Torque Transmissibility Assessment for Automotive Dry Clutch Engagement

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***Abstract*—Dry clutches are widely used in conventional and innovative automotive drivelines and represent a key element for automated manual transmissions. In practical applications it is fundamental to model the clutch behavior through its torque transmissibility characteristic, i.e. the relationship between the throwout bearing position (or the pressure applied by the clutch actuator) and the torque transmitted through the clutch during the engagement phase. In this paper a new model for the torque transmissibility of dry clutches is proposed. It is analyzed how the transmissibility characteristic depends on: friction pads geometry, cushion spring compression, cushion spring load, slip speed dependent friction. Corresponding functions are suitably composed determining the torque transmissibility expression. An experimental procedure for tuning the characteristic parameters is presented. The clutch torque transmissibility model is tested on a detailed co-simulation model with a typical automated manual transmission controller.**

I. Introduction

The automation of dry clutch engagements in automotive transmissions represent an enabling process for several vehicle optimization issues, e.g. fuel consumption and gearshift efficiency [1], vehicle dynamics [2], drivelines architectures reconfigurations [3]-[4]. Automated manual transmission (AMT) is an interesting example where the control of dry clutches shows its importance and potentialities [5], also in the case of dual-clutch transmissions [6]. Typically, AMTs present many advantages with respect to manual transmissions in terms of improvement of safety, comfort, reliability, shifting quality and driving performance, together with reduction of fuel consumption and pollutant emissions. In order to obtain such goals several model-based engagement control strategies for dry clutches in AMTs have been recently proposed in the literature, e.g. classical controller [7], optimal control [8]-[10], predictive control [11]-[12], decoupling control [13], robust control [14]-[15]. However, effective AMTs controllers are difficult to be designed without having a "good" model of the clutch torque transmissibility characteristic. Furthermore such a model can be very useful for reducing transmission calibration effort and time [16].

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The classical clutch torque transmissibility model assumes the torque to be proportional to the normal force on the clutch plate, through the dynamic friction, the number of friction surfaces and some geometrical parameters. The key problem in such a model consists of evaluating the normal force on the clutch plate, which determines the transmitted torque. In [9] such problem is overcome by considering the force as an actuation variable; unfortunately such force is not directly manipulable in practical applications. In [17]-[18] the normal force on clutch face is assumed to be related to the pressure applied by the clutch actuator; however such dependency is difficult to be parameterized and it is strongly affected by measurement issues. A typical alternative consists of relating the normal force to the actuator (throwout bearing) position. In [5] the need for such dependency in clutch engagement control is mentioned but not analyzed in details. In [19] the clutch torque is modeled as the product of the normalized actuator position and a maximum torque exponentially dependent on the slip speed: the maximum torque expression depends on parameters to be identified, but no specific identification procedures are proposed. A more complex dependence of the clutch torque on the actuator position and slip speed is presented in [4], but it is not justified. A completely different approach for the estimation of the clutch transmitted torque, based on the inversion of a driveline dynamic model, is proposed in [13] and [20]. Unfortunately, usefulness and robustness of these estimators drastically depend on the needed availability of clutch disk acceleration, disk inertia and engine torque.

In this paper we present a new detailed model for torque transmissibility characteristics of automotive dry clutches. The main contribution of this model consists of clarifying how the different driveline components of a dry clutch system explicitly influence the transmissibility; moreover we show how the proposed model allows to define an experimental procedure for tuning the parameters of the characteristic. The analysis of functional and structural links between the clutch engagement system and other driveline components is the first fundamental step for solving the modeling issues. By describing a typical engagement process, in Section II it is shown how the diaphragm spring [21]-[22] and the cushion spring [23] take part in the transmissibility characteristic. Though the torque transmitted through a dry clutch influences and is influenced by driveline components and diaphragm spring, the torque transmissibility model proposed in this paper clearly shows that the two main elements to be considered are the cushion spring load and the dry friction phenomenon. In Section III it is shown how these two elements enter separately into the proposed torque transmissibility model. In such model, so as usual in classical friction models, the dependence of the friction on the slip speed plays a key role [24]. That is taken into account by introducing an equivalent radius dependent on a friction function of the slip speed. Dry friction vs. slip velocity is a widely studied dependence and different static friction models that seem to be suitable for the automotive dry clutch engagement process have been proposed in the literature [25]-[27]. In Section IV experimental tests done on friction material and cushion spring of a real automotive dry clutch are presented and corresponding models are fitted on the measured data. Hence, in Section V by using a detailed Matlab/Simulink and CarSim co-simulation environment, the model has been integrated into a typical automated manual transmission controller and compared with other typical clutch transmissibility models. Conclusions close the paper in Section VI.

II. Clutch Engagement System

In this section we present the main components and the operations of a typical dry clutch engagement process.

A. Components

A dry clutch engagement system (see Fig. 1) consists of a steel *clutch disk*, to which a *cushion spring* and two or more *friction pads* are riveted, 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 clutch disk is connected to a *hub*. The hub rotates at the same speed of the mainshaft, which is also the clutch disk speed (ω_c). The pressure plate presses the clutch disk against the *flywheel* or keeps it apart. The pressure plate rotates at the same speed of the flywheel (ω_f). 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, say T_{fc}. When the pressure plate (and the clutch disk) rotates constrained to the flywheel, i.e. the flywheel and the clutch disk have the same speed and the transmitted 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 and the transmitted torque is almost equal to the engine torque T_e.

The *cushion spring* (also called flat spring), is a thin steel disk placed between the clutch friction pads and is designed with different local axial stiffness depending on the radius in order to ensure the desired engagement smoothness [23]. When the cushion spring is completely compressed by the pressure plate we say that *the clutch is closed*. When the pressure plate position is such that the cushion spring is not at all compressed, i.e. there is no contact between friction disk and flywheel, we say that *the clutch is open*. We say that the *clutch is in the engagement phase* when is going from open to locked–up. Note that with the proposed terminology in ordinary engagement phases, the clutch is locked–up before being closed.

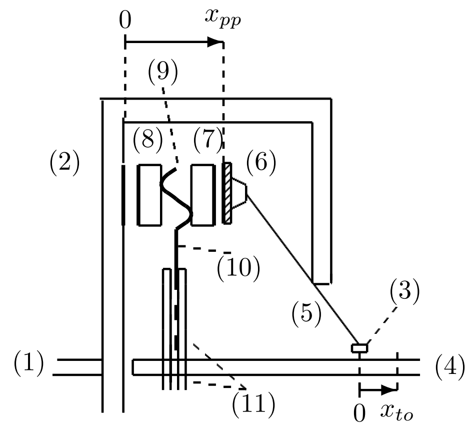

Fig. 1. A scheme of a dry clutch engagement system when the clutch is open: (1) crankshaft, (2) flywheel, (3) throwout bearing, (4) mainshaft, (5) diaphragm spring, (6) pressure plate, (7) friction pad on the pressure plate side, (8) friction pad on the flywheel side, (9) cushion spring, (10) clutch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

B. Clutch closed and locked–up

The main phenomenon that determines the torque transmitted by a dry clutch is the friction 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 cushion spring displacement is depicted in Fig. 2.

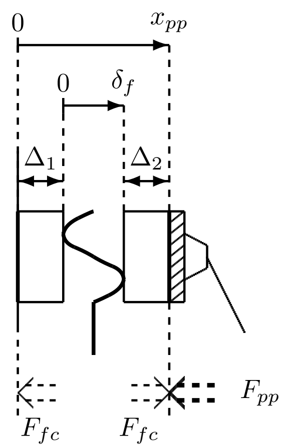

Fig. 2. A zoom of the axial displacement of the cushion spring and forces on the friction surfaces: $\delta_f = \Delta_f$ when the clutch is open, $\delta_f = 0$ when the clutch is closed, and for some $\delta_f \in (0, \Delta_f)$ the clutch will be locked-up.

Fig. 3 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 [22], $\Delta = \Delta_1 + \Delta_2$ is the total ideal thickness of the two friction pads, $\delta_f \in [0, \Delta_f]$ is the cushion spring compression (note that δ_f is set to zero in correspondence of the maximum compression of the cushion spring), $F_{fc}(\delta_f)$ is the force reaction applied by the cushion 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 $x_{pp} = \Delta + \Delta_f$ (see Fig. 1), and x_{to}^{cls} is the smallest throwout bearing position for which the cushion 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 of the throwout bearing position for which the clutch is locked–up is lower than x_{to}^{cls}. In other words when the clutch is closed ($\delta_f = 0$) 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 that it can be shown, under some simplifying assumptions, being dependent on the pressure plate force as

$$T_{fc}^{max}(x_{to}) = n\mu_s R_{eq} F_{pp}(x_{to}) \tag{1}$$

where n is the number of pairs of contact surfaces ($n = 2$ in our case), μ_s is the static friction coefficient and R_{eq} is the equivalent radius of the contact surface. The torque T_{fc}^{max} can be interpreted as the maximum transmissible torque when the clutch is closed. Indeed if the transmitted torque is larger than T_{fc}^{max} the flywheel will start slipping with respect to the clutch disk. Given the maximum engine torque T_e^{max}, (1) can be used for estimating the size of the diaphragm spring to be used. Indeed in order to avoid slipping when the clutch is closed, from (1) it must be

$$F_{pp}(x_{to}) \geq k_s \frac{T_e^{max}}{n\mu_s R_{eq}} \tag{2}$$

where k_s is a safety coefficient typically chosen between 1.2 and 1.5, depending on the vehicle type. Note that (2) 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 (see Fig. 3 for the lower friction pads thickness $\Delta' < \Delta$) which increases x_{to}^{cls} causing a corresponding increase of $F_{pp}(x_{to}^{cls})$ (see Fig. 3 and (2)). Then, due to the local positive slope of $F_{pp}(x_{to})$ around x_{to}^{cls}, we can say that wear helps to avoid unlocking when the clutch is closed.

C. Engagement manoeuver

During the clutch engagement the diaphragm spring determines the pressure plate position $x_{pp}(x_{to})$. The cushion spring compression δ_f is determined by x_{pp}. Moreover, by neglecting the dynamics of the axial motion of the clutch disk, it can be regarded as a pure elastic component devoid of mass and, therefore, the torque transmitted by the clutch, by omitting the dependencies, can be written as

$$T_{fc} = nR_\mu F_{fc} \tag{3}$$

where R_μ is a coefficient (function) taking into account the dynamic friction phenomenon. The expression (3) represents a simplified version of the proposed transmissibility model which will be more deeply analyzed and justified in Section IV. For now we describe the evolution of the main variables during a typical clutch engagement manoeuver by considering (3) and the characteristics in Fig. 3. From (3) it is clear that the cushion spring compression δ_f and the corresponding force $F_{fc}(\delta_f)$ determine the torque transmissibility. When x_{to} increases from 0 up to x_{to}^{cnt} the pressure plate position

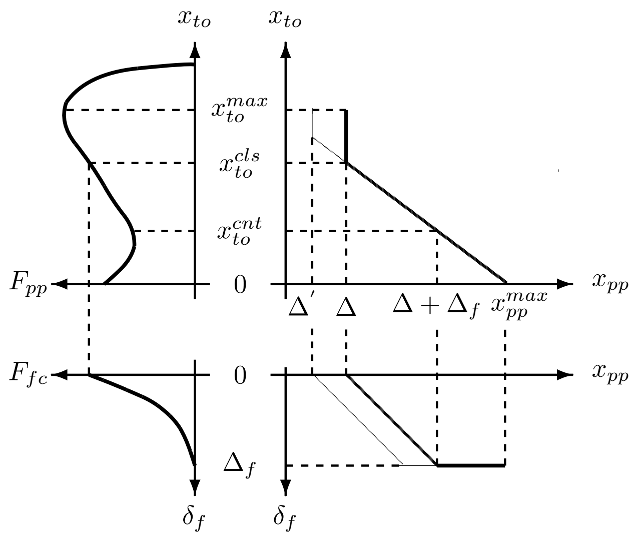

Fig. 3. Dry clutch characteristics. For a clutch engagement manoeuver the plots should be intended by considering the following sequence of dependence, given x_{to}: $x_{pp}(x_{to})$, $\delta_f(x_{pp})$ and $F_{fc}(\delta_f)$. When the clutch is mounted on the driveline x_{to} is constrained to be less than x_{to}^{max} which is the value of x_{to} corresponding to the maximum value of F_{pp}.

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 for $x_{to} \in [0, x_{to}^{cnt}]$ the clutch remains open ($\delta_f = \Delta_f$) and the torque transmitted by the clutch is zero, see (3) and note that for $\delta_f = \Delta_f$ we have $F_{fc}(\delta_f) = 0$. One should notice that wear clearly causes an increase of x_{to}^{cnt}, see Fig. 3. The value of x_{to}^{cnt} can be estimated on-line at the beginning of engagement manoeuvres as the value of the throwout bearing position x_{to} for which a nonzero clutch speed is detected.

When x_{to} becomes larger than x_{to}^{cnt} the flywheel is in contact with the friction pad, the cushion spring compression starts and the torque transmitted by the clutch becomes different from zero, see (3) with $\delta_f < \Delta_f$ which makes $F_{fc}(\delta_f) \neq 0$. When x_{to} is such that $x_{pp} = \Delta$, the cushion spring is completely compressed, i.e. $\delta_f = 0$, and the clutch is closed. Remind that during the engagement the clutch is locked–up for a value $x_{to} \in [x_{to}^{cnt}, x_{to}^{cls})$, i.e for $\delta_f \in (0, \Delta_f)$. Indeed the cushion and diaphragm springs are designed so that $F_{fc}(\delta_f = 0) = F_{pp}(x_{to}^{cls})$, see Fig. 3, and since (2) must be satisfied, the clutch will be locked–up before the cushion spring is completely compressed. Obviously the smallest value of x_{to} for which the clutch is locked–up depends also on the engine torque.

III. Transmissibility models

Before analyzing the proposed torque transmissibility model, it is interesting to reinterpret the classical clutch transmissibility models proposed in the literature as particular cases of (3). For simplicity we assume a positive slip speed (which is typical of engagement during vehicle launch) and we neglect disks abrasion and temperature effects.

A. Static models

Most common models of a dry clutch torque transmissibility can be viewed as a specification of (3), with R_μ being the product of the equivalent radius of the friction surfaces and a dynamic friction (function), and F_{fc} the normal force on the clutch plate. In [5], [17] and [18] such normal force is related to the pressure applied by the clutch actuator. In our analysis we are interested in representing the dependency of the normal force on the cushion spring compression determined by the actuator (throwout bearing) position. Transmissibility models with such type of dependency have been proposed in [19] and [4], through therein the normal force on the clutch plate was not interpreted as the cushion spring force.

The clutch torque transmission model proposed in [19] can be reformulated and rewritten in the form (3) with

$$F_{fc}(x_{to}) = \bar{F} \cdot \frac{x_{to} - x_{to}^{cnt}}{x_{to}^{cls} - x_{to}^{cnt}} \tag{4a}$$

$$R_\mu(\omega_{fc}) = \bar{R}_b - (\bar{R}_b - \bar{R}_a)\ \mathtt{exp}\left(-\frac{\omega_{fc}}{\omega_{fc_0}}\right) \tag{4b}$$

where $\omega_{fc} = \omega_f - \omega_c$ is the slip speed, with ω_f the flywheel speed and ω_c the clutch disk speed, ω_{fc_0} is the slip speed when the engagement starts, $\bar{F}$, x_{to}^{cnt}, x_{to}^{cls}, $\bar{R}_a$ and $\bar{R}_b$ are parameters to be identified. The model (4) assumes a proportionality between the transmitted torque and the actuator position, which is not realistic, as it will be shown in our model. Moreover, the estimation of the model parameters $\bar{R}_a$ and $\bar{R}_b$, is complicated by the fact they are physically meaningless.

A more complex dependence of the clutch torque on the actuator position and slip speed is presented in [4]. The clutch torque proposed in that paper can be reformulated and rewritten in the form (3) with

$$F_{fc}(x_{to}) = \check{F} \cdot \left(1 - \sqrt{1 - \left(\frac{x_{to} - x_{to}^{cnt}}{x_{to}^{cls} - x_{to}^{cnt}}\right)^2}\right) \tag{5a}$$

$$R_\mu(\omega_{fc}) = \check{R}\ \mathtt{sat}\left(\check{\mu} - \check{\beta}\omega_{fc}\right) \tag{5b}$$

with $\check{R}$ equivalent radius of the friction surfaces, and $\check{F}$, x_{to}^{cnt}, x_{to}^{cls}, $\check{\mu}$ and $\check{\beta}$ being parameters to be identified. The model assumes a piecewise linear dependence of the clutch torque on the slip speed, which is clearly different from the model structure (4). The saturation function in (5b) was not considered in [4] and it has been here introduced in order to better fit the model (5) with our experimental data. A justification for that will be clear in Section VI. It should be noticed that the function (5a) used in the model to represent the dependence of the torque on the throwout bearing position is not based on physical considerations.

In next section we show that also the proposed transmissibility model is based on the main structure represented by (3), but our modeling approach allows to clearly identify the different contributions and to propose an experimental procedure for tuning the characteristic parameters.

B. Torque estimation via dynamic model inversion

An alternative model for the torque transmissibility is based on the possible estimation of the torque transmitted by the clutch by using a driveline dynamic model, so as proposed in [13] and [20]. Assume that, with suitable rigidity hypothesis, the driveline dynamic model can be written as

$$J_f\,\dot{\omega}_f = T_e(\omega_f) - T_{fc}(x_{to}, \omega_{fc}) \tag{6a}$$

$$J_c(r)\,\dot{\omega}_c = T_{fc}(x_{to}, \omega_{fc}) - T_L(\omega_c, r) \tag{6b}$$

where J_f is the flywheel inertia (including the equivalent engine inertia), T_e is the net engine torque, $J_c(r)$ is the vehicle inertia referred to the mainshaft (including the clutch disk inertia), r is the gear ratio and T_L is an equivalent load torque referred to the cluch disk. 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 (6a) to (6b) with the assumption $\omega_f = \omega_c$, i.e. zero slip speed.

The model (6) is very simple, but can be useful for getting insights on the engagement process. For instance, simple algebraic manipulations allow to show that by assuming continuity of the load torque at the time instant when the clutch is locked-up, say $\bar{t}$, the discontinuity of the driveline acceleration at $\bar{t}$ can be written as (see [28] for details)

$$\dot{\omega}_c(\bar{t}^+) - \dot{\omega}_c(\bar{t}^-) = \frac{J_f\dot{\omega}_{fc}(\bar{t}^-) + T_e(\bar{t}^+) - T_e(\bar{t}^-)}{J_f + J_c(r)}. \tag{7}$$

Equation (7) shows that the driveline jerk can be reduced by limiting $\dot{\omega}_{fc}(\bar{t}^-)$ and by operating on the engine torque (ideally by designing a suitable engine torque discontinuity).

The torque transmitted by the clutch during the engagement might be estimated by inverting (6a):

$$\hat{T}_{fc}(x_{to}, \omega_{fc}) = -J_f\,\hat{\dot{\omega}}_f + \hat{T}_e(\omega_f) \tag{8}$$

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 (6b) one can write

$$\hat{T}_{fc}(x_{to}, \omega_{fc}) = J_c(r)\,\hat{\dot{\omega}}_c + \hat{T}_L(\omega_c, r). \tag{9}$$

The estimators (8) and (9) can be easily implemented on electronic control units but suffer from noise and uncertainties in the accelerations and torques estimations. Indeed (8) is typically used when the engine speed is constant so that the torque transmitted by the clutch can be approximated by the engine torque. Due to the above mentioned limitations and to the difficulties in getting a good estimation of the model parameters, the torque estimators based on the inversion of the dynamic model do not provide the robustness features needed for a real application. To this aim the transmissibility characteristic obtained by modeling the friction phenomenon can represent an interesting alternative.

IV. Equivalent friction radius

In this section we motivate and detail the expression (3) of the torque transmissibility characteristic, showing how the pressure distribution on the friction pads geometries and the friction phenomenon determine the expressions for the equivalent friction radius R_μ. A preliminary analysis in this direction has been presented in [30].

A. Pressure based equivalent radius

The force applied by the cushion 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

$$F_{fc} = \int_0^{2\pi} \int_{R_1}^{R_2} \sigma(\rho, \varphi, F_{fc})\, \rho\, d\rho\, d\varphi \tag{10}$$

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

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

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

$$R_\mu = \frac{\int_0^{2\pi} \int_{R_1}^{R_2} \tau(\rho, \varphi, F_{fc}) \rho^2 d\rho d\varphi}{\int_0^{2\pi} \int_{R_1}^{R_2} \sigma(\rho, \varphi, F_{fc}) \rho d\rho d\varphi} \tag{12}$$

and by using (10), the expression (11) can be rewritten as (3). A uniform distribution along the angular direction can be usually assumed, i.e. σ and τ do not depend on φ. Therefore (12) becomes

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

In order to obtain an expression for R_μ (and 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

$$\tau(\rho, F_{fc}) = \mu(\rho\omega_{fc})\, \sigma(\rho, F_{fc}) \tag{14}$$

where $\mu(v)$ is the friction coefficient (or more precisely the friction function), v being the tangential velocity. Since $v = \rho\omega_{fc}$, friction is a function of the slip speed ω_{fc} and it is different for each radius of clutch disk. If σ is assumed to be constant, (13) with (14) becomes

$$R_\mu(\omega_{fc}) = \frac{2}{R_2^2 - R_1^2} \int_{R_1}^{R_2} \mu(\rho\omega_{fc}) \rho^2 d\rho. \tag{15}$$

Instead, under the assumption of uniform wear of pads during contacts, σ will be proportional to the inverse of ρ [29] and (13) with (14) becomes

$$R_\mu(\omega_{fc}) = \frac{1}{R_2 - R_1} \int_{R_1}^{R_2} \mu(\rho\omega_{fc}) \rho d\rho. \tag{16}$$

From (15) and (16) it is clear that in order to get an expression for R_μ (and T_{fc}) we need to fix the dependence of friction on slip speed.

Different models for the function $\mu(v)$ have been proposed in the literature. The Coulomb friction model is surely the most commonly method used to describe the friction in mechanical contacts. With such a model the friction function is assumed to be proportional to the signum of the velocity. Under such an assumption, by considering ω_{fc} positive (so as typical for the vehicle launch), μ will not depend on ρ thus simplifying the computation of (15) and (16). In particular by using

$$\mu(\rho\omega_{fc}) = \mu_d\ \mathtt{sign}(\rho\omega_{fc}) = \mu_d \tag{17}$$

with μ_d being the dynamic friction coefficient, if σ is assumed to be constant, the expression (15) becomes

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

If σ is assumed to be proportional to the inverse of ρ the expression (16) becomes

$$R_\mu = \mu_d \frac{R_1 + R_2}{2}. \tag{19}$$

B. Transmissibility characteristic

The analysis above allows to rewrite the dry clutch transmissibility characteristic (3) in the following more explicit form

$$T_{fc}(x_{to}, \omega_{fc}) = n R_\mu(\omega_{fc})\ F_{fc}(\delta_f(x_{to})) \tag{20}$$

where $R_\mu(\omega_{fc})$ is given by (15) or (16), $F_{fc}(\delta_f)$ is the cushion spring characteristic and $\delta_f(x_{to}) \in [0, \Delta_f]$ is a saturated linear function dependent on wear and characterizing the diaphragm spring, which will be detailed in next section.

It is important to stress that the model (20) is general in the sense that it can be used for obtaining the torque transmissibility characteristic also for more complex friction dependencies, e.g. static, dynamic or experimentally identified.

V. Torque transmissibility tuning via experimental data

At this point it should be clear that in order to obtain the torque transmissibility characteristic (20) one needs the cushion spring characteristic $F_{fc}(\delta_f)$, the cushion spring compression function $\delta_f(x_{to})$ and the friction function $\mu(\rho\omega_{fc})$. To this aim tribological experiments have been done on a commercial dry clutch with the following geometrical parameters: $R_1 = 74$ mm, $R_2 = 104$ mm, $\Delta = 7.2$ mm.

A. Cushion spring characteristic

The compression characteristic of the cushion spring in the axial direction has been achieved through experimental tests. This load-deflection curve gives the axial spring reaction versus axial displacement obtained by compressing the cushion disk between two rigid plates. In the automotive applications, there are two prevailing cushion spring geometries: the first one, where the waved peripheral paddles are one-piece featured with the inner part riveted to the clutch hub, and a second one where $8 \div 10$ unconnected paddles are riveted to the friction surfaces as well as the clutch hub. The tested cushion spring belongs to the latter family and the paddle is composed by a couple of two steel thin sheets mounted mirrored on the hub with respect to the clutch radial plane. The Instron 4200 series IX automated Material Testing system, suitable for compression and traction testing, has been used. Appropriate load cell detects the cushion spring reaction force on the sample in consequence of a certain compression. Since the paddles appear as a system of parallel axial springs, the whole compression characteristic is the sum of the single paddle axial response and according to the repeatability of the forming process the characteristic is obtained by multiplying the single pad characteristic by the paddles number. Experiments have been realized on a single spring paddle placed on the lower steady bar, while the upper one approaching speed has been set to $0.20 \div 0.50$ mm/min (Fig. 4). In Fig. 5 it is depicted the cushion spring characteristic experimentally obtained; coherently with Fig. 3 δ_f is set to zero at the smallest throwout bearing position for which the greatest cushion spring deflection is obtained. Experimental tests carried out on different samples provide similar results.

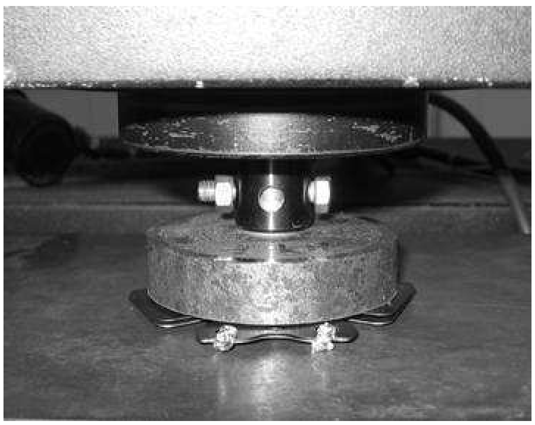

Fig. 4. Experimental set-up for the characterization of the cushion spring.

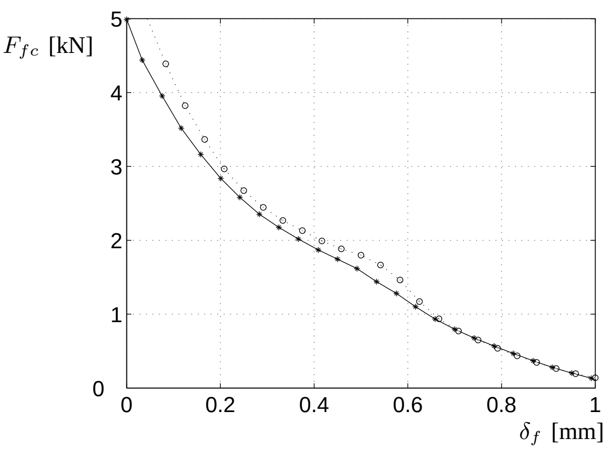

Fig. 5. Experimental data of cushion spring force vs. the cushion spring compression: first sample (continuous line with stars), second sample (dash line with circles).

B. Pressure plate lift characteristic

In order to show the dependence $\delta_f(x_{to})$ in the throwout bearing engaging range, a laboratory test has been performed on a bench clamped flywheel coupled to clutch disk and clutch mechanism (diaphragm spring and pressure plate). A couple of plunger-dial indicator with 0.01 mm resolution has been used; the measuring tools have been mounted on a rigid frame with the first plunger in contact with a washer mounted on the throwout bearing to measure its axial motion, while the second one was placed on the rivet which joins the pressure plate to one of the three drive straps. The throwout motion has been achieved through a screw-nut system. The pressure plate lift motion has been measured twice along two complete throwout bearing strokes. By measuring the friction pads thickness as well as the cushion spring axial displacement at rest, the pressure plate lift measurement allows calculating the characteristics shown in Fig. 6. Through comparison with Fig. 3 for the interpolating characteristic we have $\Delta_f = 1$ mm, $x_{to}^{cnt} = 4.02$ mm, $x_{to}^{cls} = 7.78$ mm and $x_{to}^{max} = 12$ mm.

Note that the piecewise linear interpolation has been chosen in order to have the least square error for the typical throwout bearing range between contact and lock-up, in our case $x_{to} \in [4, 7]$ mm. That range will be also confirmed by the simulation results in next section.

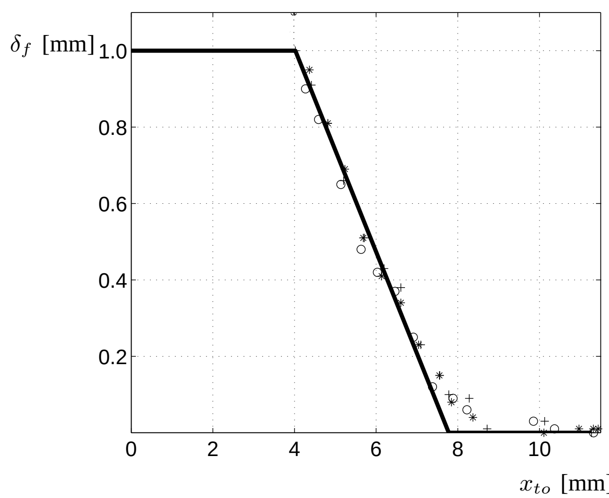

Fig. 6. Cushion spring compression vs. throwout bearing position for three different tests (circle, star, cross) and a corresponding interpolating piecewise linear characteristic (continuous).

C. Friction vs sliding velocity

In order to determine the friction function $\mu(v)$, experimental tests have been carried out on the friction material of the same dry clutch used for the cushion spring characterization. The tribometer WAZAU TRM 100 has been used. This instrument is suitable for test type pin-on-disk, disk-on-disk, ball-on-disk with dry and lubricated contacts from zero speed up to 3000 rpm, with normal loads to the contact up to 100 N. For the tests it has been used a pin with the friction material of a clutch disk glued on top, in a rectangular shape of dimensions $15 \text{x} 18\,\text{mm}^2$. The experimental tests have been executed as follows: the top disk is rotated at constant angular speed. Since the pin is fixed, the disk angular speed represents the angular slip speed ω_{fc}. The distance between the pin and axis of rotation of the disk is $R = 30$ mm, therefore the sliding velocity v is simply obtained as $v = R\omega_{fc}$. Moreover, the pin is clamped to a torque measurement system, which detects the torque on the rotating axis, i.e. T_{fc}. Besides, the normal force exerted by the disk to the pin plays the role of F_{fc}. It is so possible to get an estimate of the average pressure, known the area of the rectangular contact element. The values of speed, considered in experiments, are the typical values of a clutch engagement. Considering the friction pads average pressure during an engagement ($0 \div 0.3\,$MPa), we have considered two different load forces: $F_{fc} = 55\,$N which corresponds to an average pressure of $0.2\,$MPa and $F_{fc} = 82\,$N which corresponds to a pressure of $0.3\,$MPa. Fig. 7 shows the friction μ versus the sliding velocity v, obtained by dividing the measured torque by the normal force and the radius corresponding to the pin position.

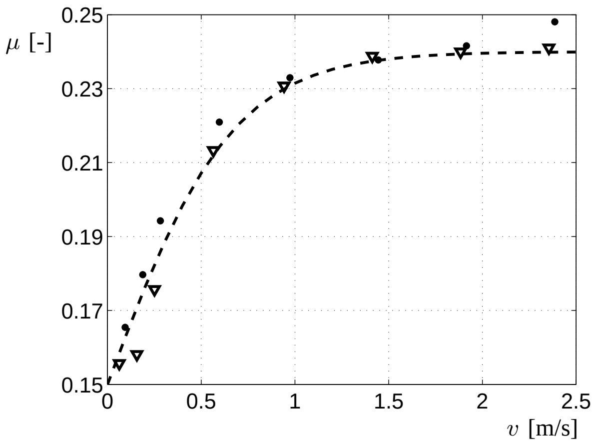

Fig. 7. Friction coefficient measured at differential average pressure: 0.2MPa (dots), 0.3MPa (triangles) and interpolating curve (dashed).

For each test, the different values have been extrapolated with the tribometer in steady-state conditions. Moreover, the contact surface temperature was monitored by an infrared sensor, so as to exclude possible thermal effects. The average temperatures during the experiments were 27°C and 39°C, respectively. Based on the experimental results, the function that fits better our experimental data (for strictly positive slip speed) is given by

$$\mu(\rho\omega_{fc}) = \mu_s + (\mu_d - \mu_s)\ \mathtt{tanh}(\gamma\rho\omega_{fc}) \tag{21}$$

where μ_s, μ_d and α have been identified and the corresponding values are: $\mu_s = 0.15$, $\mu_d = 0.24$ and $\gamma = 1.5$ s/m.

By substituting (21) in (15) and (16) we have obtained numerically the dependencies of R_μ on the slip speed ω_{fc}. The corresponding functions are shown in Fig. 8.

D. Experimental torque transmissibility map

The clutch torque transmissibility map can be now obtained by substituting in (20) the experimental characteristics shown above. By using the equivalent radius function in Fig. 8, the cushion spring characteristic in Fig. 5 and the linear expression for $\delta_f(x_{to})$ derived from Fig. 3, the resulting transmitted torque map depending on slip speed and throwout bearing position is shown in Fig. 9.

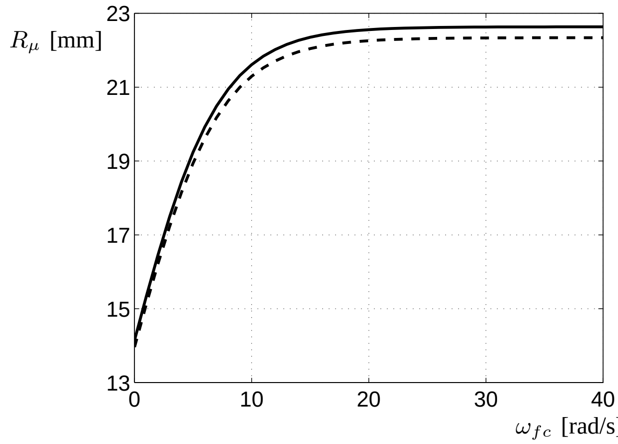

Fig. 8. Equivalent radius R_μ vs. slip speed ω_{fc} using the friction function (21) in (15) (continuous) and (16) (dashed).

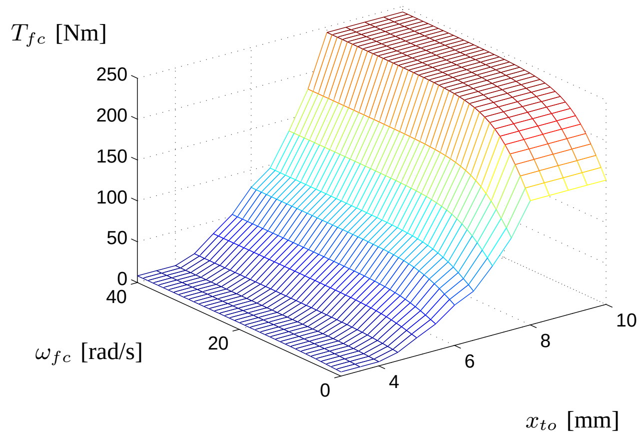

Fig. 9. Transmitted torque map for different slip speeds and throwout bearing positions.

VI. Transmissibility characteristic for AMT control

In this section we present the use of the dry clutch torque transmissibility characteristic in an AMT control scheme for vehicle launch and we compare the controller performance with different transmissibility torque models. The control scheme is taken from an experimentally evaluated controller presented in [12]. The Matlab/Simulink controller is co-simulated with a CarSim detailed powertrain model. The control scheme is shown in Fig. 10. In particular the control design assumes that the engine and clutch torques are almost the same because a constant engine speed is requested during the engagement, see (6a).

The inputs for the powertrain CarSim model are the throttle α and the throwout bearing position x_{to}. The output of the Proportional Integral controller provides the reference clutch torque. The reference position x_{to}^{ref} is obtained by entering in the clutch torque map in Fig. 9 with the reference torque T_{fc}^{ref} and the actual speed ω_{fc}. By assuming the presence of a closed loop position controlled actuator, we fix $x_{to} = x_{to}^{ref}$.

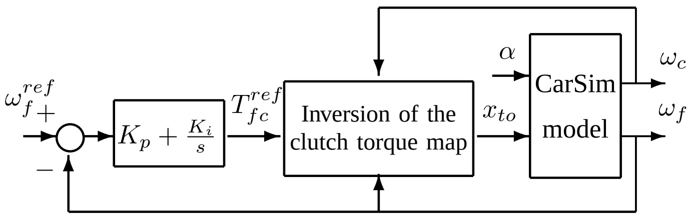

Fig. 10. A block scheme of the AMT control system. Note that we assume $x_{to} = x_{to}^{ref}$.

A medium-size car is considered as case study and in Fig. 11 the engine map is shown. The controller gains, chosen by trial and error, are $K_p = 50$ Nms and $K_i = 20$ Nm. The reference engine speed is $\omega_f^{ref} = 100$ rad/s.

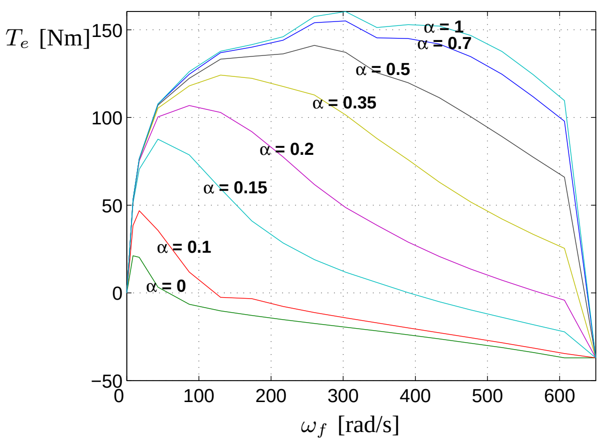

Fig. 11. Engine map $T_e(\omega_f, \alpha)$.

Fig. 12 and Fig. 13 show the co-simulation results of two typical start-up manoeuvres with slow torque and fast torque requests. The lock-up is detected with the conditions of slip speed less than 3 rad/s. When lock-up is detected the throwout bearing position x_{to} is moved to x_{to}^{cls} with a constant slew rate. A performance comparison has been done by inverting

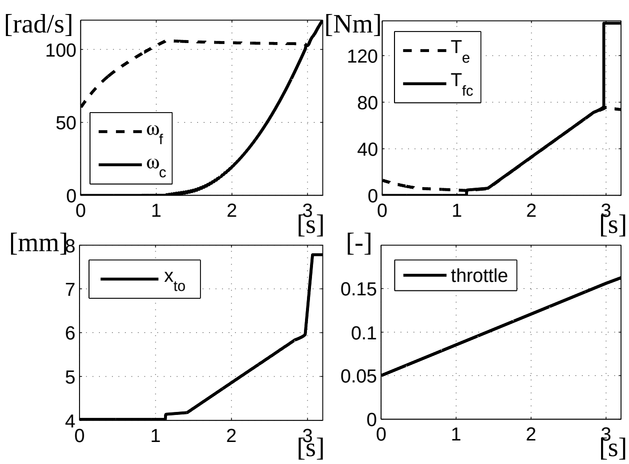

Fig. 12. Vehicle start-up co-simulation: slow torque request.

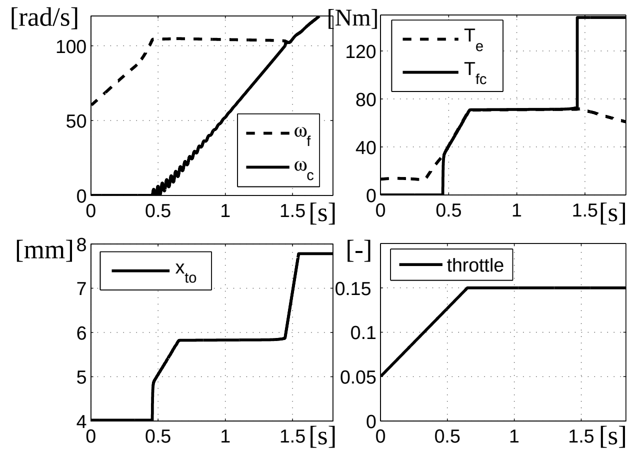

Fig. 13. Vehicle start-up co-simulation: fast torque request.

the different transmissibility models presented above. To this aim the parameters of the models (4) and (5) have been chosen by using our experimental procedure and results. By substituting the cushion spring compression $\delta_f(x_{to})$ shown in Fig.6 into the cushion spring force characteristic $F_{fc}(\delta_f)$ reported in Fig. 5, one determines the relationship $F_{fc}(x_{to})$ corresponding to our experimental procedure. By approximating our data $F_{fc}(x_{to})$ with the expressions (4a) and (5a) a possible parameters choice is: $\bar{F} = 3$ kN and $\check{F} = 15$ kN. The parameters of (4b) and (5b) have been obtained through their approximation with our experimental data shown in Fig.8. That motivates also the use of the saturation function in (5b), which was not considered in [4]. The resulting parameters are $\bar{R}_a = 14$ mm, $\bar{R}_b = 32$ mm in (4b) and $\check{R} = 90$ mm, $\check{\mu} = 0.16$, $\check{\beta} = -0.0066$ in (5b). Tables I-II show performance in terms of engagement time duration Δt, dissipated energy

$$E_d = \int_0^{\Delta t} T_{fc} \omega_{fc} dt \tag{22}$$

and slip acceleration $\dot{\omega}_{fc}$ at lock-up. The results are reasonable with respect to a typical clutch engagement during a start-up mode and the performance are very similar by inverting the different transmissibility models. What must be emphasized is that the parameters of our model have a clear physical meaning and they are achievable through a precise experimental procedure, differently for the parameters of models (4) and (5). On the other hand, the torque transmissibility map obtained with our approach allowed to detail the models (4) and (5) and to obtain the corresponding parameters.

TABLE I
CONTROLLER PERFORMANCE FOR A SLOW THROTTLE SIGNAL.

Case	Δt [s]	E_d [J]	$\lvert\dot{\omega}_{fc}(t^-)\rvert$ [rad/s^2]
model (20)	2.532	3487	157.4
model (4)	2.531	3485	151.2
model (5)	2.531	3479	152.8

VII. Conclusion

A novel model of dry clutch torque transmissibility during

TABLE II
CONTROLLER PERFORMANCE FOR A FAST THROTTLE SIGNAL.

$Case$	Δt [s]	E_d [J]	$\lvert\dot{\omega}_{fc}(t^-)\rvert$ [rad/s^2]
$model$ (20)	1.032	3475	199.2
$model$ (4)	1.030	3469	188.1
$model$ (5)	1.033	3482	195.2

engagement has been proposed. The model is obtained by analyzing the friction phenomena that generate the transmitted torque and by clarifying the influence of the cushion spring compression on the transmitted torque. Differently from the models previously presented in the literature, the slip speed and throwout position dependent functions that appear in the proposed model have a clear physical meaning. Indeed, a tuning procedure for the torque characteristics parameters has been proposed and experimentally validated. The main steps of such procedure are the evaluation of the following characteristics: cushion spring compression vs throwout bearing position, cushion spring force vs its compression, friction vs sliding velocity. An equivalent radius function takes into account the pressure distribution dependence. The classical equivalent radius expression have been shown to be recovered from the proposed model with the simplifying assumption of uniform pressure (or uniform wear) and constant friction. It is also shown how other transmissibility models can be recasted into the proposed formulation and the corresponding parameters obtained by exploiting the proposed experimental procedure. A typical control scheme for clutch engagement in automated manual transmission, where the inversion of the transmissibility model is needed, has been considered. Numerical co-simulations based on a detailed CarSim powertrain model show that the use of the proposed torque transmissibility model provides realistic results in terms of engagement time, dissipated energy and slip acceleration of clutch lock-up.

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