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Modeling Torque Transmissibility for Automotive Dry Clutch Engagement

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Abstract— **The dry clutch is a fundamental component in many automotive drivelines. For instance, the clutch torque transmissibility characteristic severely influences gearshift performance in automated manual transmissions. In this paper a novel model of the torque transmissibility of dry clutches is proposed. The dependence of the transmissibility characteristic on wear, slip speed and friction pads geometry and the influence of the diaphragm spring, the flat spring and the torsional damper springs is taken into account and pointed out. The sensitivity of a clutch engagement control scheme with respect to the clutch characteristic uncertainties is analyzed.**

I. INTRODUCTION

Automated engagements of dry clutches in cars and trucks are used to gradually connect the engine to the driveline avoiding undesired jerks, shocks and excessive wear [1]-[3], while in modern hybrid electric vehicles are typically used for the actuation of energy flow management [4]. Dry clutches are often used in Automated Manual Transmissions (AMTs), since they present many advantages with respect to other automatic transmissions in terms of improvement of safety, comfort, reliability, shifting quality and driving performance together with reduction of fuel consumption and pollutant emissions. Several gearshift and clutch engagement control strategies for AMTs have been proposed in literature, see among others [5]-[10]. In those contributions the torque transmitted by the clutch has been modeled as a simple stick-slip friction phenomenon by neglecting the presence of clutch components whose influence on the quality of the clutch engagement manoeuvre should be carefully evaluated. Indeed the clutch torque transmissibility characteristic is highly nonlinear [11] and robust AMTs controllers are difficult to be designed without having a good model of such characteristic [12]. This paper provides a contribution to the definition of a more detailed dry clutch torque transmission model, that includes the influence of geometry, flat spring, diaphragm spring and damper springs characteristics. The model is shown to be useful for the sensitivity analysis of AMTs controllers.

II. CLUTCH ENGAGEMENT SYSTEM AND DRIVELINE

A dry clutch engagement system (see Fig. 1) consists of a steel *clutch disk*, to which are riveted a *flat spring* and two or more *friction pads*, and a *diaphragm spring* (usually a Belleville washer spring) which transforms a *throwout bearing* position x_{to} into a corresponding position x_{pp} of the *pressure plate* (also called push plate) mounted on the diaphragm spring terminal. The pressure plate presses the clutch disk against the *flywheel* or keeps it apart. The friction between the external pads on the two sides of the clutch disk and the flywheel and pressure plate respectively, generates the torque transmitted from the engine to the driveline through the clutch, say T_{fc}. When the pressure plate (and the clutch disk) rotates constrained to the flywheel, i.e. flywheel and clutch disk have the same speed and the engine torque is less than the static friction torque, we say that *the clutch is locked–up*. In such operating conditions the engine is directly connected to the driveline. The *flat spring* (also called cushion spring), is a thin steel disk placed between the clutch friction pads and is designed with different radial stiffness in order to ensure the desired smoothness of engagement. When the flat spring is completely compressed by the pressure plate we say that *the clutch is closed*, whereas when the pressure plate position is such that the flat spring is not compressed we say that *the clutch is open*. We say that the *clutch is in the engagement phase* when is going form open to locked–up. The clutch disk is connected to a *hub* by means of *torsional dampers springs* which damp out torsional vibrations. The hub rotates at the same speed of the mainshaft. If the torsional damper springs are assumed to be rigid, that is also the clutch disk speed.

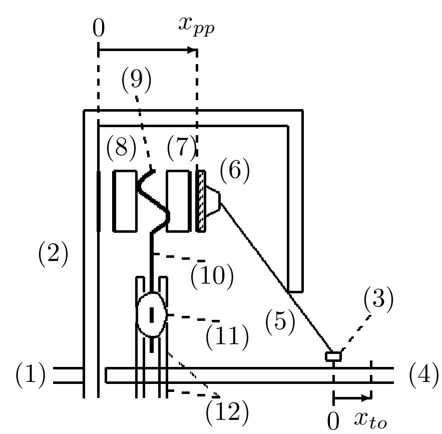

Fig. 1. A scheme of a dry clutch engagement system when the clutch is open: (1) crankshaft, (2) flywheel, (3) throwout bearing, (4) mainshaft, (5) diaphragm spring, (6) pressure plate, (7) friction pad on the pressure plate side, (8) friction pad on the flywheel side, (9) flat spring, (10) clutch disk, (11) torsional damper spring, (12) hub.

In the paper we model the dependence of the torque

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transmitted by the clutch on the throwout bearing position, i.e. the characteristic $T_{fc}(x_{to})$, and we analyze how such characteristic influences the driveline dynamics. Consider the driveline scheme shown in Fig. 2. Let us use the subscripts *e*, *f*, *c*, *h*, *g*, *w* to indicate engine, flywheel, clutch disk, hub, (primary of) gearbox and wheels, respectively. A dynamic model of the driveline can be obtained by applying the torques equilibria at the different nodes of the driveline scheme in Fig. 2. The dynamics of the speeds of the different driveline shafts can be modeled as (J are inertias, ω speeds, T torques and θ angles) [5], [9]:

$$J_e\,\dot{\omega}_e = T_e(\omega_e) - b_e\omega_e - T_{ef}(\theta_{ef},\omega_{ef}) \tag{1a}$$
$$J_f\,\dot{\omega}_f = T_{ef}(\theta_{ef},\omega_{ef}) - T_{fc}(x_{to}) \tag{1b}$$
$$J_c\,\dot{\omega}_c = T_{fc}(x_{to}) - T_{ch}(\theta_{ch}) \tag{1c}$$
$$J_h\,\dot{\omega}_h = T_{ch}(\theta_{ch}) - T_{hg}(\theta_{hg},\omega_{hg}) \tag{1d}$$
$$J_g(r)\,\dot{\omega}_g = T_{hg}(\theta_{hg},\omega_{hg}) - b_g\omega_g - \frac{1}{r}\,T_{gw}(\theta_{gw},\omega_{gw}) \tag{1e}$$
$$J_w\dot{\omega}_w = T_{gw}(\theta_{gw},\omega_{gw}) - T_w(\omega_w) \tag{1f}$$

and the angles dynamics

$$\dot{\theta}_e = \omega_e \tag{2a}$$
$$\dot{\theta}_{ef} = \omega_{ef} = \omega_e - \omega_f \tag{2b}$$
$$\dot{\theta}_{ch} = \omega_{ch} = \omega_c - \omega_h \tag{2c}$$
$$\dot{\theta}_{hg} = \omega_{hg} = \omega_h - \omega_g \tag{2d}$$
$$\dot{\theta}_{gw} = \omega_{gw} = \frac{1}{r}\,\omega_g - \omega_w \tag{2e}$$

where T_e is the engine torque (assumed to be a control input of the model), $T_{fc}(x_{to})$ is the torque transmitted by the clutch, x_{to} is the throwout bearing position (the second control input) and T_w is the equivalent load torque at the wheels (a disturbance). The gear ratio is r (which here includes also the final conversion ratio), and J_c is an equivalent inertia that includes the masses of clutch disk, friction pads and flat spring, while J_h includes the hub. Furthermore

$$J_g(r) = J_{g1} + \frac{J_{g2}}{r^2} \tag{3a}$$
$$T_{ef}(\theta_{ef},\omega_{ef}) = k_{ef}\theta_{ef} + b_{ef}\omega_{ef} \tag{3b}$$
$$T_{hg}(\theta_{hg},\omega_{hg}) = k_{hg}\theta_{hg} + b_{hg}\omega_{hg} \tag{3c}$$
$$T_{gw}(\theta_{gw},\omega_{gw}) = k_{gw}\theta_{gw} + b_{gw}\omega_{gw} \tag{3d}$$
$$T_w(\omega_w) = T_{w0} + \frac{1}{2}\,\rho_a A c_d R_w^3 \omega_w^2 \tag{3e}$$

being $k_\bullet$ elastic stiffness coefficients, $b_\bullet$ viscous damping, T_{w0} a constant load torque, ρ_a the air density, A the front surface vehicle area, c_d the air resistance coefficient, R_w the wheels radius. The damper spring characteristic $T_{ch}(\theta_{ch})$ can be typically expressed as a piecewise linear function [13]: $T_{ch}(\theta_{ch}) = k_{ch_1}\theta_{ch}$ for $\theta_{ch} \in [0, \theta_{ch_1}]$, $T_{ch}(\theta_{ch}) = k_{ch_2}\theta_{ch} + (k_{ch_1} - k_{ch_2})\theta_{ch_1}$ for $\theta_{ch} \in [\theta_{ch_1}, \theta_{ch_2}]$, $T_{ch}(\theta_{ch}) = k_{ch_3}\theta_{ch} + (k_{ch_2} - k_{ch_3})\theta_{ch_2} + (k_{ch_1} - k_{ch_2})\theta_{ch_1}$ for $\theta_{ch} \geq \theta_{ch_2}$, with $\theta_{ch_2} > \theta_{ch_1} > 0$ and $T_{ch}(\theta_{ch}) = -T_{ch}(-\theta_{ch})$ for $\theta_{ch} < 0$, i.e. $T_{ch}(\theta_{ch})$ is an odd function. When the clutch is locked–up the flywheel speed ω_f and the clutch disk speed ω_c are equal. The corresponding locked–up model can be obtained by adding (1b) and (1c) with the assumption $\omega_f = \omega_c$.

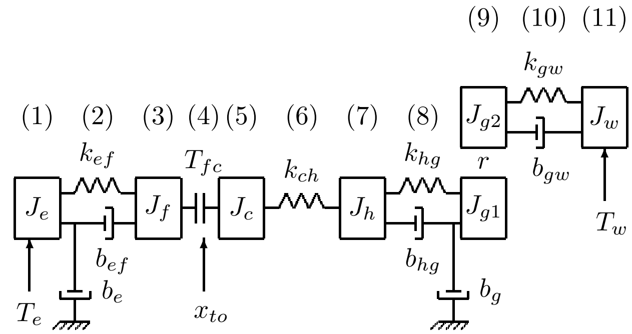

Fig. 2. A typical driveline scheme: (1) engine, (2) crankshaft, (3) flywheel, (4) equivalent representation for the dry clutch friction torque generation, (5) clutch disk, (6) damper spring, (7) hub, (8) mainshaft (hub–gearbox shaft), (9) gearbox and differential, (10) driveshaft, (11) wheels and vehicle.

It is possible to carry out a reduction of the driveline model (1)-(2), by assuming rigidity of shafts and springs: $\omega_e = \omega_f$ (rigid crankshaft), $\omega_c = \omega_h$ (rigid damper spring), $\omega_h = \omega_g$ (rigid mainshaft), $\omega_g = r\omega_w$ (rigid driveshaft). By adding (1a)-(1b) and (1c)-(1f) one obtains the reduced order model

$$(J_e + J_f)\,\dot{\omega}_f = T_e(\omega_f) - b_e\omega_f - T_{fc}(x_{to}) \tag{4a}$$
$$(J_c + J_v(r))\,\dot{\omega}_c = T_{fc}(x_{to}) - b_g\omega_c - \frac{1}{r}\,T_w\left(\frac{\omega_c}{r}\right) \tag{4b}$$

where
$$J_v(r) = J_h + J_g(r) + \frac{1}{r^2}J_w \tag{5}$$

is the vehicle inertia reported to the mainshaft. By assuming T_e and T_{fc} as control inputs, the model (4) would be simple enough for the design of AMT strategies by using classical control techniques. Unfortunately, the real control variable is the throwout position x_{to} and therefore the knowledge of the nonlinear characteristic $T_{fc}(x_{to})$ is fundamental for the controller implementation.

III. Clutch Torque Transmissibility

The torque transmitted by the clutch can be estimated by inverting the driveline dynamic model. For simplicity let us consider the reduced order model (4). By inverting (4a) one can write

$$\hat{T}_{fc}(x_{to}) = -(J_e + J_f)\,\hat{\dot{\omega}}_f - b_e\omega_f + \hat{T}_e(\omega_f). \tag{6}$$

where hats are used for variables that need to be estimated because of well known difficulties for their direct measurement in real vehicles. By inverting (4b) one can write

$$\hat{T}_{fc}(x_{to}) = (J_c + J_v(r))\,\hat{\dot{\omega}}_c + b_g\omega_c + \frac{1}{r}\hat{T}_w\left(\frac{\omega_c}{r}\right). \tag{7}$$

The model (7) can be also used when the clutch is locked–up in order to estimate the clutch efficiency, i.e the reduction of the engine torque transmitted to the driveline due to the presence of the clutch. The estimators (6) and (7) can be easily implemented on electronic control units but suffer for noise and uncertainties in the accelerations and engine torque estimations. Indeed (6) is typically used when the engine speed is constant so that the torque transmitted by the clutch can be approximated by the engine torque minus the friction losses. Due to the above mentioned limitations, the torque estimators based on the inversion of the dynamic model are used only to adapt the static nominal transmissibility characteristic which is obtained by modeling the friction phenomenon.

A. Transmissible Torque

The torque transmitted by a dry clutch is generated by the friction phenomenon between the friction pads mounted on the two sides of the clutch disk and the flywheel and pressure plate, see Fig. 1. A zoom of the clutch engagement scheme focusing on the axial flat spring displacement is depicted in Fig. 3.

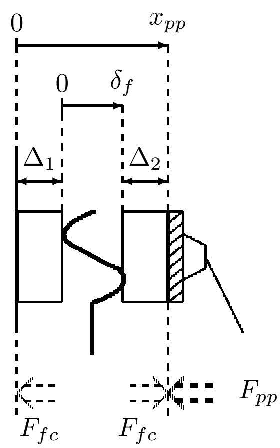

Fig. 3. A zoom of the axial displacement of the flat spring and forces on the friction surfaces: $\delta_f = \Delta_f$ when the clutch is open and $\delta_f = 0$ when the clutch is closed.

Fig. 4 shows typical characteristics of the most relevant variables: $F_{pp}(x_{to})$ is the force on the throwout bearing due the diaphragm spring reported on the pressure plate [14], $\Delta = \Delta_1 + \Delta_2$ is the total thickness of the two friction pads, $\delta_f \in [0, \Delta_f]$ is the flat spring compression, $F_{fc}(\delta_f)$ is the force applied by the flat spring on the friction surfaces (flywheel and pressure plate), x_{to}^{cnt} is the throwout bearing position for which the friction pad and the flywheel come in *contact*, i.e. the throwout position corresponding to $x_{pp} = \Delta + \Delta_f$, and x_{to}^{cls} is the smallest throwout bearing position for which the flat spring is completely compressed (*the clutch is closed*), i.e. the smallest throwout position corresponding to $x_{pp} = \Delta$ and $\delta_f = 0$.

Usually the smallest value for which the clutch is locked–up is lower than x_{to}^{cls}. In other words when the clutch is closed it should be already locked–up. In order to avoid undesired clutch unlocking when the clutch is closed, i.e. $x_{to} \in [x_{to}^{cls}, x_{to}^{max}]$ and $\delta_f = 0$, the torque produced by the engine must be lower than the maximum static friction torque

$$T_{fc}^{max}(x_{to}) = n\mu_s R_{eq} F_{pp}(x_{to}) \tag{8}$$

where n is a number of pairs of contact surfaces ($n = 2$ in our case), μ_s is the static friction coefficient, R_{eq} is the equivalent radius of the contact surface (whose expression will be detailed in the sequel). The torque T_{fc}^{max} can be interpreted as the maximum transmissible torque when the clutch is closed. Indeed if $T_e > T_{fc}^{max}(x_{to})$ the flywheel will start slipping with respect to the clutch disk. In practice such situation should not occur because, given the maximum engine torque T_e^{max}, (8) is used for sizing the diaphragm spring to be used, i.e. when the clutch is closed it must be

$$F_{pp}(x_{to}) \geq k_s \frac{T_e^{max}}{n\mu_s R_{eq}} \tag{9}$$

where k_s is a safety coefficient typically chosen between 1.2 and 1.5, depending on the vehicle type. Note that (9) should hold for any $x_{to} \in [x_{to}^{cls}, x_{to}^{max}]$. Since x_{to}^{cls} is defined by geometric constraints, the wear affects such quantity. However, wear reduces Δ which increases x_{to}^{cls} causing a corresponding increase of $F_{pp}(x_{to}^{cls})$ (see Fig. 4 and (9)).

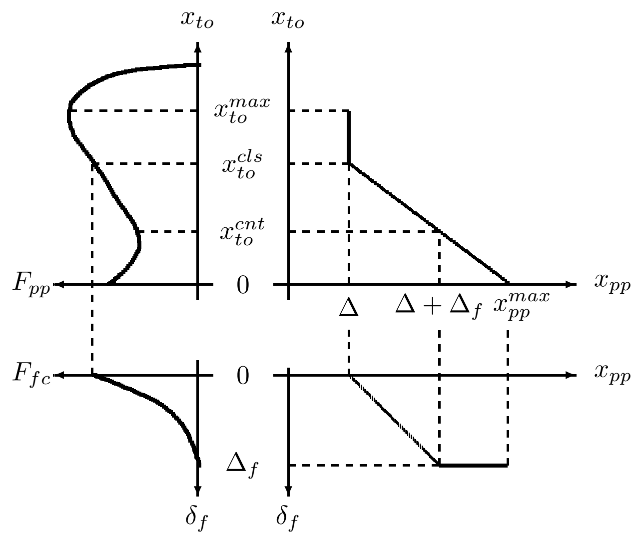

Fig. 4. Dry clutch characteristics. For a clutch engagement manoeuvre the plots should be intended by considering the following sequence of dependence: $F_{pp}(x_{to})$, $x_{pp}(x_{to})$, $\delta_f(x_{pp})$ and $F_{fc}(\delta_f)$. When the clutch is mounted on the driveline x_{to} is constained to be less than x_{to}^{max} which is the value of x_{to} corresponding to the maximum value of F_{pp}.

During the first part of a clutch engagement, since for small δ_f the stiffness of the diaphragm spring is much larger than the force $F_{fc}(\delta_f)$ applied by the flat spring on the friction surfaces, the diaphragm spring determines the pressure plate position $x_{pp}(x_{to})$. The flat spring compression δ_f is determined by x_{pp}. Therefore, due to the small mass of the flat spring, the torque transmitted by the clutch can be approximated by using an expression similar to (8):

$$T_{fc}(x_{to}) = n\mu R_{eq} F_{fc}(\delta_f(x_{pp}(x_{to}))) \tag{10}$$

where μ is the *dynamic* friction coefficient. From (10) it is clear that the flat spring compression δ_f and the corresponding force $F_{fc}(\delta_f)$ determine the torque transmissibility.

Let us describe the evolution of the main variables during a typical clutch engagement manoeuvre by considering the characteristics in Fig. 4. When x_{to} increases from 0 up to x_{to}^{cnt} the pressure plate position will decrease from x_{pp}^{max} to $\Delta + \Delta_f$. The clutch disk is moved towards the flywheel. The friction torque generated in this phase is negligible because the hub resistance for longitudinal translations on the mainshaft is very small. Then the clutch remains open ($\delta_f = \Delta_f$) and the torque transmitted by the clutch is zero, see (10) and note that for $\delta_f = \Delta_f$ it is $F_{fc}(\delta_f) = 0$. Note that x_{to}^{cnt} depends on the friction pad wear (Δ is the total thickness of the friction pads). When x_{to} becomes larger than x_{to}^{cnt} the flywheel is in contact with the friction pad, the flat spring compression starts and the torque transmitted by the clutch becomes different from zero, see (10) with $\delta_f < \Delta_f$ which makes $F_{fc}(\delta_f) \neq 0$. When x_{to} is such that $x_{pp} = \Delta$, the flat spring is completely compressed, i.e. $\delta_f = 0$, and the clutch is closed. Remind that during the engagement the clutch lock–up starts for a value $x_{to} \in [x_{to}^{cnt}, x_{to}^{cls})$, i.e for strictly positive values of δ_f. Indeed the flat and diaphragm springs are designed so that $F_{fc}(\delta_f = 0) = F_{pp}(x_{to}^{cls})$, see Fig. 4, and since (9) must be satisfied, the clutch will be locked–up before that the flat spring is completely compressed. Obviously the smallest value of x_{to} for which the clutch is locked–up can also depend on the engine torque.

B. Geometric and Pressure Based Model

The force applied by the flat spring on the friction surfaces (by assuming symmetry of the n contact surfaces and by omitting for simplicity the dependence on x_{to}) can be written as

$$F_{fc} = \int_0^{2\pi} \int_{R_1}^{R_2} \sigma(\rho, \varphi, F_{fc})\, \rho\, d\rho\, d\varphi \tag{11}$$

where ρ and φ are the radial and angular geometric variables of the friction pad surface ($\rho = 0$ at the center of the clutch disk), the parameters R_1 and R_2 are the inner and outer radii of the clutch friction pads, and σ is the pressure distribution on the friction pads. Within the above geometric framework the friction torque can be written as

$$T_{fc} = n \int_0^{2\pi} \int_{R_1}^{R_2} \tau(\rho, \varphi, F_{fc})\, \rho^2\, d\rho\, d\varphi \tag{12}$$

where τ is the distribution of tangential stress along the friction surfaces of clutch. By defining

$$R_\mu = \frac{\int_0^{2\pi} \int_{R_1}^{R_2} \tau(\rho, \varphi, F_{fc}) \rho^2 d\rho d\varphi}{\int_0^{2\pi} \int_{R_1}^{R_2} \sigma(\rho, \varphi, F_{fc}) \rho d\rho d\varphi} \tag{13}$$

and by using (11), the expression (12) can be rewritten as

$$T_{fc} = n R_\mu F_{fc}, \tag{14}$$

which has the same form of (10). An uniform distribution along the angular direction can be usually assumed, i.e. σ and τ do not depend on φ. Therefore (13) becomes

$$R_\mu = \frac{\int_{R_1}^{R_2} \tau(\rho, F_{fc}) \rho^2 d\rho}{\int_{R_1}^{R_2} \sigma(\rho, F_{fc}) \rho d\rho}. \tag{15}$$

In order to obtain an expression for T_{fc} one must now detail the tangential stress τ and the normal pressure σ. To this aim a typical assumption made in friction mechanics is

$$\tau(\rho, F_{fc}) = \mu(\rho \omega_{sl})\, \sigma(\rho, F_{fc}) \tag{16}$$

where $\mu(v)$ is the friction coefficient (or more precisely the friction function), v being the tangential velocity. Since $v = \rho\omega_{sl}$, the friction coefficient is a function of the slip speed $\omega_{sl} = \omega_f - \omega_c$ and it is different for each radius of clutch disk. Different models for the function $\mu(v)$ have been proposed in the literature [15]. The friction function is assumed to depend on the signum of the velocity, i.e. Coulomb friction, or is modeled by using a smooth approximation of the signum function in order to avoid the numerical problems due to the simulation of discontinuities in Coulomb friction at zero velocity [16]. Note that by considering $\mu(\rho\omega_{sl})$ as a signum function and ω_{sl} positive, μ will not depend on ρ thus simplifying the computation of (13).

By using (16) and μ constant in (16), (15) becomes

$$R_\mu = \mu \frac{\int_{R_1}^{R_2} \sigma(\rho, F_{fc}) \rho^2 d\rho}{\int_{R_1}^{R_2} \sigma(\rho, F_{fc}) \rho d\rho}. \tag{17}$$

The only function still to be defined is the pressure distribution σ. If σ is assumed to be constant (17) becomes

$$R_\mu = \mu \frac{2}{3} \frac{R_2^3 - R_1^3}{R_2^2 - R_1^2} \tag{18}$$

and (14) is equal to (10) with $\mu R_{eq} = R_\mu$ and R_μ given by (18) which is a classical expression used for the friction torque [15]. Under the assumption of uniform wear of pads during contacts σ will be proportional to the inverse of ρ [15] and (17) becomes

$$R_\mu = \mu \frac{1}{2}(R_1 + R_2). \tag{19}$$

In this case (14) is equal to (10) with $\mu R_{eq} = R_\mu$ and R_μ given by (19) which is another well known expression used for the friction torque. In the simulation results such characteristic will be indicated as the *nominal characteristic*.

During the closing phase of the clutch engagement, the contact between the clutch disk and the flywheel is designed to begin at the outer sector of the flywheel and to continue towards the inner area, with a consequent increase of the contact surface area, i.e. the contact occurs for $\rho \in [r_i(F_{fc}), R_2]$. If the pressure distribution is assumed to be constant the expression of R_μ will be (18) with $r_i(F_{fc})$ replacing R_1 and the torque (14) can be expressed as

$$T_{fc} = n\mu \frac{2}{3} \frac{R_2^3 - r_i^3(F_{fc})}{R_2^2 - r_i^2(F_{fc})} F_{fc}. \tag{20}$$

Remind that F_{fc} depends on the flat spring compression δ_f which is determined by the pressure plate position x_{pp} which can be varied by means of the throwout bearing movements (x_{to}) through the diaphragm spring.

If the pressure distribution is assumed to be proportional to the inverse of ρ the expression of R_μ will be (19) with

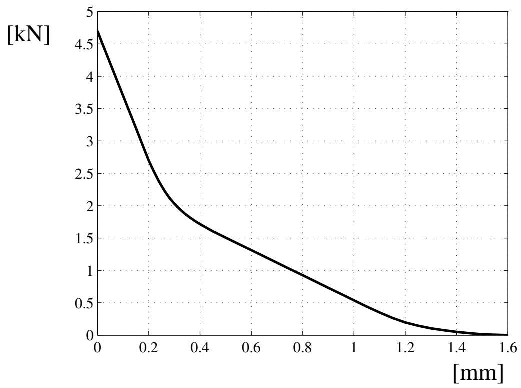

Fig. 5. Flat spring force vs. the flat spring compression: $F_{fc}(\delta_f)$.

$r_i(F_{fc})$ replacing R_1 and the torque (14) can be expressed as

$$T_{fc} = n\mu\frac{1}{2}\left(R_2 + r_i(F_{fc})\right) F_{fc}. \qquad (21)$$

Other expressions for the clutch torque characteristic can be obtained by assuming different expressions for the pressure distribution. A possible alternative expression for σ is

$$\sigma(\rho, F_{fc}) = k(F_{fc})(R_2 - \rho)(\rho - r_i(F_{fc})) \qquad (22)$$

for $\rho \in [r_i, R_2]$, and zero otherwise. For each value of F_{fc} the value $k(F_{fc})$ can be directly derived by means of the equilibrium condition (11). In particular after some algebra the expression (12) of the transmitted torque becomes

$$T_{fc} = n\mu\frac{3R_2^2 + 4R_2 r_i(F_{fc}) + 3r_i^2(F_{fc})}{5(R_2 + r_i(F_{fc}))}F_{fc}. \qquad (23)$$

By resuming, in order to obtain the torque transmitted by the clutch with the proposed model one has to fix the pressure distribution and the internal contact surface radius dependence on the flat spring force, and the flat spring force as a function of the flat spring compression. All such information depend of course on the specific clutch set–up.

IV. AMT Controller Sensitivity

In this section we present a sensitivity analysis of an AMT control scheme in the presence of uncertainties in the dry clutch torque transmissibility characteristic. The driveline model parameters are derived from [5] and [9]; the others are shown in Table I. The characteristic $F_{fc}(\delta_f)$ which is depicted in Fig. 5 has been obtained from specific laboratory experimental tests carried out using a real clutch disk. A block scheme of the simulated closed loop system is shown in Fig. 6. Two decoupled feedback loops with PI controllers are designed by using the reduced order model (4). The output of the controller on ω_e is added to T_{fc}^{ref} in order to obtain the reference engine torque T_e^{ref}, see [5] for details. The reference transmission torque T_{fc}^{ref} is transformed into a reference throwout bearing position x_{to}^{ref} by inverting the nominal torque transmission characteristic, i.e. by computing F_{fc} from (14) with R_μ given by (19) and by inverting the characteristics $F_{fc}(\delta_f)$, $\delta_f(x_{pp})$ and $x_{pp}(x_{to})$, see Fig. 4 and

TABLE I

Clutch parameters.

μ_s, μ	static (dynamic) friction coefficients	0.27	-
R_1	min radius of clutch disk	0.074	m
R_2	max radius of clutch disk	0.104	m
Δ	total thickness of friction pads	6.0	mm
Δ_f	flat spring total expansion	1.6	mm
x_{pp}^{max}	maximum pressure plate position	8.1	mm
x_{to}^{cls}	throwout position at clutch closing	8.4	mm
x_{to}^{max}	maximum throwout position	10.08	mm

Fig. 5. The reference signal x_{to}^{ref} is actuated by means of a controlled actuator whose closed loop model is represented with a unitary gain first-order transfer function $A(s)$ with a time constant equal to 0.1s. The output of $A(s)$ is the throwout bearing position x_{to}. The actuator position x_{to} and the engine torque (note that we assume $T_e = T_e^{ref}$) are the inputs of the dynamic model (1)-(2). The torque transmissibility characteristic $T_{fc}(x_{to})$ in the model of the driveline is chosen among the expressions (20), (21) or (23). The internal radius is assumed constant and equal to R_1, or linearly varying from $r_i(F_{fc}(\delta_f = \Delta_f)) = R_2$ to $r_i(F_{fc}(\delta_f = 0)) = R_1$, or the concave function $r_i(F_{fc}) = -\alpha(F_{fc}^3 - F_{fc}) + R_2$ with $\alpha = 2.89 \cdot 10^{-13}$, or the convex function obtained as the complement of the concave with respect to the linear function. The characteristic $F_{fc}(\delta_f)$ is not varied and is that depicted in Fig. 5.

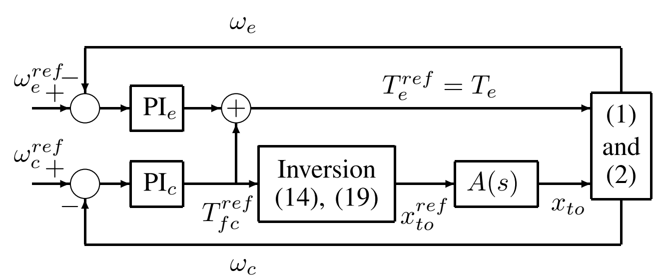

Fig. 6. A block scheme of the simulated AMT control system.

Fig. 7 shows the relative percentage error for the different transmitted torque characteristics with respect to the nominal characteristic. Four groups of curves can be clearly distinguished, each group corresponds to a given profile of the internal radius function $r_i(F_{fc})$. The differences within each group are due to the different pressure distribution functions $\sigma(\rho, F_{fc})$ (constant, proportional to the inverse of ρ and quadratic). It is clear that the main influence on the torque transmission is given by the different geometric shapes of the internal radius of the friction surface. Simulation results in the nominal case will be now compared with those corresponding to the largest error in the torque transmission characteristic, i.e. the characteristic (20) with $r_i(F_{fc})$ concave. The controllers parameters are: $K_{P_e} = 1200$, $K_{I_e} = 40$, $K_{P_c} = 30$ and $K_{I_c} = 60$. Dynamic simulations of the critical engagement manoeuvre at vehicle start–up have been considered. Fig. 8 shows the flywheel and

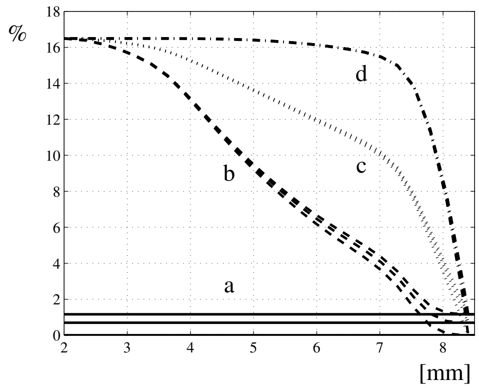

Fig. 7. Torque transmission relative percentage error of (20), (21) and (23) with respect to the nominal characteristic vs. throwout bearing position x_{to}: (a) $r_i = R_1$ (continuous), (b) $r_i(F_{fc})$ convex (dashed), (c) r_i linear (dotted), (d) r_i concave (dash—dotted).

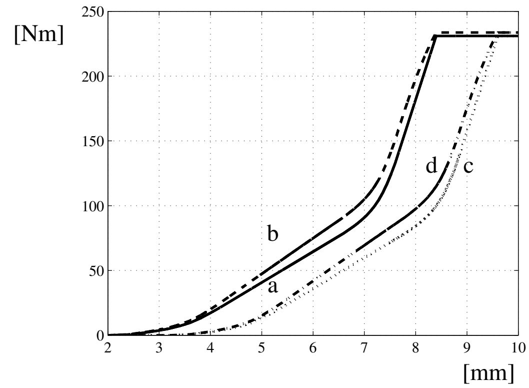

Fig. 9. Torque transmitted T_{fc} vs. throwout bearing position x_{to}: (a, c) nominal characteristic, (b, d) characteristic (20) with $r_i(F_{fc})$ concave, (a, b) no wear, (c, d) in the presence of wear, i.e. $\Delta = 5.7$mm.

clutch speeds and the torques profiles for nominal operating conditions. The transmitted torque vs. the throwout bearing position characteristics for dynamic simulations are shown in Fig. 9. The transmitted torque time evolutions are quite different even though the tracking of the speed profiles remain almost unchanged. Therefore different torques will determine different dissipated energy during the engagement and then also different temperature variations.

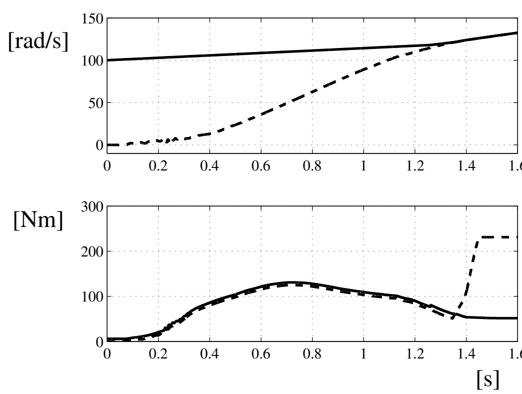

Fig. 8. Speeds (ω_f continuous, ω_c dashed) and torques (T_e continuous, T_{fc} dashed) time evolutions at vehicle start–up.

V. Conclusion

A novel model of dry clutch torque transmissibility has been proposed. The model is obtained by analyzing the friction phenomena that generate the transmitted torque and by clarifying the influence of the diaphragm and flat springs. The dependence of the transmitted torque on contact friction surfaces and on pressure distribution (uniform pressure or uniform wear) has been pointed out. The sensitivity analysis on an AMT control scheme has shown that uncertainties in clutch torque characteristic can severely affect the performance of the clutch engagement: modeling in detail the torque transmitted by the clutch is a fundamental issue in order to design robust engagement control strategies.

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