# A Dry Clutch Engagement Controller with Thermal Effects Compensation

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***Abstract*—Single and dual dry clutches are widely used in automated manual transmissions because of their reliability and efficiency, low fuel consumption, reduction of pollutant emissions. These goals are achieved by adopting a "good" clutch model and by designing a robust control strategy. For this purpose, the clutch torque estimation is a key issue as it can be used to implement a feedback action in order to compensate for the temperature effects and model uncertainties. In this paper a transmissibility model of a dry clutch which includes the thermal effects is proposed and used for the design of a corresponding closed-loop control strategy. The control action consists of two contributions, one based on the inversion of the torque characteristic and another derived from an estimate of the clutch torque through an observer. A start-up engagement maneuver in a medium size passenger car is considered to validate the controller and its effectiveness is confirmed by numerical results.**

## I. Introduction

Dry clutches are widely used, both in single and dual configurations, for many different automotive driveline architectures which include an automation of the engagement process [1]. Indeed, the use of a dry clutch combined with intelligent actuators (both for clutches and gearbox), which manages the shifting process, allows one to obtain efficient Automated Manual Transmissions (AMTs).

The main advantages of dry clutch transmissions with respect to the other type of transmissions are: fuel savings, pollutant reduction, fast response coupled to high comfort, high efficiency and reliability, low production and maintenance costs. Unfortunately, the clutch torque transmissibility is highly influenced by the temperature rise due to the heat generation caused by frictional losses [2, 3]. In the literature specific experiments were carried out in order to characterize the friction coefficient when temperature and contact pressure vary [4, 5, 6]. Thermal behavior in a clutch system is a complex phenomenon, the temperature distribution on the friction pads is typically not uniform [7] and the temperature field is strongly affected by the contact pressure distribution [8]. Numerical finite element methods have been considered in the literature in order to take into account nonlinear behaviors from a geometrical point of view, to select different materials, and to analyze the corresponding time-spatial distributions [9, 10].

The use of lumped thermal models represents a good compromise between the complexity of the phenomenon and the computational load available for the control algorithms to be implemented in commercial control units. The influence of the temperature must be then related to the variables which can be directly measured or actuated. For instance, the torque transmitted by a dry clutch is directly influenced by the cushion spring load-deflection and it happens that at high temperatures there is an axial thermal expansion of the clutch components which produces a reduction of the total actuator stroke and consequently an increase of the clutch normal force for a given actuator position [11, 12, 13]. In [14] a numerical thermal model has been validated with experimental results showing that the temperature affects the cushion spring characteristic in two ways: firstly, the thermal load induces a thermal expansion that results in an axial size increase and consequently in a change of the position of the actuator corresponding to the torque transmission start. Secondly, by increasing the temperature, the material stiffness changes and this results in local modifications of the load-deflection characteristic slope. Those two effects together lead to model uncertainties during the engagement phase: the actuator position for which a nonzero torque is transmitted, i.e., the so-called kiss point (see Sec. II), presents large variations. This issue must be properly taken into account in order to avoid an unexpected correspondence between the throwout bearing position and the actual torque transmitted by the clutch. Indeed, the thermal expansion of the clutch components produces a reduction of the total actuator stroke and consequently an increase of the clutch normal force for a given actuator position, see [15]. Simultaneous influence of temperature, slip speed and contact pressure on the friction coefficient have been taken into account in [16] and discussed by using experimental data in [17].

Many classical and advanced control strategies for dry clutches have been proposed in the literature, e.g., see [18, 19, 20, 21] and the references therein. On the other hand not many authors have analyzed the thermal effects on the engagement performance, which is a key aspect to be considered [17, 22]. In this paper we propose a torque transmissibility model which includes the thermal effects through the indication of the explicit dependences of the different variables. This model, together with a torque estimator, represent the two main elements of the proposed engagement control scheme which address the temperature influence issue.

The remaining part of the paper is organized as follows: in Sec. II a temperature-dependent model of the torque transmission is proposed together with a corresponding lumped thermal model. An estimator for the transmitted torque is described in Sec. III and the proposed control strategy is presented and validated through simulation results in Sec. IV. Finally, concluding remarks are presented in Sec. V.

## II. Temperature-dependent transmissibility

Dry clutches exhibit significant torque capacity variations. Thus, the estimation of the clutch torque as a function of the actuation variable is essential for a precise clutch engagement control. The normal force provided by the diaphragm spring which axially presses the disc pads against cushion spring reaction, say $F_{fc}$, is a direct input for the transmitted torque model. In particular, it can be considered as a function of the throwout bearing position $x_{to}$. Temperature effects in a dry clutch can refer to changes in friction characteristics and axial thermal expansion phenomena. The first one of these effects influences the friction coefficient $\mu$ while the second influences the normal force $F_{fc}$. By including the temperature effects in the dry clutch transmissibility map, the following expression for a temperature-dependent torque characteristic can be written

$$T_{fc}(x_{to}, \omega_{fc}, \theta_c) = 2R_\mu(\omega_{fc}, \theta_c)F_{fc}(\xi_{to}(x_{to}, \theta_c)), \quad (1)$$

where $\omega_{fc} = \omega_f - \omega_c$ is the slip speed given by the difference between the flywheel speed $\omega_f$ and the clutch disc speed $\omega_c$, $\theta_c$ is an equivalent clutch temperature and the factor 2 is due to the number of friction surfaces. The variable $\xi_{to}$ is a normalized throwout position defined as

$$\xi_{to}(x_{to}, \theta_c) = \mathrm{sat}_0^1 \left\{ \frac{x_{to} - x_{to}^{cnt}(\theta_c)}{x_{to}^{cls}(\theta_c) - x_{to}^{cnt}(\theta_c)} \right\}, \quad (2)$$

where $\mathrm{sat}_0^1\{\cdot\}$ is the saturation function between 0 and 1, $x_{to}^{cnt}$, called *kiss point* or incipient sliding point, is the actuator position when the clutch disc comes into contact with the flywheel on one side and with the pressure plate on the other, $x_{to}^{cls}$ is the actuator position when the clutch is completely closed. In particular, the dependencies $x_{to}^{cnt}(\theta_c)$ and $x_{to}^{cls}(\theta_c)$ can be represented with suitable polynomial functions approximating corresponding experimental data. In the torque mode (1), the function $R_\mu$ is the generalization of the so called *equivalent friction radius* proposed in [23], that takes into account the friction phenomena and the specific clutch geometry:

$$R_\mu(\omega_{fc}, \theta_c) = \frac{1}{R_2 - R_1} \int_{R_1}^{R_2} \mu(\rho\omega_{fc}, \theta_c)\rho \, \mathrm{d}\rho, \quad (3)$$

where $R_1$ and $R_2$ are the inner and outer radii of the pads, respectively, $\rho$ is the radius variable, and $\mu(\rho\omega_{fc}, \theta_c)$ is the *friction function* that depends on the tangential velocity $\rho\omega_{fc}$ [18] and on the clutch temperature $\theta_c$. We consider a model of the friction function constituted by the combination of exponential and linear dependencies on $\omega_{fc}$ as

$$\mu(\omega_{fc}, \theta_c) = \mu_d(\theta_c) + (\mu_0 - \mu_d(\theta_c) + \alpha_1\omega_{fc}) \exp(-\alpha_2\omega_{fc}) \quad (4)$$

where the dynamic friction $\mu_d(\theta_c)$ is approximated by a second order polynomial [6, 17, 24].

The map (1) for a driveline of a typical medium size passenger car is shown in Fig. 1 for different temperatures and in Fig. 2 for different slip speeds. For the calculation of the equivalent radius we considered (3)–(4) with $R_1 = 74\,\mathrm{mm}$, $R_2 = 108\,\mathrm{mm}$, $\mu_0 = 0.15$, $\alpha_1 = 2 \cdot 10^{-2}\,\mathrm{s/rad}$, $\alpha_2 = 0.3\,\mathrm{s/rad}$. The interpolating polynomial for the temperature dependence of the dynamic friction coefficient are 0.0024 for the linear term and 0.1834 for the quadratic term. The actuator parameters are $x_{to}^{cnt} = 4.02\,\mathrm{mm}$, $x_{to}^{cls} = 7.78\,\mathrm{mm}$. The coefficients of the polynomial function $F_{fc}(\xi_{to})$ are $\beta_1 = 26.1195 \cdot 10^{-3}\,\mathrm{kN}$, $\beta_2 = -1.3046\,\mathrm{kN}$ and $\beta_3 = 5.8185\,\mathrm{kN}$. The equivalent clutch temperature variable $\theta_c$ can be obtained by considering a dynamic model of the clutch temperature behavior. To this aim we introduce the following assumptions: inside the clutch housing there is a uniform temperature distribution and the clutch and pressure plate/flywheel temperatures are the same; when the clutch is engaged there is an equal thermal exchange between the clutch disc and the left and right pressure plates; when the clutch is open it happens only convective heat exchange

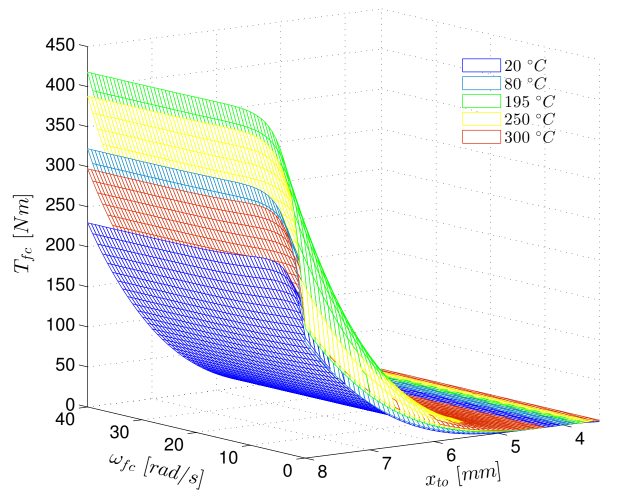


Figure 1: Torque map $T_{fc}$ versus the throwout position $x_{to}$ and the slip speed $\omega_{fc}$ for different temperatures $\theta_c$.

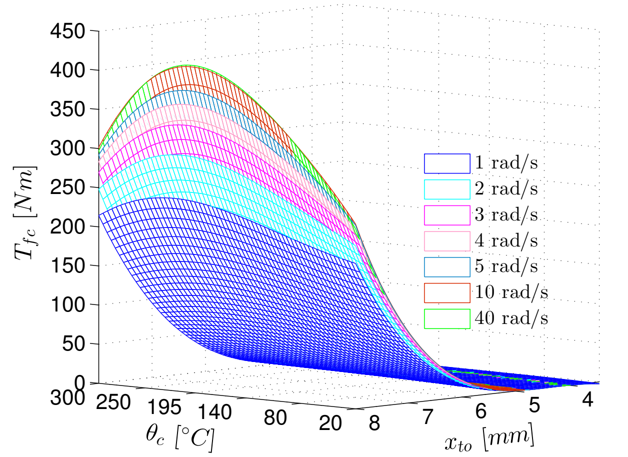


Figure 2: Torque map $T_{fc}$ versus the throwout position $x_{to}$ and the temperature $\theta_c$ for different slip speeds $\omega_{fc}$.

with the room air; when the clutch is closed it happens only conductive heat exchange between the clutch disc and the cushion spring; all slip friction work is converted into frictional heat. Under these assumptions, by considering a dual dry clutch system and by introducing the equivalent thermal capacitances for the two clutches, say $C_1$ and $C_2$, and for the central disc or "body", say $C_b$, a possible thermal model can be written as

$$\begin{aligned} C_1\dot{\theta}_{c1} =& \gamma_1\delta_1(\theta_b-\theta_{c1})+\gamma_2(1-\delta_1)(\theta_h-\theta_{c1})+ \\ &+\gamma_3 T_{fc1}\omega_{fc1} \end{aligned} \tag{5a}$$

$$\begin{aligned} C_2\dot{\theta}_{c2} =& \gamma_1\delta_2(\theta_b-\theta_{c2})+\gamma_2(1-\delta_2)(\theta_h-\theta_{c2})+ \\ &+\gamma_3 T_{fc2}\omega_{fc2} \end{aligned} \tag{5b}$$

$$\begin{aligned} C_b\dot{\theta}_b =& -\gamma_1\delta_1(\theta_b-\theta_{c1})-\gamma_1\delta_2(\theta_b-\theta_{c2})+ \\ &+\gamma_2(2-\delta_1-\delta_2)(\theta_h-\theta_b)+ \\ &+\gamma_3(T_{fc1}\omega_{fc1}+T_{fc2}\omega_{fc2}), \end{aligned} \tag{5c}$$

where $\theta$ represent the different temperatures: $\theta_b$ the pressure-plate and flywheel temperature, $\theta_{c1}$ and $\theta_{c2}$ the two clutches temperatures, $\theta_h$ the room air temperature. The logic variables $\delta_1$ and $\delta_2$ represent the state of the corresponding clutches with $\delta_1 = 1$ ($\delta_2 = 1$) if the first (second) clutch is not open. The temperature of the room air acts as an input of the system and it is assumed to be directly measurable. The constants $\gamma_1$, $\gamma_2$ and $\gamma_3$ can be obtained experimentally or through dedicated finite element thermal simulations.

The model (5) can be easily adapted to the case of a single clutch based automated transmission or AMT [18]. In particular, one can assume $\theta_c = \theta_{c1}$, $T_{fc} \equiv T_{fc1}$ and $\omega_{fc} \equiv \omega_{fc1}$. Therefore, by considering only (5a) and (5c) with $\delta_2 = 1$, $\theta_{c2} = \theta_b$, $\omega_{fc2} = 0$, during single clutch engagement ($\delta_1 = 1$) one obtains

$$C_1\dot{\theta}_c = \gamma_1(\theta_b-\theta_c)+\gamma_3 T_{fc}\omega_{fc} \tag{6a}$$

$$C_b\dot{\theta}_b = -\gamma_1(\theta_b-\theta_c)+\gamma_2(\theta_h-\theta_b)+\gamma_3 T_{fc}\omega_{fc}. \tag{6b}$$

The thermal model (6) will be used in the proposed engagement control scheme in order to evaluate the clutch temperature.

## III. Transmitted torque estimation

The accuracy of the clutch torque control is highly influenced by torque transmission characteristic variations. Due to the difficulties of an on-board torque measurement, a torque estimator can be adopted and the engagement controller is adjusted accordingly. To this aim a prerequisite is a proper driveline model oriented to the clutch control. A model of a transmission equipped with a single dry clutch can be obtained by considering kinematics and dynamics of the different driveline elements [25]. By introducing the rigidity assumptions for the mainshaft, the following second order dynamic model can be written

$$J_f\dot{\omega}_f = -b_f\omega_f + T_e(\omega_f) - T_{fc}(x_{t_0},\omega_{fc},\theta_c) \tag{7a}$$

$$J_c(r)\dot{\omega}_c = -b_c\omega_c + T_{fc}(x_{t_0},\omega_{fc},\theta_c) - T_L(\omega_c, r) \tag{7b}$$

where $J_f$ is the flywheel inertia, $J_c$ is the equivalent vehicle inertia evaluated at the clutch, $b_f$ and $b_c$ are the damping coefficients, $r$ is the gear ratio, $T_e$ is the engine torque, $T_L$ is the load torque. For the sake of simplicity the temperature dependence in (7) has been omitted. Analogous procedures can be applied in order to obtain a driveline model for the case of dry dual clutches [26].

The driveline dynamic model (7) can be used to design a torque observers. From (7) one can write

$$\hat{T}_{fc} = -J_f\hat{\dot{\omega}}_f - b_f\omega_f + T_e(\omega_f) \tag{8a}$$

$$\hat{T}_{fc} = J_c(r)\hat{\dot{\omega}}_c - b_c\omega_c + T_L(\omega_c, r) \tag{8b}$$

where hats indicate the estimated variables. That type of simple observer suffers from noise and uncertainties in the accelerations and torques estimations [20]. Indeed, the equation (8a) is typically used when the engine speed is constant so that the torque transmitted by the clutch can be computed by the engine torque without making any derivative.

In general, the torque estimators based on the inversion of the dynamic model do not provide the required robustness features [27]. A possible solution for getting better performance consists of an observer based on the engine speed error together with a constant model for the transmitted torque. The dynamic model of this type of observers can be expressed as

$$J_f\dot{\hat{\omega}}_f = -b_f\hat{\omega}_f + T_e(\omega_f) - \hat{T}_{fc} + \ell_1(\omega_f - \hat{\omega}_f) \tag{9a}$$

$$\dot{\hat{T}}_{fc} = \ell_2(\omega_f - \hat{\omega}_f) \tag{9b}$$

where $\ell_1$ and $\ell_2$ are the observer gains to be designed.

## IV. Engagement control strategy

The engagement maneuver for a dry clutch is a challenging control problem. In automated manual transmissions the shifts are decided by the driver, but performances of the (automated) engagement process are determined by the strategies implemented in the transmission control unit. In this section, the main control objectives for an automated engagement of a dry clutch are described and the proposed control scheme, which includes the torque models discussed above, is presented. The section ends with the presentation of simulation results which demonstrate the effectiveness of the controller for the critical maneuver of vehicle start-up.

### A. Control design

The engagement control of dry clutch must be designed by taking into account several, sometimes conflicting, objectives. First of all, the clutch should engage as fast as possible. Unfortunately, this goal is usually in contrast with a smooth engagement, which is a typical desired behavior for the passengers. Moreover the engagement time is clearly lower bounded by the fastest actuator response. A fast engagement also determines high slip speeds with high friction torques and then high friction losses in spite of the shorter time interval of the slipping phase, say $[0, t_f]$. Indeed, the energy dissipated during the engagement is given by the time integral of the dissipated power resulting from the product of the transmitted torque and the slip speed and it can be computed as

$$E_d = \int_0^{t_f} T_{fc}(\tau)\omega_{fc}(\tau)\mathrm{d}\tau. \tag{10}$$

Clearly $E_d$ can be reduced having low transmitted torques when the slip is high and viceversa. By reducing $t_f$ this desired combination becomes difficult to be fulfilled.

Another desired behavior related to the clutch engagement is the minimization of driveline oscillations induced by the different system dynamics before and after lock-up. A performance index corresponding to the so-called no-lurch conditions can be obtained in terms of the acceleration driveline discontinuity at lock-up. From (7) with simple algebraic manipulations, see [28] for the details, one can write

$$\dot{\omega}_c(t_f^+) - \dot{\omega}_c(t_f^-) = \frac{J_f}{J_f + J_c(r)}\dot{\omega}_{fc}(t_f^-). \tag{11}$$

Therefore at lock-up, the smaller the slip acceleration, the smoother the engagement.

As mentioned above, the temperature is a key variable which influences the engagements. Then, whatever the control technique, one should always take into account the robustness of the controlled system with respect to the temperature variations. In this work the engagement control issue is viewed as a tracking problem with two speeds reference signals and it is approached through a decoupling multivariable feedback control strategy [28]. In addition, a feedforward action can be considered and the clutch torque required for the feedback action can be obtained as an observer output. A block diagram for the proposed engagement control strategy is shown in Fig. 3 where the clutch blocks implement the torque maps given in Fig. 2 and their inversion. All controllers of the block scheme correspond to proportional-integral transfer functions. The decoupling is performed by adding the reference clutch torque to the output of the engine controller.

### B. Simulation results

Numerical tests have been carried out in order to verify the effects of the temperature and the behavior of the proposed control scheme in different operating conditions. The values of the parameters adopted are indicated in Tab. I. The clutch engagement starts at $t = 0\,\mathrm{s}$ with an initial engine speed equal to $1000\,\mathrm{rpm}$. The speeds track the corresponding reference signals, so as shown in Fig. 4. The simulations were performed at a first stage by considering the ideal case where there is a perfect inversion of the clutch friction model. The results for these specific conditions are presented in the first column of Tab. II. The engagement time, the dissipated energy and the slip acceleration at the clutch lock-up are the performance variables during the clutch engagement and correspond to the particular reference signals chosen. A mismatch condition has been simulated by considering two different functions for the clutch friction model and the clutch characteristic inversion, respectively. Furthermore, thermal effects were considered. The performance degradation in terms of increase of the engagement duration is evident from Tab. II. Then, we introduced the feedback action on the clutch torque estimate by the observer. From the third column in Tab. II it is evident that the proposed scheme is able to compensate for thermal effects and model uncertainties thus achieving performances very close to the ideal case. Figure 5 shows the throwout bearing position $x_{to}$ with and without the feedback controller and its output $x_{to_{fb}}$, as well. The transmitted and estimated clutch torques are shown in Fig. 6. At lock-up the actuator position is rapidly forced to its maximum value (rest position) and the clutch torque is assumed to be equal to the static friction torque provided by the normal force of the diaphragm spring. Finally, Fig. 7 shows the temperatures of the clutch and of the body during the engagement.

| *Symbol* | *Value* | *Unit* | *Description* |
|---|---|---|---|
| $J_f$ | 0.189 | $\mathrm{kg\,m^2}$ | Engine and flywheel inertia |
| $J_c$ | 0.578 | | Driveline and vehicle inertia |
| $b_f$ | 0.030 | $\mathrm{N\,m\,s}$ | Engine and flywheel damping |
| $b_c$ | 0.012 | | Vehicle equivalent damping |
| $\ell_1$ | −300 | $\mathrm{N\,m\,s}$ | Torque observer parameters |
| $\ell_2$ | 4252 | $\mathrm{N\,m}$ | |
| $C_b$ | 6928 | $\mathrm{J\,K^{-1}}$ | Thermal capacity of the body |
| $C_1$ | 70 | | Thermal capacity of the clutch 1 |
| $\gamma_1$ | 14.58 | $\mathrm{W\,K^{-1}}$ | Thermal model coefficients |
| $\gamma_2$ | 25.00 | | |
| $\gamma_3$ | 0.5 | Dimensionless | |

Table I: System parameters.

| | *I* | *A* | *B* |
|---|---|---|---|
| $t_f$ [s] | 1.217 | 1.423 | 1.243 |
| $E_d$ [kJ] | 3.619 | 3.670 | 3.623 |
| $\dot{\omega}_{fc}(t_f^-)$ [$\mathrm{rad\,s^{-2}}$] | 35.761 | 12.254 | 16.578 |

Table II: Simulation results for the performance variables: ideal case with perfect matching of model and process ($I$), without feedback controller ($A$), with feedback controller ($B$).

## V. Conclusion

Thermal effects on the torque transmission for dry clutches have been considered. The proposed transmissibility model includes the influence of the temperature and the slip speed on the transmitted torque. The clutch temperature estimation is obtained through a thermal dynamic model. The inversion of the characteristic performs a feedforward control action which is combined with a torque observer. The closed loop simulation results have shown the effectiveness of the proposed control strategy in order to compensate for temperature influence on the torque characteristic.

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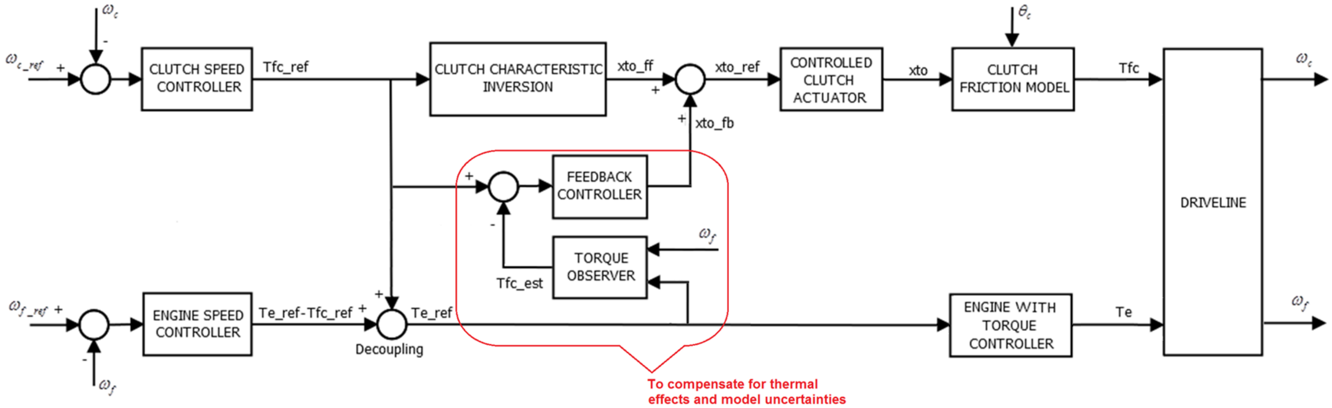


Figure 3: Clutch control scheme.

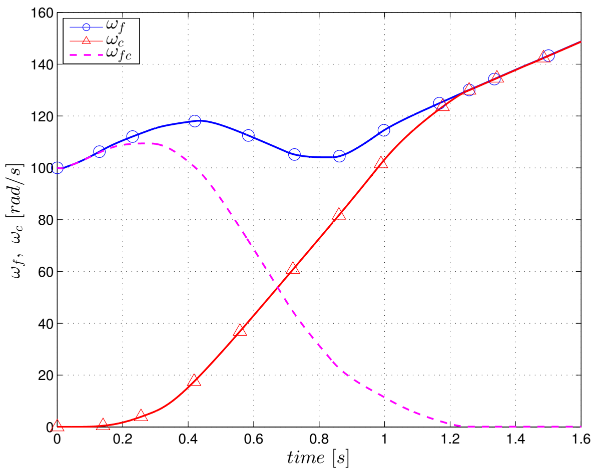


Figure 4: Time evolutions of the clutch, engine and slip speeds during a typical engagement at start-up.

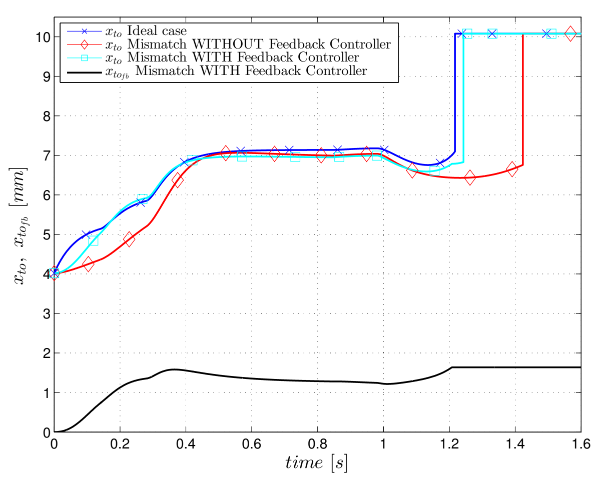


Figure 5: Time evolutions of the throwout bearing position $x_{to}$ and of the feedback component $x_{to_{fb}}$.

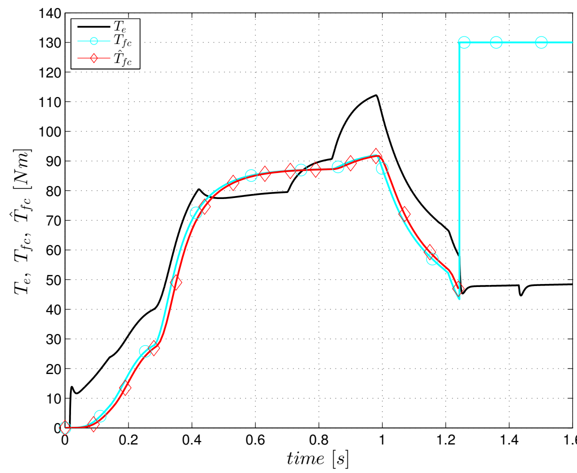


Figure 6: Time evolutions of the torque $T_e$, $T_{fc}$ and $\hat{T}_{fc}$ in the presence of thermal effects. After the clutch lock-up the clutch torques go to the static friction torque.

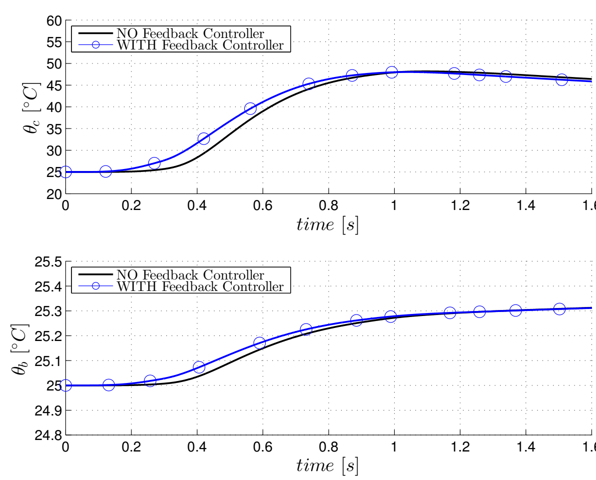


Figure 7: Temperatures $\theta_c$ and $\theta_b$ during clutch engagement at start-up. The initial temperatures are set at the ambient temperature. The temperature $\theta_b$ is almost constant during the engagement.

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