl} and engine speed ω_e as parameters to be controlled.

Before describing in detail the stages of the method, it should be recalled that a CMPC control system comprises:

1—an internal model (control model) capable of simulating the behavior of the system to be controlled more rapidly than real time;

2—a reference trajectory that defines the desired behavior in a closed loop;

3—the receding horizon principle: only the first component of the optimum control trajectory is applied, while repeating all the calculations in the next sampling interval;

4—characterization of the control trajectory by a finite number of "moves" (or other parameters); and

5—constrained on-line optimization to determine the future control strategy.

1—Definition of a Model Simulating the Behavior of the Powertrain

It defines a control model capable of simulating the behavior of the system to be controlled more rapidly than real time. According to an example, a non-linear system in the form of a discrete-time state is used as the control model:

$$\begin{cases} X_{k+1} = f(X_k, u_k) \\ y = g(X_k) \end{cases}$$

where $X \in R^n$, $\mu \in R^m$, $y \in R^q$ are respectively the state of the system, the controls (system inputs) and the system outputs (controlled variables).

In the case of clutch control:

the system controls (U_k) or system inputs can be:

the engine torque set value; and

the clutch torque set value.

The system outputs (Y_k) can be:

the engine speed ω_e;

the primary shaft speed ω_c;

or another speed of the transmission directly linked with the primary shaft speed.

The system state X_k can be

all the variables indicating the physical state of the powertrain (speeds or linear combinations of speeds). This state is deter-

mined by the complexity selected for the model while accounting for torsions, frictions, mechanical play.

CMPC predictive control (standard) calculates at each control period an optimum control sequence which according to the invention is calculated is a set of control sequences depending on a parameter that specifies a sliding velocity trajectory to provide a solution to a constrained optimization problem:

$$\begin{cases} J_k(\cdot) = \sum_{\substack{i=k+1 \\ UNDER\,THE\,CONSTRAINTS}}^{k+N} \begin{pmatrix} (r_k - y_k)^T Q(r_k - y_k) + \\ (u_{k-1} - u_{k-2})^T R(u_{k-1} - u_{k-2}) \end{pmatrix} \\ X_{k+1} f(X_k, u_k) \\ y = g(X_k) \\ u_k = u_{k-1} + du_k \\ X_{min} \le X_k \le X_{max} \\ \alpha_k \le u_k \le \beta_k \\ du_k^{min} \le du_k \le du_k^{max} \end{cases}$$

where e is the error between the set value and the real measurement, N the prediction horizon, Q and R are weighting matrices respectively associated with the set value-measurement errors and with the controls.

A simple model disregarding certain transmission dynamics can be selected. In this case, uncertainty terms that can be reconstructed by estimation are preferably introduced. The simplest model has the form as follows:

$$J_e \dot{\omega}_e = T_e^{SP} - T_e^{SP} + \delta_e(\bullet)$$

$$[J_c + J_{eq}(i_g, i_d)] \dot{\omega}_c = \text{sign}\,(\omega_{sl}) T_c^{SP} - \delta_e(\bullet)$$

where:

J_e is the inertia of the engine, the crankshaft and the clutch part on the engine side;

J_c is the inertia of the primary shaft and of the rest of the clutch;

$J_{eq}(i_g, i_d)$ is the equivalent inertia of the transmission downstream from the clutch that depends on the engaged gear ratio i_g and on the differential ratio i_d;

ω_{sl} is the sliding velocity between engine and primary shaft;

T_e^{SP} is the engine torque set value that the engine control system translates into a net torque T_e on the crankshaft; and

T_c^{SP} is the clutch torque set value that the clutch control system translates into a torque T_c transmitted by the clutch.

The term $\delta_e(\bullet)$ brings together all the uncertainties affecting the crankshaft dynamics which is friction in the engine, errors in the engine parameters, errors in the clutch torque transmissibility curve and neglected dynamics, etc.

The terms $\delta_e(\bullet)$ and $\delta_c(\bullet)$ are reconstructed by dynamic estimation such as for example by Kalman type estimators.

More complex models can be used. However, it is always better to integrate uncertainty terms to be reconstructed in parallel with the control, through the available measurements.

2—Definition of the Constraints

Two types of constraint are defined which are the constraints on the engine and clutch actuators, denoted by C_k^U, and the driving quality constraints, denoted by C_k^G.

The constraints on the actuators, C_k^U respect the engine and clutch operating limits. For example, the clutch must not transmit more than x NM torque during standing start, or the engine should not be required to produce a larger amount of

torque than it can at a given speed. These constraints on the actuators are typically limitations on the maximum and minimum values of the actuator torques and their derivatives:

$$T_e \in \lfloor T_e^{min}, T_e^{min}(\omega_e) \rfloor$$

$$T_c \in \lfloor T_c^{min}, T_c^{min}(\omega_e) \rfloor$$

$$\dot{T}_e \in \lfloor \dot{T}_e^{min}, \dot{T}_e^{min}(\omega_e) \rfloor$$

$$\dot{T}_c \in \lfloor \dot{T}_e^{min}, \dot{T}_e^{min}(\omega_e) \rfloor$$

The quality constraints C_k^G must first ensure that there are no oscillations upon engagement, which occurs at the time t_f, a condition provided by the equality of the speeds and of the accelerations upon engagement, that is, at the time t_f:

$$\omega_e(t_f) = \omega_c(t_f)$$

$$\dot{\omega}_e(t_f) = \dot{\omega}_c(t_f)$$

This stability condition can be ensured by constraints on the sliding velocities between the engine and the primary shaft:

$$\omega_{sl}(t_f) = 0$$

$$\dot{\omega}_{sl}(t_f) \cong 0$$

where ω_{sl} is the sliding velocity between the engine and the primary shaft. Maintaining these constraints for a sufficient time also allows ensuring stability on the rest of the transmission (equality of speeds and accelerations between primary shaft and wheels).

3—Definition of the Reference Trajectory for the Sliding Velocity

In order to respect these constraints, a reference trajectory is defined for the sliding velocity, by defining for each control interval k a set value for the sliding velocity. These values are such that the quality constraints are respected during and at the end of the clutch engagement phase.

Set values are therefore defined for the sliding velocity while seeing to it that it is cancelled out after time t_f. An exogenic data vector is first defined:

$$v \triangleq \begin{pmatrix} \hat{\delta} \\ N_f^* \\ X_{pedal} \end{pmatrix} \in R^2 \times R \times R \left(\triangleq \text{means «by definition»} \right)$$

where

$$\hat{\delta} \triangleq (\hat{\delta}_e, \hat{\delta}_c)$$

are the outputs of the estimators, N_f^* is the duration of the clutch closing phase (on which the standing start duration depends) in number of sampling intervals (therefore $N_f^* = t_f / \tau_s$, where τ_s is the sampling period) and X_{pedal} is the position of the accelerator pedal.

The criterion to be minimized can then be written in the equivalent parametric form:

$$\min_{U_k \in C_k^U} J_v(U_k, X_k, r_k, \dots)$$

N_f^* is a parameter to be optimized to respect the constraints on the actuators. This time is linked with a trajectory of sliding velocity ω_{sl} as detailed hereafter.

Trajectories (set of values fixed for each control interval k) ω_{sl}^{ref} of the sliding velocity are associated with the position of the accelerator pedal. This association having a direct interactivity (transparent) with the driver's wishes (during standing start). The pressure on the accelerator pedal is translated into a clutch engagement time t_f and this time is used to define a sliding velocity set value ω_{sl}^{ref} of that allows providing the transmission stability condition:

$$X_{Pedal} \mapsto t_f = f(X_{Pedal}) \mapsto \omega_{sl}^{ref} = h(t, t_f)$$

Thus, according to the invention, the expression of the reference trajectory at the time k, along prediction horizon N, is as follows:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N_f^*}{(1 + \lambda \cdot i/N_f^*)^2} \omega_{sl}^k$$

At the time $i = N_f^*$, the sliding velocity will be zero, also respecting the first constraint on the transmission stability. Parameter λ is selected to be large enough for the engagement to take place with a sliding velocity derivative very close to zero. The second constraint on the sliding velocity is thus also respected. In practice, □□□ is sufficient.

While the sliding velocity follows the aforementioned trajectory, the engine speed has to be servo-controlled by a set value ω_{sl}^{ref} that has to be as small as possible in order to limit the consumption, but has to guarantee at the same time that the engine can produce the torque required by the driver, from the pedal torque mapping (depending on the pedal torque and on the position of the accelerator pedal), and limited by the maximum torque available at a given engine speed. This reference trajectory can be expressed as follows:

$$\omega_e^{ref} = \max \{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value,

T is the maximum torque depending on speed ω_e;

T_e^d is the torque required by the driver through the pedal torque map.

The above equation simply indicates that the engine speed has to be regulated at ω_e^0 in case of a low torque request, that is a torque that can be produced by the engine at speed ω_e^0. On the other hand, when the torque request is such that it can no longer be produced at engine speed ω_e^0, set value ω_e^{ref} has to be increased to a level compatible with the request.

4—Calculation of the Optimum Control Trajectories by CMPC Control

A set of trajectories for the sliding velocity is calculated by a constrained model predictive control law referred to as CMPC control.

A quadratic optimization criterion J(•) depending on the state of the powertrain X_k, the controls U_k, the set values r_k, the constraints on the actuators C_k^U and the driving quality constraints C_k^G is defined:

$$\begin{cases} \min J(U_k, X_k, r_k, \dots) \\ \textit{UNDER THECONSTRAINTS} \\ \quad U_k \in C_k^U \\ g(X_k, U_k, \dots) \in C_k^G \end{cases}$$

The purpose of the control system according to the invention is the control of sliding velocity ω_{sl} and engine speed ω_e, respectively, by set values ω_{sl}^{ref} and ω_e^{ref} defined above. This objective can be formulated in quadratic form using the initial definition of the parametric criterion associated with the driving quality:

$$J_{v(k)}(U(k), \omega(k)) \triangleq \sum_{i=1}^{N} \left\| \begin{bmatrix} \omega_{sl}(k+i) - \omega_{sl}^{ref}(k+i) \\ \omega_e(k+i) - \omega_e^{ref}(k+i) \end{bmatrix} \right\|_Q^2$$

$$U(k) = [u^T(k), \dots, u^T(k+N-1)]^T \in C^U(k) \subset R^{2N}$$

and N is the prediction horizon.

Predictions $\omega_{sl}(k+\bullet)$ and $\omega_e(k+\bullet)$ are based on the initial state $\omega(k)$, the sequence of the future controls U and the exogenic data vector v. This prediction dynamics is obtained from the control model

$$J_e \dot\omega_e = T_e^{SP} - T_c^{SP} + \delta_e(\bullet)$$

$$[J_c + J_{eq}(i_g, i_d)]\dot\omega_c = \text{sign}(\omega_{sl}^{ec})T_c^{SP} - \delta_c(\bullet)$$

by introducing the sliding velocity and the engine speed as shown below:

$$\dot\omega_{sl}(N_f^*, X_{Pedal}) = J_e^{-1}(u_1 - u_2 + \hat\delta_e - J_{cw}^{-1}(i_g i_d)^{-1} \lfloor \text{sign}(\omega_{sl})u_2 - \hat\delta_c \rfloor$$

$$\dot\omega_e = J_e^{-1}(u_1 - u_2 + \hat\delta_e$$

where $u_1 = T_e^{SP}$ and $u_2 = T^{SP}$.

By writing the prediction on a horizon N_p in a compact manner, the result is:

$$A_p(J_e, J_{cw}(i_g, i_d)) \cdot U = S_p(\omega(k+i), \hat\delta(k+i), N_f^*, X_{Pedal})$$

with $H \triangleq A_p^T A_p \in R^{2N_p \times 2N_p}$, $W \triangleq S_p^T A_p \in R^{1 \times 2N_p}$.

Matrices A_p and S_p can be increased to take into account other specifications for the system to be controlled. It is for example possible to include an equality constraint on the engine torque desired at the clutch closing time t_f to have it equal to the pedal torque required by the driver: $T_e(t_f) = T_e^d(t_f)$.

For each selection of the length of prediction horizon N, the optimum control that minimizes the non-constrained quadratic criterion is:

$$\left\{ \hat U_N \triangleq -H^{-1} \cdot W(N, \hat\delta, X_{pedal}, \omega_{sl}^k, \omega_e^k) \right\}_{N \in \{N_f, \dots\}}$$

5—Determination of the Optimum Control Trajectory Respecting all the Constraints

Among all the control trajectories obtained in the previous stage, the one respecting the constraints on the actuators is selected by means of a dichotomic search technique.

For various values of N, the optimum control is calculated starting from a nominal value $N_f = t_f(X_{Pedal})/\tau_s$ (τ_s control sam-

pling period) and using dichotomic search to find a sliding velocity trajectory that respects the control saturation constraints and the constraint of conservation of the sign of ω_{s1}: sign $(\omega_{sl}(k)) \geq \omega 0$. This sign conservation constraint allows keeping an affine control prediction model and thus to obtain an analytical solution.

Dichotomic search is possible considering that the high values of N allow going slower, with a lower torque request. This parametrization allows ω_{sl} to converge towards zero without a sign change.

FIG. 4 illustrates the principle of dichotomic search of the predictions of the sliding velocity respecting the constraints. The sliding velocity reference is as defined before:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N_f^*}{(1 + \lambda \cdot i/N_f^*)^2} \omega_{sl}^{ref}(k)$$

In FIG. 4, the duration of the clutch sliding phase, required by the clutch pedal, is assumed to be 1 s, which gives, with a 50-ms sampling period, a value of N_f^* equal to 21 samples (Samp) denoted by N_{max} in FIG. 4. The dichotomic search is carried out on a fixed sliding prediction horizon of 5 samples in this example. In this horizon, a set of trajectories parametrized by an integer NN that is varied between a minimum value N that identifies the trajectory that an engagement time equal to N_f^* would eventually give and a very large maximum value, that will definitely meet all the constraints, is calculated. In this set of trajectories, it is found, by dichotomic search which is the one corresponding to the smallest value NN respecting the constraints.

Thus, by defining a set value for the sliding velocity, as a function of the clutch engagement time N_f^*, the prediction model is control-affine, since ω_{sl}^{ref} does not change its sign along a prediction (and, more generally, along any standing start). It is therefore possible to convert the initial criterion

$$J_{v(k)}(U(k), \omega(k)) \triangleq \sum_{i=1}^{N} \left\| \begin{bmatrix} \omega_{sl}(k+i) - \omega_{sl}^{ref}(k+i) \\ \omega_e(k+i) - \omega_e^{ref}(k+i) \end{bmatrix} \right\|_Q^2$$

to be solved under constraint to an equivalent criterion

$$J_{v(k)}(U, \dots) = U^T \cdot H(J_e, J_{cw}(i_g, i_d)) \cdot U + 2\lfloor W(\hat\delta, N_f^*, X_{Pedal}, \omega^k) \rfloor \cdot U$$

that is a function of parameter N_f^*.

Now, the solution is obtained analytically (therefore through a calculation that poses no real-time problem):

$$\left\{ \hat U_N \triangleq -H^{-1} \cdot W(N, \hat\delta, X_{pedal}, \omega_{sl}^k, \omega_e^k) \right\}_{N \in \{N_f, \dots\}}$$

Parameter N_f^* then just has to be varied to seek the solution with N_f^* being minimal to bring controls U_n into the constraint space (allowable solution with N_f^* minimal). If this search is performed by dichotomy, the cost in terms of calculation remains limited and therefore compatible with real time (the driving quality constraints are respected thanks to the "shape" of the sliding velocity trajectory).

The invention thus allows real-time control while using CMPC control.

The invention claimed is:

1. A method of controlling an automobile clutch in an automated transmission system within a vehicle comprising an engine connected to a primary shaft via the clutch, comprising:

translating a driver's request on an accelerator pedal in terms of sliding velocity ω_{sl}, defined by a difference between engine speed ω_e and primary shaft speed ω_c;

defining constraints on engine and clutch actuators regarding operating limits;

defining driving quality constraints to guarantee comfort during a clutch engagement phase;

respecting the driving quality constraints by defining a reference trajectory for the sliding velocity by defining, for control intervals, a set value $\omega_{sl}{}^{ref}$ for the sliding velocity as a function of a clutch engagement time N^*_f;

from an expression of the reference trajectory, defining an analytical expression allowing real-time calculation of a set of control trajectories for the sliding velocity, by a constrained model predictive control law;

selecting, with a dichotomic search from the set of the control trajectories for the sliding velocity, a control trajectory that respects constraints on the actuators; and

controlling the clutch with the selected control trajectory for the sliding velocity.

2. A method as claimed in claim 1, wherein the constraints on the actuators are defined by imposing minimum and maximum values on engine torque and on clutch torque and minimum and maximum values on derivatives of the engine torque and the clutch torques.

3. A method as claimed in claim 1, wherein the driving quality constraints are defined by requiring a zero sliding velocity and zero derivative during a clutch engagement phase.

4. A method as claimed in claim 2, wherein the driving quality constraints are defined by requiring a zero sliding velocity and a zero sliding velocity derivative during a clutch engagement phase.

5. A method as claimed in claim 1, wherein set value $\omega_{sl}{}^{ref}$ is defined for each control interval k by:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N^*_f}{(1 + \lambda \cdot i/N^*_f)^2} \omega_{sl}^k$$

wherein N_p is an integer defining a prediction horizon and λ is a parameter to be calibrated.

6. A method as claimed in claim 2, wherein set value $\omega_{sl}{}^{ref}$ is defined for each control interval k by:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N^*_f}{(1 + \lambda \cdot i/N^*_f)^2} \omega_{sl}^k$$

wherein N_p is an integer defining a prediction horizon and λ is a parameter to be calibrated.

7. A method as claimed in claim 3, wherein set value $\omega_{sl}{}^{ref}$ is defined for each control interval k by:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N^*_f}{(1 + \lambda \cdot i/N^*_f)^2} \omega_{sl}^k$$

wherein N_p is an integer defining a prediction horizon and λ is a parameter to be calibrated.

8. A method as claimed in claim 4, wherein set value $\omega_{sl}{}^{ref}$ is defined for each control interval k by:

$$\forall i \in \{1, \dots, N_p - 1\}$$

$$\omega_{sl}^{ref}(k+i) = \frac{1 - i/N^*_f}{(1 + \lambda \cdot i/N^*_f)^2} \omega_{sl}^k$$

wherein N_p is an integer defining a prediction horizon and λ is a parameter to be calibrated.

9. A method as claimed in claim 5, wherein λ is selected so that engagement occurs with a quasi-zero sliding velocity derivative.

10. A method as claimed in claim 6, wherein λ is selected so that engagement occurs with a quasi-zero sliding velocity derivative.

11. A method as claimed in claim 7, wherein λ is selected so that engagement occurs with a quasi-zero sliding velocity derivative.

12. A method as claimed in claim 8, wherein λ is selected so that engagement occurs with a quasi-zero sliding velocity derivative.

13. A method as claimed in claim 1, wherein a second set value $\omega_{sl}{}^{ref}$ is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max \{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

14. A method as claimed in claim 2, wherein a second set value $\omega_{sl}{}^{ref}$ is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max \{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

15. A method as claimed in claim 3, wherein a second set value $\omega_{sl}{}^{ref}$ is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max \{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

16. A method as claimed in claim 4, wherein a second set value $\omega_{sl}{}^{ref}$ is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max \{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed co_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

17. A method as claimed in claim **5**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e,

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

18. A method as claimed in claim **6**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed w_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

19. A method as claimed in claim **7**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

20. A method as claimed in claim **8**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

21. A method as claimed in claim **9**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

22. A method as claimed in claim **10**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω^{ref} is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

23. A method as claimed in claim **12**, wherein a second set value ω_{sl}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

24. A method as claimed in claim **11**, wherein a second set value ω_{st}^{ref} is generated for the engine speed, defined by:

$$\omega_e^{ref} = \max\{\omega_e^0, T^{-1}(T_e^d(X_{Pedal}, \omega_e))\}$$

where:

ω_e^0 is a constant idle set value;

T is a maximum torque depending on speed ω_e;

T_e^d is an engine torque required by a driver; and

X_{Pedal} is a position of the accelerator pedal.

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