# A Survey on Modeling and Engagement Control for Automotive Dry Clutch

Antonio Della Gatta♮, Luigi Iannelli♮ *Senior Member, IEEE*, Mario Pisaturo◇, Adolfo Senatore◇, Francesco Vasca♮ *Senior Member, IEEE*

***Abstract*—Single and dual dry clutches are widely used in automotive transmissions because of their reliability and efficiency paired with low fuel consumption and reduction of pollutant emissions. A prerequisite to attain these goals in automated manual transmissions is the adoption of a "good" clutch model and the design of a robust control strategy. Several papers have been dedicated to modeling and engagement control of dry clutches, both in mechanical and control engineering literatures. The wide practical interest of the application and the maturity of the research field motivate the need for a state of the art analysis on the topic. This paper aims at exploring clutch transmissibility models and engagement control strategies proposed in the last decades. An overview of dry clutch torque modeling is proposed together with a discussion on the main factors which affect the torque transmission, such as temperature, sliding speed, contact pressure, wear. Then, the different clutch actuator architectures and control strategies investigated in the literature are presented.**

## I. INTRODUCTION

Automotive transmissions based on dry clutch architectures are widely used, so many solutions with different technologies have been proposed in these years [1]. Since its invention in the late $19^{th}$ century, the manual transmission has been remarkably improved and its advantages in terms of costs, reliability, efficiency, low fuel consumption and pollutant emissions made it the most common transmission type, especially in the European market [2]. An evolution of this solution is the Automated Manual Transmission (AMT) which was introduced about thirty years ago for race cars and it is currently adopted for many different driveline architectures.

The main difference between manual transmissions and AMTs is that the latter are equipped with an intelligent actuator which manages the shifting process. So, a conventional car can easily be converted into one with AMT by improving its performances. On the other hand, the main disadvantage of AMTs is the torque gap during gearshifts which reduces driving comfort and efficiency. A technological solution that mitigates this problem is the dual dry clutch transmission, sometimes referred to as twin-clutch transmission or double-clutch transmission. It is based on two separate dry clutches for odd and even gear sets and it can fundamentally be described as two separate AMTs with their respective clutches and contained within one housing. In order to familiarize with the structure of a dual dry clutch system one can consider the scheme in Fig. 1. The architecture of a single clutch transmission can be easily deduced as a simplification of that with dual clutches.

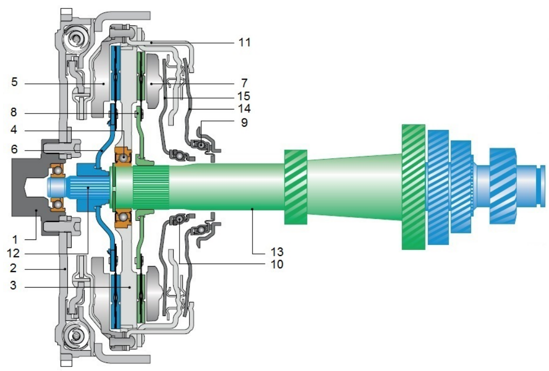


Figure 1: A typical dual dry clutch scheme [3] - (1) *crankshaft*, (2) *dual mass flywheel*, (3) *central plate*, (4) *support bearing*, *pressure plates of the first* (5) *and second* (7) *clutches*, *disc of the first* (6) *and second* (8) *clutches*, *engagement bearing of the first* (9) *and second* (10) *clutches*, (11) *tie rod*, *first* (12) *and second* (13) *transmission input shafts*, *lever spring of the first* (14) *and second* (15) *clutches*.

The main advantages of dry clutch transmissions with respect to the other type of transmissions are: fuel saving, pollutant reduction, fast response coupled to high comfort, high efficiency and reliability, low production and maintenance costs. Unfortunately, dry clutches suffer for limited transmissible torque because of drawbacks related to durability, thermal capacity and stability of their characteristics [4]. Furthermore, in order to ensure the driving comfort, it is also necessary to eliminate or minimize the associated vibrations [5].

The advantages and performances of automated dry clutches, both in single and dual configurations, are strongly influenced by the clutch torque transmissibility model implemented in the control strategy. The interest on modeling and engagement controls for dry clutches is demonstrated by the wide literature on these topics and by the continuously development of these transmissions in real cars. Time is mature for proposing a state of the art analysis on these topics, which is the goal of this paper.

The paper is organized as follows: in Sec. II a deep analysis on main factors which affect torque transmission is proposed. A focus on thermal effects is reported in Sec. III whereas lumped thermal models are discussed in Sec. IV. Section V describes different actuators (electrohydraulic, electropneumatic

(♮) Dipartimento di Ingegneria, Università del Sannio, Piazza Roma 21, 82100 Benevento, Italy, email: dellagatta@unisannio.it, luiannel@unisannio.it, vasca@unisannio.it
(◇) Dipartimento di Ingegneria Industriale, Università di Salerno, Via Giovanni Paolo II 132, 84084 Fisciano (Salerno), Italy, email: mpisaturo@unisa.it, a.senatorev@unisa.it

and electromechanical) used in automotive applications. In Sec. VI and Sec. VII clutch torque estimation and control strategies are debated.

## II. Dry clutch torque modeling

An accurate transmissibility model is crucial to improve performances of automated dry clutches. Indeed, 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 basic model of a single clutch consists of a static friction characteristic which relates the transmitted torque to the normal force, the slip speed, the actuation variable and the temperature. In the following we assume that the clutch engagement is controlled through the actuator position. In some cases, the torque characteristic is represented as a function of the actuator pressure, however the corresponding models can be derived from those presented below by assuming a static relation between the piston chamber pressure and the actuator position. Moreover, by collecting torque characteristics of single clutches, one can easily obtain transmissibility models of a dual clutch, as well.

### A. Torque dependence on pressure distribution

In a dry clutch, the frictional torque is closely related to the clamping load and sliding between clutch disc, flywheel and pressure plate. The friction force results from the normal force provided by the diaphragm spring which axially presses the disc pads against cushion spring reaction, $F_{fc}$.

Finite Element (FE) and dedicated experiments can be carried out in order to determine the specific dependence of the pressure on the radial direction. In [6] it is demonstrated with experimental and FE results that the cushion spring acts mainly along the framed contact lines. The numerical results shown in Fig. 14 in [7] indicate a pressure distribution that varies almost linearly with the radial direction with maximum and minimum values at the outer and inner disc radii, respectively. A contact analysis on a three dimensional element model is proposed in [8] by taking into account the effects of the pressure and centrifugal force loads.

The dependences of the contact pressure can be represented through a corresponding transmitted torque model. For each pair of friction surfaces, a typical expression of the transmitted torque in a dry clutch can be written as

$$T_{fc}(F_{fc}, \omega_{fc}) = 2\, R_\mu(F_{fc}, \omega_{fc})\, F_{fc} \tag{1}$$

where $R_\mu$ is the so called *equivalent friction radius* [9], that takes into account the friction phenomena and the specific geometry, $\omega_{fc} = \omega_f - \omega_c$ is the slip speed, $\omega_f$ is the flywheel speed, $\omega_c$ is the clutch disc speed, and the factor 2 is due to the number of friction surfaces.

Experimental results on clutch material pads proved clear influence of the contact pressure and the slip speed on the friction coefficient [10]. In particular, the higher the contact pressure, the higher the friction coefficient with a nearly linear relationship. One more interesting outcome deals with slip speeds greater than $3\,\mathrm{m/s}$: the friction coefficient reaches an asymptotic value. Under the hypothesis of uniform pressure distribution along the angular direction and an uniform wear of pads during contacts [11], the equivalent radius is

$$R_\mu(\omega_{fc}) = \frac{1}{R_2 - R_1} \int_{R_1}^{R_2} \mu(\rho\omega_{fc})\rho \, d\rho \tag{2}$$

where $R_1$ and $R_2$ are the inner and outer radii of the pads, $\rho$ is the radius variable, and $\mu(\rho\omega_{fc})$ is the *friction function* that depends on the tangential velocity $\rho\omega_{fc}$ [12].

### B. Friction dependence on the slip speed

In the literature the friction function $\mu(\rho\omega_{fc})$ has been modeled through different approaches. In the following, by referring to typical clutch engagement maneuvers, it is assumed for simplicity that the slip speed is always positive as well as the friction function is defined as non-negative. An anti-symmetric behavior is usually assumed for negative speeds.

A classical model is the Coulomb friction that corresponds to a constant *dynamic* friction coefficient, i.e., $\mu(\rho\omega_{fc}) = \mu_d$. It is well known that the *static* friction coefficient $\mu_s$ (also called *breakaway* friction coefficient) is larger than the dynamic one. At zero speed the friction behavior has a discontinuity given by $\mu_s - \mu_0$, i.e., $\mu(0^+) = \mu_0$ different from $\mu_s$. For instance, in the case of the Coulomb friction one has $\mu_0 = \mu_d$. In other models the friction varies with the slip speed, tending to a constant value, say $\mu_d$, for large speeds.

In [13] a regularization of such behavior has been proposed with the following continuous friction function

$$\mu(\rho\omega_{fc}) = \mu_d + (\mu_0 - \mu_d) \exp\left(-\frac{\rho\omega_{fc}}{\alpha_1}\right)^{\alpha_2} \tag{3}$$

where the so-called *Stribeck velocity* $\alpha_1$ determines the decay rate of the sliding friction coefficient with the sliding velocity of the friction surfaces. The parameter $\alpha_1$ depends on the material properties and on the surface finish. The parameter $\alpha_2$ determines the shape of the curve [14].

In [15], by interpolating experimental data, the following friction function has been suggested

$$\mu(\rho\omega_{fc}) = \mu_0 + (\mu_d - \mu_0)\, \tanh\left(\frac{\rho\omega_{fc}}{\alpha_1}\right)^{\alpha_3}, \tag{4}$$

with $\alpha_3 = 1/3$. In [12] an expression similar to (4) is considered with $\alpha_3 = 1$. The positive parameter $\alpha_1$ in (4) plays the role of the Stribeck velocity in (3).

Other authors consider a friction function dependent only on the slip speed and not on the sliding velocity, i.e., $\mu(\rho\omega_{fc}) = \mu(\omega_{fc})$. A linear dependence can be expressed as

$$\mu(\omega_{fc}) = \max\left\{\mu_s - \alpha_2\omega_{fc}, \mu_d\right\} \tag{5}$$

where $\alpha_2$ is a positive parameter that determines the linear rate of variation of the friction which saturates when the constant dynamic friction is reached [16]. The combination of exponential and linear dependencies can be modeled by the following friction function [17]

$$\mu(\omega_{fc}) = \mu_d + (\mu_0 - \mu_d + \alpha_4\omega_{fc}) \exp\left(-\alpha_5\omega_{fc}\right), \tag{6}$$

where the radial dependence in (3) has been eliminated and the coefficient $\alpha_4$ weights the new linear relation on the slip speed.

The different friction radii obtained through the friction functions are shown in Fig. 2. ADG: ~~In general, it does not exist a unique function which best approximates the dry clutch friction, as different factors could be relevant for a given application~~In [18] a customized pin-on-disc setup has been proposed for the experimental characterization of static and dynamic friction behaviors for various sliding sample pairs. The test bench described in [19] is dedicated to establish the friction coefficient model on the basis of the sliding speed, the pressure and the surface temperature of clutch plates.

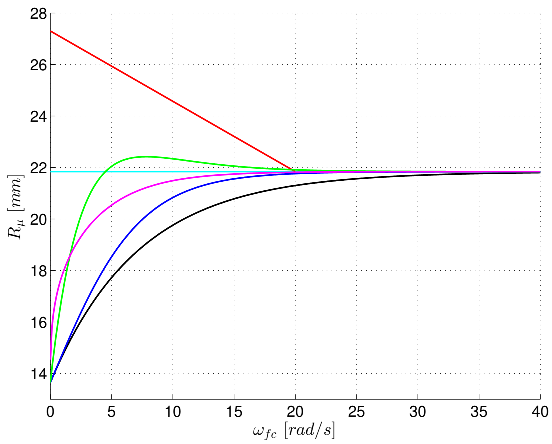


Figure 2: Equivalent friction radius $R_\mu$ versus slip speed $\omega_{fc}$ substituting into (2) a constant friction function $\mu_d$ (cyan line), the friction function (3) (black curve), (4) with $\alpha_3 = 1$ (blue curve) and $\alpha_3 = 1/3$ (magenta curve), (5) (red line), (6) (green curve) respectively. The parameters are: $\mu_s = 0.30$, $\mu_0 = 0.15$, $\mu_d = 0.24$, $R_1 = 74\,\mathrm{mm}$, $R_2 = 108\,\mathrm{mm}$, $1/\alpha_1 = 1.5\,\mathrm{s/m}$, $\alpha_2 = 3 \times 10^{-3}\,\mathrm{s/rad}$, $\alpha_4 = 2 \times 10^{-2}\,\mathrm{s/rad}$, $\alpha_5 = 0.3\,\mathrm{s/rad}$.

ADG: By observing Fig. 2, if you neglect the linear characteristic which is seems moreover more representative than an ideal behavior, the other friction characteristics exhibit an exponential trend and these are more adherent to the real behavior of the dry clutch. In particular, the friction characteristic shown in green color, for $\omega_{fc}$ values close to zero (and therefore near the engagement) exhibits an overshoot which seems typical of the friction torque transmission in dry clutches. In any case, apart from this observation, it's possible to affirm there is no specific friction characteristic that could be considered the best approximation of the real behavior of the dry clutch system.

### C. Torque dependence on actuation variable and wear

The force $F_{fc}$ in (1) is a direct input for the transmitted torque model. In real applications this force is determined by a specific actuator (see Section V). In particular, it can be considered as a function of the throwout bearing position $x_{to}$, see Fig. 1 in [12] and Fig. 5 in [20].

Then the dry clutch transmissibility characteristic (1) can be rewritten as

$$T_{fc}(x_{to}, \omega_{fc}) = 2\, R_\mu(\omega_{fc}) F_{fc}(x_{to}) \tag{7}$$

where $R_\mu(\omega_{fc})$ can be expressed by (2). The normal force can be expressed in terms of the normalized effective displacement of the actuator position defined by

$$\xi_{to}(x_{to}) = \mathrm{sat}_0^1 \left\{ \frac{x_{to} - x_{to}^{cnt}}{x_{to}^{cls} - x_{to}^{cnt}} \right\}, \tag{8}$$

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. The actuator variable $\xi_{to}$ can be statically related to the cushion spring load-deflection by assuming rigidity of the corresponding leverage mechanism [12].

A proportional relationship between the actuator position and the transmitted torque can be described by

$$F_{fc}(x_{to}) = \bar{F}\xi_{to}(x_{to}) \tag{9}$$

where $\bar{F}$ is the diaphragm spring load and $0 \leq \xi_{to} \leq 1$, see [21].

In [15] specific experiments have been carried out in order to determine the dependence of the force $F_{fc}$ on $x_{to}$ through the cushion spring deflection. In particular, it is noted that the dependence of the diaphragm spring compression on the throwout bearing position can be modeled with a saturated linear function, and the following expression was identified from experimental data:

$$F_{fc}(x_{to}) = \beta_1 \xi_{to}(x_{to}) + \beta_2 \left(\xi_{to}(x_{to})\right)^2 + \beta_3 \left(\xi_{to}(x_{to})\right)^3. \tag{10}$$

The model (10) can be also interpreted as a polynomial approximation for

$$F_{fc}(x_{to}) = \bar{F}\left[1 - \sqrt{1 - \left(\xi_{to}(x_{to})\right)^2}\right] \tag{11}$$

which is the expression investigated in [16].

Figure 3 shows the different dependencies of $F_{fc}$ on $x_{to}$ for some typical values of an automotive dry clutch. The nonlinear functions all agree on the convexity of the dependence.

The clutch torque is also influenced by wear [22]. Indeed, the wear's effect increases the clutch clearance due to the reduction of the friction plate thickness [23] and, at the same time, influences the friction coefficient behavior [18]. In [12] it is shown that the wear increases $x_{to}^{cls}$ which corresponds to a larger pressure plate force when the clutch is closed, thus better preventing unlocking. In [4] authors performed specific experiments in order to achieve clutch wear model and characterize the influence of the wear compensation mechanism on the clutch characteristics. Their model predicts the friction plate wear based on the dissipated energy and on the clutch temperature. Finally, in [6] an FE analysis has been proposed to predict facing wear as a function of the contact pressure.

## III. Thermal effects on torque transmission

The main goal of a dry clutch system is to transmit the engine torque to the gearbox and consequently to the wheels. So, to ensure an efficient torque transmission it is desirable that the clutch lining has a high friction coefficient

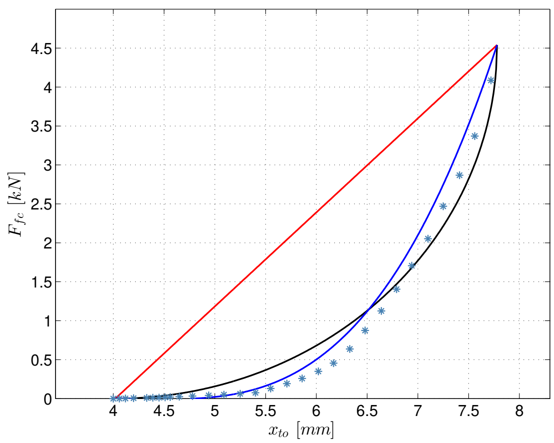


Figure 3: Cushion spring force $F_{fc}$ versus throwout bearing position $x_{to}$ using the functions (9) (red line), (11) (black curve) and a third order polynomial as proposed in [15] (blue curve). The parameters are: $x_{to}^{cnt} = 4.02\,\mathrm{mm}$, $x_{to}^{cls} = 7.78\,\mathrm{mm}$, $\bar{F} = 4.540\,\mathrm{kN}$, $\beta_1 = 26.1195 \times 10^{-3}\,\mathrm{kN}$, $\beta_2 = -1.3046\,\mathrm{kN}$, $\beta_3 = 5.8185\,\mathrm{kN}$. The star-marked points represent experimental data.

which remains constant over a wide range of pressures and temperatures. Unfortunately, the clutch torque transmissibility is highly influenced by the temperature rise due to the heat generation caused by frictional losses [24]. In the literature specific experiments were carried out in order to characterize the behavior of the friction coefficient when temperature and contact pressure vary. For instance, in [25, 26] it has been verified experimentally that at high temperatures (more than $200\,^{\circ}\mathrm{C}$ experimentally verified) a significant variation of the friction coefficient occurs when varying the contact pressure. Moreover, above a certain critical temperature, namely $250\,^{\circ}\mathrm{C} - 300\,^{\circ}\mathrm{C}$, the friction pads start having permanent damage [27]. In automotive applications, it is not unusual to attain such temperatures especially when the system is undergone to repeated clutch engagements. Even below these critical values the temperature increase determines major modifications of the clutch transmissibility.

Thermal behavior in a clutch system is a complex phenomenon and the temperature distribution on the friction pads is typically not uniform. Due to the intrinsic difficulty to perform experimental tests in order to assess the temperature field in a dry clutch during the slipping phase, the FE approach is widely used to appraise the interface temperature [28]. In this section, assumptions and exploitations of clutch FE thermal models are discussed.

### A. FE models for temperature distribution analysis

Specific assumptions must be introduced in order to construct suitable FE models. In particular, by considering the axial symmetry of the system, a 2D-model can be considered in numerical simulations instead of a more complex and, sometime worthless, 3D-model. In addition, it is possible to assume that at each contact pair the thermal flux consists of half of the total heat generated during the clutch engagement. Under such hypotheses a thermal symmetry condition, i.e. zero heat flux, can be imposed on the plane where the clutch disc is in contact with the flat spring, see Fig. 15 in [28]. Moreover, it can be assumed that the heat flows in one direction, perpendicular to the contact surfaces. Finally, due to the typical time durations and temperatures of engagement maneuvers, the radiant phenomenon can be neglected.

As analyzed in the previous section, in particular by recalling how (2) has been obtained, two further hypotheses used to build up FE models are uniform pressure and uniform wear. In fact, as highlighted in [29, 30, 31, 32] the temperature field is strongly affected by the contact pressure distribution.

### B. Geometry and materials

Numerical FE methods allow one to take into account nonlinear behaviors from a geometrical point of view, to select different materials, and to analyze the corresponding time-spatial distributions [33]. In [34, 35] the authors analyzed the effects of groove pattern and size on the temperature field and thermal energy. The influence of torque variations and pressure plate thickness on the surface temperature distribution have been also investigated [36]. In [37] different geometrical configurations (full disc, four and eight friction surfaces) and different lining materials ($SiC$, $Si_3N_4$ and $Al_2O_3$) for clutch disc have been analyzed by using the Ansys software. The results therein highlight that the temperature difference between a full disc and a disc with eight friction surfaces is about $48\,^{\circ}\mathrm{C}$, i.e. a decrease of $37\%$. In fact, the higher is the number of friction surfaces the higher is the cooling effect of the grooves. Furthermore, also the lining material has a strong influence on clutch disc thermal response. Indeed, the alumina $Al_2O_3$ has shown better thermal behavior in relation to the two other materials.

### C. Effects on cushion spring load-deflection

The torque transmitted by a dry clutch is directly influenced by the cushion spring load-deflection. Indeed, 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 [23, 38, 39, 40]. FE analyses have been used also to investigate the temperature effects on the cushion spring load-deflection behavior, so as shown in [41] where a numerical thermal model has been validated with experimental results. 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 kiss point. Secondly, by increasing the temperature, the material stiffness changes and this results in local modifications of the load-deflection characteristic slope. These two effects together lead to model uncertainties during the engagement phase. Indeed, the kiss point changes and if this issue is not taken into account the transmission control unit could lead to a wrong position of the throwout bearing and the actual transmitted torque will be different from the expected one.

### D. Contact pressure and slip speed

Simultaneous influence of temperature, slip speed and contact pressure on the friction coefficient have been taken into account in [42] where the authors proposed an original experimental friction map, see Fig. 2 in that paper, which confirms the high variability of the frictional response observed over a broad range of operating conditions. This behavior is also mentioned in [27] and discussed by using experimental data in [15].

FE analyses have been proposed to simulate the frictional behavior between contact surfaces in order to obtain the temperature and contact pressure distributions on the friction surfaces in single disc [43] and multi-disc clutches [44, 45]. In [46] a coupled thermal-mechanical FE analysis has been carried out to study the interaction between the contact pressure distribution and the thermal field during a clutch engagement by taking into account surfaces roughness. Also in this case the authors underline that the contact pressure is affected by the thermal level. Moreover, it is shown that the maximum contact pressure of a rough surface is higher than a flat surface.

## IV. Temperature-dependent lumped models

The use of parameter distributed models for temperature estimation requires a high computational effort which is often not compatible with the currently available commercial control units. On the other hand, the temperature influence on the torque characteristic cannot be disregarded and must be accurately taken into account into the model [47, 48]. Lumped models which approximate the temperature spatial distribution can be a valuable solution for the control design.

### A. Static torque characteristics

The torque lumped model (7) can be generalized by including the temperature dependencies on the equivalent radius and on the cushion spring force. Therefore the temperature-dependent torque model can be written as

$$T_{fc}(x_{to}, \omega_{fc}, \theta_c) = 2R_\mu(\omega_{fc}, \theta_c)F_{fc}(x_{to}, \theta_c) \qquad (13)$$

where $\theta_c$ is a scalar variable representing an equivalent clutch temperature.

A typical approach for considering the temperature effects on the equivalent radius consists of including the temperature dependence on the dynamic friction, say $\mu_d(\theta_c)$. For this function suitable low order polynomials are used to approximate the corresponding experimental data [15, 47, 49, 50]. In [51] the experiments were carried out by considering different values of the normal force and, for fixed normal force and slip speed, an expression similar to (13) was used to compute the dependence of the friction on the temperature.

The expression (13) highlights the temperature dependence of the cushion spring force, see [40, 41, 52]. By considering the friction models presented above, the temperature effects on the cushion spring force can be modeled by generalizing (8) with

$$\xi_{to}(x_{to}, \theta_c) = \mathrm{sat}_0^1 \left\{ \frac{x_{to} - x_{to}^{cnt}(\theta_c)}{x_{to}^{cls}(\theta_c) - x_{to}^{cnt}(\theta_c)} \right\}, \qquad (14)$$

where $x_{to}^{cnt}(\theta_c)$ and $x_{to}^{cls}(\theta_c)$ can be represented with suitable polynomial functions approximating corresponding experimental data. Temperature effects on the kiss point position $x_{to}^{cnt}(\theta_c)$ are discussed in [15] (see Fig. 6 therein) where experimental results of three partial clutch engagements at different temperatures are analyzed. It is clearly deduced that the transmitted clutch torque is highly influenced by the temperature also for the same throwout bearing maneuvers. Similar experimental results have been showed on a heavy duty truck equipped with a single-plate dry clutch, [53]. Figures 2, 7, 8 therein demonstrate the variation of the transmitted torque due to thermal effects. That variation can be modeled in terms of a thermal dependence of the throwout bearing position.

The map (13) is shown in Fig. 4 for different temperatures and in Fig. 5 for different slip speeds. For the calculation of the equivalent radius it has been considered (2) with (6) and $\mu_d(\theta_c)$ given by a second order interpolating function. The values of $F_{fc}(x_{to}, \theta_c)$ are calculated by substituting (14) into (10).

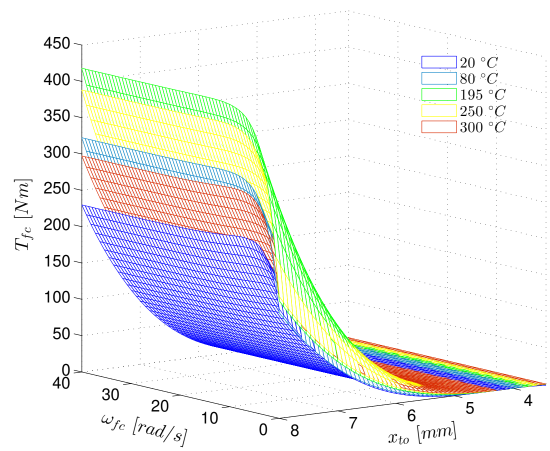


Figure 4: Torque map $T_{fc}$ versus the throwout position $x_{to}$ and the slip speed $\omega_{fc}$ for different temperatures $\theta_c$.

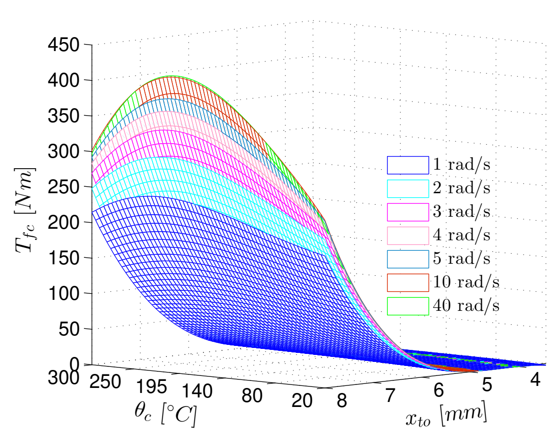


Figure 5: Torque map $T_{fc}$ versus the throwout position $x_{to}$ and the temperature $\theta_c$ for different slip speeds $\omega_{fc}$.

### B. Dynamic thermal models

In order to estimate the temperature(s) of a clutch system, dynamic models with different levels of complexity have been proposed in literature. In [54] a seven states thermal model that includes temperatures of bell housing air, skin, ambient, engine coolant and transmission oil has been suggested. A lower order model can be obtained by considering five temperatures as state variables, i.e. those of the two pressure plates, the two clutch discs and the central plate, and by assuming the air temperature in the housing to be an external input [15]. In that model, the temperature of the clutch disc was assumed equal to the averaged value of the two friction facings temperatures. In [53, 55] it has been used a model with three thermal state variables for capturing the dynamic behavior in an heavy-duty truck dry clutch at slipping conditions. The engine heat can be also taken into account for the determination of the clutch dynamic thermal response [49]. Here below we discuss a simple dynamic thermal model which can be a guideline for catching the major phenomena influencing the clutch control. This model can be easily adapted in order to obtain the other models presented in the literature. Moreover, this class of models allows one to analyze the thermal dynamics in a dual dry clutch as a simple generalization of that for a single clutch.

As in [15, 56], the following major assumptions for the model construction are considered: uniform temperature distribution inside the clutch housing; all slipping energy is converted to frictional heat; 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 with the room air; when the clutch is closed it happens only conductive heat exchange; the conductive heat exchange between the clutch disc and the cushion spring is always existing; the clutch and pressure plate/flywheel temperatures are the same.

Under the above stated assumptions and by introducing the equivalent thermal capacitances for the two clutches, say $C_1$ and $C_2$, and for the body, say $C_b$, a possible thermal model can be written as

$$\begin{aligned} C_1\dot{\theta}_{c1} =& \gamma_1\delta_1(\theta_b - \theta_{c1}) + \gamma_2(1-\delta_1)(\theta_h - \theta_{c1}) + \\ &+ \gamma_3 T_{fc1}\omega_{fc1} \qquad (15a) \\ C_2\dot{\theta}_{c2} =& \gamma_1\delta_2(\theta_b - \theta_{c2}) + \gamma_2(1-\delta_2)(\theta_h - \theta_{c2}) + \\ &+ \gamma_3 T_{fc2}\omega_{fc2} \qquad (15b) \\ C_b\dot{\theta}_b =& -\gamma_1\delta_1(\theta_b - \theta_{c1}) - \gamma_1\delta_2(\theta_b - \theta_{c2}) + \\ &+ \gamma_2(2-\delta_1-\delta_2)(\theta_h - \theta_b) + \\ &+ \gamma_3(T_{fc1}\omega_{fc1} + T_{fc2}\omega_{fc2}), \qquad (15c) \end{aligned}$$

where $\theta$ represent the different temperatures: $\theta_b$ for the pressure-plate and flywheel, $\theta_{c1}$ and $\theta_{c2}$ for the two clutches, $\theta_h$ for the room air. 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 is 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 FE simulations. The model can be simply adapted for the case of a single clutch by considering only (15a) and (15c) with $\delta_2 = 1$, $\theta_{c2} = \theta_b$ and $\omega_{fc2} = 0$.

The model (15) can be further simplified by observing that the body temperature $\theta_b$ has no direct influence on the clutch torque and it might be measurable. Moreover, it is possible to neglect the conductive and convective heat transfers during the slip phase.

The use of further assumptions can simplify the derivation of an heat exchange model [57]. In particular, the heat absorbed by the clutch friction can be neglected if the thermal capacity of the friction lining is much lower than that of the pressure plate and flywheel. Furthermore, the pressure plate and flywheel are usually not affected by radial thermal conductivity, i.e. uniform temperature distribution on the friction disc can be assumed.

## V. Clutch actuators

The clutch actuator is the linkage between the transmission control unit and the clutch assembly. Thus its operation accounts for critical issues and drastically affects the clutch performance. In recent years different actuation systems have been developed to automate clutches: electrohydraulic [58, 59], electropneumatic [60, 61] or electromechanical [62, 63, 64] systems. Both electrohydraulic and electropneumatic actuators are relatively complex systems because they need pump, tank, filter, pipeline and valves. In detail, electrohydraulic actuators are usually employed in cars, whereas electropneumatic actuators are widely used in commercial vehicles [60], as pressurised air is already available for a number of truck services, even though this feature makes harder the modelling task due to the air compressibility [65]. Their working principle is similar to electrohydraulic actuators but in this case the compressed energy of the gas is used as a source of the force transmission. Conversely, an electromechanical actuator involves an electrical motor as power source and one worm gear, screw nut or ball screw as speed reduction mechanism [66]. They have been introduced into the market more recently to replace electrohydraulic systems.

The main advantages of the electromechanical actuation solutions include easier integration into transmission system [67] and potentially better total efficiency due to low losses in electrical motors [62, 68]. The actuator control task is designed as a position control problem or a force control problem according to the clutch architecture, passively closed or actively closed, respectively. Indeed, in dry clutches the transmitted torque is a nonlinear function of the throwout bearing displacement, which is driven by the clutch actuator [66]. Furthermore, dry clutch systems are also affected by other nonlinear effects such as friction coefficient uncertainty, thermal dynamics hysteresis, and wear [65, 69]. So, also a small error in the throwout bearing position could result in an unexpected higher (or lower) clutch torque producing jerks, overheating or even engine halt. That is amplified in dual clutch transmissions where the gear shifting process sees one clutch (on-coming clutch) to be engaged, while another (off-going clutch) needs to be disengaged. Thus, a precise coordination of the on-coming clutch and the off-going clutch is critical, which otherwise will cause undesirable torque interruption and oscillations [70, 71]. In order to overcome the latter drawback, in the last decades numerous control strategies have been suggested to improve management of clutch actuators. Some authors proposed to use a position sensor in order to have a direct measure of the throwout bearing position [72, 73, 74, 75]. Others proposed to adopt a pressure sensor because for both electrohydraulic and electropneumatic actuators it is possible to control the throwout bearing position by controlling the piston chamber pressure [70, 74, 76, 77]. Instead in [78] authors presented a mechatronic add-on system which can be used to implement a force trajectory without replacing the traditional diaphragm spring.

The main disadvantages of those approaches are due to the complexity and feasibility to integrate position and/or pressure sensors into a clutch assembly. Thus, a further approach for overcoming such difficulties is to replace sensors by nonlinear observers [61, 70, 76, 79, 80]. Unfortunately, the nonlinearities listed above require an appropriate mathematical modeling to implement a good observer and consequently to obtain acceptable control performance. Definitely, the main challenge of current development of clutch actuators is represented by high accuracy and quick response. From this point of view the undergoing research is still of wide interest.

## VI. Transmitted torque estimation

The accuracy of the clutch torque control is highly influenced by torque transmission characteristic variations. This has been clearly shown not only for conventional cars but also for dry dual clutches [15] and heavy duty trucks [53, 81]. Due to the difficulties of an on-board torque measurement, a torque estimator is usually adopted [82] and the engagement controller is adjusted accordingly. To this aim a prerequisite is a proper driveline model oriented to the clutch control.

### A. Clutch-oriented driveline models

A model of a transmission equipped with a single dry clutch can be obtained by considering kinematics and dynamics of the different driveline elements [83]. By introducing the rigidity assumptions for the mainshaft, the following second order dynamic model can be written

$$J_f \dot{\omega}_f = -b_f \omega_f + T_e(\omega_f) - T_{fc}(x_{t_0}, \omega_{fc}) \tag{16a}$$

$$J_c(r) \dot{\omega}_c = -b_c \omega_c + T_{fc}(x_{t_0}, \omega_{fc}) - T_L(\omega_c, r) \tag{16b}$$

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 (16) has been omitted.

Analogous procedures can be applied in order to obtain a driveline model for the case of dry dual clutches [84]. A corresponding model can be written as

$$\begin{aligned} J_f \dot{\omega}_f =& - b_f \omega_f + T_e(\omega_f) \\ & - T_{fc1}(x_{to1}, \omega_{fc1}) - T_{fc2}(x_{to2}, \omega_{fc2}) \quad (17a) \\ J_c(r_1, r_2) \dot{\omega}_c =& - b_c \omega_c + r_1 T_{fc1}(x_{to1}, \omega_{fc1}) \\ & + r_2 T_{fc2}(x_{to2}, \omega_{fc2}) - T_L(\omega_c, r_1, r_2) \quad (17b) \end{aligned}$$

where $\omega_{fc1} = \omega_f - r_1 \omega_c$, $\omega_{fc2} = \omega_f - r_2 \omega_c$, $r_1$, $r_2$ are the gear ratios and $T_{fc1}$, $T_{fc2}$ are the torques transmitted by the two clutches, respectively, see [85]. Control-oriented dynamic models of more involved driveline architectures can be obtained by using similar arguments [86].

### B. Torque observers

The parameters of the torque models analyzed in the previous section depend on the coating material of the clutch disc. An alternative model which is independent from the type of friction, is based on the estimation of the torque transmitted by the clutch.

The basic idea for torque observers exploits the driveline dynamic model [12, 87]. From (16) one can write

$$\hat{T}_{fc} = -J_f \hat{\dot{\omega}}_f - b_f \omega_f + T_e(\omega_f) \tag{18a}$$

$$\hat{T}_{fc} = J_c(r) \hat{\dot{\omega}}_c - b_c \omega_c + T_L(\omega_c, r) \tag{18b}$$

where hats indicate the estimated variables. These type of simple observer suffers from noise and uncertainties in the accelerations and torques estimations [88, 89] and parameters identification [90]. Indeed, the equation (18a) 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. A typical solution for getting better performance consists of an observer based on the engine speed error together with a constant model for the transmitted torque, see among others [91]. The dynamic model of this type of observers can be expressed as

$$J_f \dot{\hat{\omega}}_f = -b_f \hat{\omega}_f + T_e(\omega_f) - \hat{T}_{fc} + \ell_1(\omega_f - \hat{\omega}_f) \tag{19a}$$

$$\dot{\hat{T}}_{fc} = \ell_2(\omega_f - \hat{\omega}_f) \tag{19b}$$

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

A Kalman filter with a model structure analogous to (19) is discussed in [92]. Kalman filtering techniques are also used in [93] for the training phase of a neural network which provides an online torque estimation by considering also the actuator dynamics. The actuator model is analyzed in [94] with a different perspective. In particular, the torque estimation is obtained by considering a disturbance observer based on the actuator dynamics and the transmitted torque is determined from the reaction load on the actuator. A similar idea based on the estimation of the load torque is used in [95] where a sliding mode observer is proposed.

Observers have been also proposed for transmissions with dry dual clutches. A good overview of the results on that topic can be found in the recent paper [96]. Fuzzy techniques are adopted in [97] by estimating a state which includes the unknown clutch torque, similarly to the idea expressed by (19b).

The transmitted torque can be estimated online by adapting the parameters of the model $T_{fc}(x_{to}, \omega_{fc})$ expressed by (7) or by using similar quasi steady-state characteristics. Recursive least square techniques have been used for this purpose: in [98] the friction coefficient is estimated under different operating conditions; in [99, 100] three parameters of an exponential-like torque characteristic are estimated. In [101] the kiss point $x_{to}^{cnt}$ is estimated online by exploiting the input shaft speed delay time and the maximum crankshaft acceleration.

## VII. Engagement controls

The engagement maneuver for a dry clutch is a challenging control problem. In AMTs 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, after describing the main control objectives for AMTs, classical and modern clutch control strategies proposed in the last years are presented.

### A. Control objectives

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. Indeed, the energy dissipated during the engagement is given by the integral in time of the dissipated power resulting from the product of the transmitted torque and the slip speed. Clearly this dissipation can be reduced having low transmitted torques when the slip is high and viceversa. By reducing the engagement time duration 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 diriveline discontinuity at lock-up. From (16) with simple algebraic manipulations, see [102] for the details, one can write

$$\dot{\omega}_c(t_f^+) - \dot{\omega}_c(t_f^-) = \frac{J_f}{J_f + J_c(r)} \dot{\omega}_{fc}(t_f^-). \tag{20}$$

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.

Another well known undesired phenomenon during the clutch engagements are self-excited vibrations [103], which are mainly due to the slope of the friction coefficient vs. the slip speed [104]. In [105] the influence of the friction coefficient variations on the judder is investigated and a multivariate statistical method is proposed in order to represent, classify, and predict judder depending on friction in a dry clutch system. A robust controller has been proposed in [106] specifically dedicated to counteract the judder phenomenon.

### B. Classical control strategies

The engagement control issue can be viewed as a tracking problem with two speeds reference signals and can be approached through a decoupling multivariable feedback control strategy [102]. This type of strategy can also include adaptations which allow comfort improvements [107]. Decoupling controllers have been analysed for dry dual clutch transmission systems, as well [108, 109].

The more complex problem of supervising the whole gearshift process is discussed in [110]. A hierarchical supervisor discriminates among five different AMT operating conditions: engaged, slipping-opening, synchronization, go-to-slipping and slipping-closing. Thus a hybrid controller manages the different phases and the transitions among them. Such idea has been further extended in [111] where decoupled and cascaded feedback loops for different engagement phases and controllers have been proposed.

The analysis in [112] focuses on the combination and coordination of classical controllers for engine and clutch during the torque and inertia phases of a dry dual clutch. The cross-coupling of the two feedback loops in dual clutch transmissions must be carefully handled and the reference signals of the clutches profiles should be designed in order to prevent undesirable effects, such as those due to the system nonlinearities [113].

### C. Optimal control

The engagement control problem can be viewed as a constrained optimization problem. This idea was firstly presented and analyzed in [114, 115] where the minimization problem was based on a quadratic function performance index dependent on the slip speed and normal force.

Optimal clutch controller was proposed also for commercial trucks in [116] where a model parameters identification procedure has been considered, as well.

A mixed approach that considers optimal control theory with heuristic choices can be used to cope with the complexity of the problem by combining an open-loop look-up table, aiming to reduce the slipping time, with an observer-based optimal control so to assure the engagement comfort [117, 118].

A robustness analysis of optimal controllers represents a classical line of investigation and we can find those aspects applied to the clutch engagement problem in [119], where the optimal controller exploits a clutch torque observer, and in [120] where a linear quadratic regulator is proposed together with a reduced-order observer.

Robust control techniques based on quantitative feedback theory [121] and $H^\infty$ control [122] have been also applied for the engagement control design.

The control problem of interest can be effectively solved by using model-based approaches. To this aim, the main difficulties are related to the constraints on the engagement process, such as finite-time duration and torques limitations, together with the hybrid nature of the driveline model which changes its structure after the clutch lock-up. Those features suggest the interest for formulating the problems by using the model predictive control (MPC) approach. That was firstly proposed in [123] where the hybrid nature of the control solution was investigated. In [124] the authors compared the MPC performance with those achievable with a piecewise linear-quadratic controller designed by using a piecewise quadratic Lyapunov function. The MPC is able to explicitly take into account the problem constraints [125] and to reject disturbances, such as vehicle mass and road grade [126]. The good performance of MPC have been verified also experimentally [127, 128]. On the other hand, the robustness of the MPC is paid with a large computational burden which might be prohibitive for commercial automotive control units. Therefore suitable strategies for reducing the computational costs and for supporting the model uncertainties, for instance by compensating for the clutch wear [129] or an adaptation to different road conditions [130], should be applied.

Another idea could be to use optimization strategies in order to generate the reference signals which are tracked by means of lower level controllers. This can be done by considering reference speeds, for single [131] and dual [132] clutches.

The application of optimization strategies for dual clutches is more complicated because of the required coordination and optimal torques allocation to be generated through the two clutches actuators [133, 134]. A specific application of the optimal control theory to the case of dual dry clutches has been proposed in [135] by interpreting the performance index in terms of friction work, shock intensity and engine torque. That approach has been combined with a torque observer in [136] while a different approach, based on $H^{\infty}$ and on the optimization of torque and inertia phases in dry dual clutch, is presented in [137].

Optimal control techniques have been applied not only to the mechanical engagement process, but also for controlling the actuator. An experimental validation of an optimal clutch fill process for the synchronization of on-coming and off-going clutches in a clutch-to-clutch system is presented in [138]. The optimal control problem deals with the clutch fill process of the actuator, i.e., the operating conditions when the on-coming clutch moves to the kiss point.

### D. Adaptive, sliding, fuzzy and neural network techniques

Nonlinear control strategies have also been considered by the literature in the field. In [139] the authors proposed a gearshift with speed and torque control designed through the nonlinear backstepping methodology. A Lyapunov based slip speed control was designed in [140], while adaptive control techniques have been proposed and analyzed with Lyapunov stability arguments in [141].

Sliding mode techniques have been recently applied to the engagement problem with different perspectives and goals. A possible approach consists of choosing the sliding surface for the actuator position tracking [142], which can be also done by combining the actuator controller with a feedforward compensation on the clutch normal force [143]. Sliding techniques are also adopted in order to track suitable speeds reference signals which optimize the clutch engagement process [144, 145]. The typical robustness property of the sliding control can be exploited in order to compensate for the nonlinear behavior of the clutch friction torque especially at low slip speeds [146]. In fact, sliding techniques have been used also for the estimation of the transmitted torque [136, 147] and actuator position corresponding to the kiss points [148].

Soft computing techniques seem to be quite interesting for the detection of driver intentions in AMTs. A fuzzy estimator has been presented in the earlier work [149] and artificial neural networks have been shown to be useful for reconstructing the rapid modifications of the driver intentions which influence the clutch engagements [150, 151]. Soft computing techniques can be also combined with dynamic programming for reference trajectories generation [152] and with optimal control in dry dual clutches [153]. Instead, the fuzzy controller presented in [154] is aimed at tracking pre-defined reference trajectories for the slipping and engine speeds during vehicle launch. Similar techniques can be also used for the actuator position tracking with the objective to get better performances and accuracy on the clutch slip control [155].

### E. Numerical co-design

AMT models with single and dual dry clutch have been widely used as numerical co-design tools for clutch design [49, 156] and fault diagnosis purposes [157, 158, 159]. Those models have been implemented within the typical numerical environments dedicated to the simulation of dynamic systems. Matlab/Simulink is used in most cases but also other tools such as Adams [92] and Modelica [160] have been chosen.

The usefulness of numerical co-design techniques is particularly evident in hardware-in-the-loop (HIL) platforms, which allow to test the transmission controls in early stages of their development and also to reduce the time and costs of calibration. In [161] the HIL environment is used for rapid prototyping of an energy-efficient and smooth clutch engagement and to verify the possibility to generate the optimal clutch and engine torque control inputs by means of dynamic programming. The rapid prototyping HIL platform presented in [162] provides a tool for testing transmission control units when the transmitted torque model is linearly dependent on the clutch position determined by the actuator. HIL setups for dry dual clutches are used in [163, 164], where transmission control units are validated during functional tests and failure recovery procedures, and in [136] for the validation of sliding mode torque observers.

The analysis proposed in [165, 166] concentrates on the effects of (multiple) friction elements in changing the structure of the driveline models. The hybrid nature of the AMT model introduces several difficulties for the HIL numerical integration which must satisfy the strict real-time constraints.

## VIII. Conclusion

This overview paper focuses on modeling and engagement control of automated dry clutches for passenger cars. The literature analysis has provided a coherent organization and discussion of most relevant research contributions dealing with models of the friction torque delivered by single and dual dry clutches. Thermal effects, clutch actuator technologies, torque estimation, and engagement control strategies have been investigated. The performances of dry clutches driven by electronic boards in automated manual transmissions and dual-clutch transmissions notably depend on the combination of physical factors as pad material temperature, slip speed, and contact pressure. A significant part of the paper explores the points of strength as well as the limits and drawbacks related to theoretical or experimental analysis routes. Different levels of deepening about clutch torque assessment are often perceivable by going through the overview: simpler models could be sometimes preferred to more complex ones, to provide real time torque estimation in on-board environment, even though delivering lower accuracy. The analysis also encompasses how classical control, optimal control and adaptive techniques have found a challenging field of application over last years, and main results from simulations, hardware in the loop and bench test analyses. This overview effort intends to provide key elements to designers of automated clutches and related control strategies with the purpose of improving knowledge about driveline scheme based on dry clutch managed by automatic control. Such a target can be especially useful to developers of next generation of vehicles as current forecast about automated dry clutches market is expected to gain traction in all countries where higher fuel economy and ease of driving on congested roads continue to influence buying decisions.

## References


[1] R. Fischer, F. Küçükay, G. Jürgens, R. Najork, and B. Pollak, *The Automotive Transmission Book*. Switzerland: Springer Int. Publishing, 2014.

[2] A. Senatore, "Advances in the automotive systems: An overview of dual-clutch transmissions," *Recent Patents on Mech. Eng.*, vol. 2, no. 2, pp. 93–101, 2009.

[3] *The dry double clutch*, Schaeffler Automotive Aftermarket, Apr. 2014.

[4] M. Hoic, N. Kranjcevic, Z. Herold, and J. Deur, "Experimental characterization and modeling of dry dual clutch wear," *SAE Int. J. of Passenger Cars- Mech. Systems*, vol. 7, no. 1, pp. 326–337, 2014.

[5] K. L. Kimmig, D. P. Bührle, D. R. Kolling, R. Daikeler, and M. Baumann, "Highest level of comfort: the dry double clutch faces the challenge," in *10th Schaeffler Symp.*, Detroit, Michigan, Apr. 2014, pp. 230–243.

[6] S. Sfarni, E. Bellenger, J. Fortin, and M. Malley, "Numerical and experimental study of automotive riveted clutch discs with contact pressure analysis for the prediction of facing wear," *Finite Elements in Analysis and Design*, vol. 47, no. 2, pp. 129–141, 2011.

[7] O. I. Abdullah, J. Schlattmann, and A. M. Al-Shabibi, "Stresses and deformations analysis of a dry friction clutch system," *Tribology in Industry*, vol. 35, no. 2, pp. 155–162, 2013.

[8] O. I. Abdullah and J. Schlattmann, "Contact analysis of a dry friction clutch system," *ISRN Mech. Eng.*, vol. 2013, pp. 1–9, 2013.

[9] F. Vasca, L. Iannelli, A. Senatore, and M. Scafati, "Modeling torque transmissibility for automotive dry clutch engagement," in *American Control Conf.*, Seattle, WA, USA, Jun. 2008, pp. 306–311.

[10] A. Senatore, V. D'Agostino, R. D. Giuda, and V. Petrone, "Experimental investigation and neural network prediction of brakes and clutch material frictional behaviour considering the sliding acceleration influence," *Tribology Int.*, vol. 44, no. 10, pp. 1199–1207, 2011.

[11] B. Bhushan, Ed., *Modern Tribology Handbook Vol. 1 - Principles of Tribology*. Boca Raton, FL, USA: CRC Press, 2001.

[12] F. Vasca, L. Iannelli, A. Senatore, and G. Reale, "Torque transmissibility assessment for automotive dry-clutch engagement," *IEEE/ASME Trans. on Mechatronics*, vol. 16, no. 3, pp. 564–573, 2011.

[13] B. Armstrong-Helouvry, P. Dupont, and C. Canudas de Wit, "A survey of models, analysis tools and compensation methods for the control of machines with friction," *Automatica*, vol. 30, no. 7, pp. 1083–1138, 1994.

[14] E. L. Duque, M. A. Barreto, and A. de T. Fleury, "Use of different friction models on the automotive clutch energy simulation during vehicle launch," in *ABCM Symp. Series Mechatronics, Volume 5*, S. C. A. Alfaro, J. M. S. T. Motta, and V. J. D. Negri, Eds. Rio de Janeiro, Brazil: ABCM, 2012, ch. 8, pp. 1375–1389.

[15] G. Pica, C. Cervone, A. Senatore, M. Lupo, and F. Vasca, "Dry dual clutch torque model with temperature and slip speed effects," *Intelligent Industrial Systems*, vol. 2, no. 2, pp. 133–147, 2016.

[16] W. Lhomme, R. Trigui, and A. Bouscayrol, "Switched causal modeling of transmission with clutch in hybrid electric vehicles," *IEEE Trans. on Vehicular Tech.*, vol. 57, no. 4, pp. 2081–2088, 2008.

[17] Z. Zhao, L. He, Y. Yang, C. Wu, X. Li, and J. K. Hedrick, "Estimation of torque transmitted by clutch during shifting process for dry dual clutch transmission," *Mech. Systems and Signal Processing*, vol. 75, no. 1, pp. 413–433, 2016.

[18] M. Hoić, M. Hrgetić, and J. Deur, "Design of a pin-on-disc-type CNC tribometer including an automotive dry clutch application," *Mechatronics*, vol. 40, pp. 220–232, 2016.

[19] X. Lin, J. Xi, and S. Hao, "The calculation model of the friction torque on a dry clutch," *Proc. of the Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. to appear, doi: 10.1177/0954407016686010, 2017.

[20] S. Sfarni, E. Bellenger, J. Fortin, and M. Malley, "Finite element analysis of automotive cushion discs," *Thin-Walled Structures*, vol. 47, no. 4, pp. 474–483, 2009.

[21] L. Guzzella and A. Sciarretta, *Vehicle Propulsion Systems, Introduction to Modeling and Optimization*. London, UK: Springer-Verlag, 2013.

[22] H.-J. Drexl, U. Teichert, and M. Dettmar, "Clutch wear analyzing tech.: A tool for longer clutch life," *SAE Tech. Paper, doi: 10.4271/982848*, 1998.

[23] S. Mauro, G. M. M. Velardocchia, G. Serra, F. Amisano, and G. Ercole, "The influence of the push-plate mechanical characteristic on torque transmissibility in diaphragm spring clutches," in *3rd Aimeta Int. Tribology Conf.*, Vietri sul Mare, Salerno, Italy, Sep. 2002.

[24] S. Sun, Y. Lei, Y. Fu, C. Yang, and et al., "Analysis of thermal load for dry clutch under the frequent launching condition," *SAE Tech. Paper, doi: 10.4271/2013-01-0814*, 2013.

[25] A. Senatore, C. D'Auria, and M. Pisaturo, "Frictional behaviour and engagement control in dry clutch based automotive transmissions," *Vehicle Eng.*, vol. 4, pp. 1–12, 2017.

[26] A. Chaikittiratana, S. Koetniyom, and S. Lakkam, "Static/kinetic friction behaviour of a clutch facing material: Effects of temperature and pressure," *Int. J. of Mech., Aerospace, Industrial, Mechatronic and Manufacturing Eng.*, vol. 6, no. 6, pp. 1048–1051, 2012.

[27] H. Feng, M. Yimin, and L. Juncheng, "Study on heat fading of phenolic resin friction material for micro-automobile clutch," in *Int. Conf. on Measuring Tech. and Mechatronics Automation*, Changsha, China, Mar. 2010, pp. 596–599.

[28] M. Pisaturo and A. Senatore, "Simulation of engagement control in automotive dry-clutch and temperature field analysis through finite element model," *Applied Thermal Eng.*, vol. 93, pp. 958–966, 2016.

[29] O. I. Abdullah and J. Schlattmann, "Finite element analysis of temperature field in automotive dry friction clutch," *Tribology in Industry*, vol. 34, no. 4, pp. 206–216, 2012.

[30] ——, "Thermal behavior of friction clutch disc based on uniform pressure and uniform wear assumptions," *Friction*, vol. 4, no. 3, pp. 228–237, 2016.

[31] M. Pisaturo, A. Senatore, and V. D'Agostino, "Automotive dry-clutch control: Engagement tracking and FE thermal model," in *IEEE 20th Jubilee Int. Conf. on Intelligent Eng. Systems*, Budapest, Hungary, Jun. 2016, pp. 69–74.

[32] ——, "Dry-clutch temperature rise in engagement maneuver: Friction coefficient map and simulation through FE analysis," in *FISITA World Automotive Cong.*, Busan, Korea, Sep. 2016.

[33] Y. Wang, Y. Li, N. Li, H. Sun, C. Wu, and T. Zhang, "Time-varying frictional thermal characteristics research on a dry clutch," *Proc. of the Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 228, no. 5, pp. 510–517, 2014.

[34] O. I. Abdullah and J. Schlattmann, "Effect of band contact on the temperature distribution for dry friction clutch," *Tribology in Industry*, vol. 35, no. 4, pp. 317–329, 2013.

[35] ——, "Finite element analysis for grooved dry friction clutch," *Advances in Mech. Eng. and its Applications*, vol. 2, no. 1, pp. 121–133, 2012.

[36] ——, "Computation of surface temperatures and energy dissipation in dry friction clutches for varying torque with time," *Int. J. of Automotive Tech.*, vol. 15, no. 5, pp. 733–740, 2014.

[37] E. Mouffak and M. Bouchetara, "Transient thermal behavior of automotive dry clutch discs by using Ansys software," *Mechanika*, vol. 22, no. 6, pp. 562–570, 2016.

[38] V. D'Agostino, N. Cappetti, M. Pisaturo, and A. Senatore, "Improving the engagement smoothness through multi-variable frictional map in automated dry clutch control," in *ASME Int. Mech. Eng. Cong. and Exposition*, Houston, Texas, USA, Nov. 2012, pp. 9–19.

[39] A. Myklebust, "Modeling and estimation for dry clutch control," Ph.D. dissertation, Linköping University, Linköping, Sweden, 2013.

[40] M. Hoic, N. Kranjcevic, Z. Herold, J. Deur, and V. Ivanovic, "Experimental characterization and modeling of dry dual clutch expansion effects," *SAE Int. J. of Passenger Cars- Mech. Systems*, vol. 6, pp. 775–785, 2013.

[41] N. Cappetti, M. Pisaturo, and A. Senatore, "Modelling the cushion spring characteristic to enhance the automated dry-clutch performance: The temperature effect," *Proc. of the Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 226, no. 11, pp. 1472–1482, 2012.

[42] M. Pisaturo, C. D'Auria, and A. Senatore, "Friction coefficient influence on the engagement uncertainty in dry-clutch AMT," in *American Control Conf.*, Boston, MA, USA, Jul. 2016, pp. 7561–7566.

[43] M. Akhtar, O. I. Abdullah, and J. Schlattmann, "Transient thermoelastic analysis of dry clutch system," *Machine Design*, vol. 5, no. 4, pp. 141–150, 2013.

[44] S. Zhao, G. E. Hilmas, and L. R. Dharani, "Behavior of a composite multidisk clutch subjected to mechanical and frictionally excited thermal load," *Wear*, vol. 264, no. 11–12, pp. 1059–1068, 2008.

[45] O. I. Abdullah, M. Akhtar, and J. Schlattmann, "Determination of temperatures field for multi-disc clutches," in *69th STLE Annual Meeting and Exhibition*, Lake Buena Vista, FL, USA, May 2014, pp. 1–8.

[46] O. I. Abdullah, J. Schlattmann, and M. Lytkin, "Effect of surface roughness on the thermoelastic behaviour of friction clutches," *FME Trans.*, vol. 43, no. 3, pp. 241–248, 2015.

[47] G. Pica, C. Cervone, A. Senatore, and F. Vasca, "Temperature-dependent torque transmissibility characteristic for automotive dry dual clutches," in *European Control Conf.*, Linz, Austria, Jul. 2015, pp. 2126–2131.

[48] Z. Zhang, S. Xiaohui, and Y. Bin, "Simulation and experiment of characteristics of transmitted torque of dry clutch engagement," *SAE Tech. Paper, doi: 10.4271/2017-01-0402*, 2017.

[49] A. Penta, P. Warule, S. Patel, and L. Dhamija, "Development of high fidelity dynamic model with thermal response for single plate dry clutch," *SAE Int. J. of Passenger Cars - Mech. Systems*, vol. 10, pp. 378–384, 2017.

[50] J. Deur, V. Ivanovic, Z. Herold, and M. Kostelac, "Dry clutch control based on electromechanical actuator position feedback loop," *Int. J. of Vehicle Design*, vol. 60, no. 3-4, pp. 305–326, 2012.

[51] G. Lawrence, G. Mace, N. Bowler, G. Goddard, and D. Morrey, "Measurement of the interfacial plate temperatures within a carbon clutch, and determination of effects upon its friction characteristics," *SAE Tech.*

*Paper, doi: 10.4271/2006-01-3634*, 2006.

[52] M. Hoic, N. Kranjcevic, Z. Herold, J. Deur, and V. Ivanovic, "Modeling of dry dual-clutch axial dynamics," *The Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. to appear, doi: 10.1177/0954407017694964, 2017.

[53] A. Myklebust and L. Eriksson, "Modeling, observability, and estimation of thermal effects and aging on transmitted torque in a heavy duty truck with a dry clutch," *IEEE/ASME Trans. on Mechatronics*, vol. 20, no. 1, pp. 61–72, 2015.

[54] K. Hebbale, F. Samie, and J. Kish, "Dry dual clutch transmission (DCT) thermal model," *SAE Tech. Paper, doi: 10.4271/2015-01-1144*, 2015.

[55] A. Myklebust and L. Eriksson, "Torque model with fast and slow temperature dynamics of a slipping dry clutch," in *IEEE Vehicle Power and Propulsion Conf.*, Seoul, Korea, Oct. 2012, pp. 851–856.

[56] H. Liu, C. Lee, Y. J. Noh, J. H. Cho, H. R. Lee, J. E. Koh, and J. W. Kang, "A study on thermal analytical model for a dry dual clutch," *J. of Drive and Control*, vol. 12, no. 1, pp. 1–8, 2015.

[57] K. Ye, Q. Liu, B. Gao, and H. Chen, "Thermal-friction modeling and analysis for automotive dry clutch systems," in *35th Chinese Control Conf.*, Chengdu, China, Jul. 2016, pp. 2252–2257.

[58] G. Lucente, M. Montanari, and C. Rossi, "Modelling of an automated manual transmission system," *Mechatronics*, vol. 17, no. 2-3, pp. 73–91, 2007.

[59] J. Qu, Y. Zhang, G. Qu, and L. Liu, "Design of automatic clutch system and simulation of engagement process for amt," *Applied Mechanics and Materials*, vol. 69, pp. 67–72, 2011.

[60] B. Szimandl and H. Nemeth, "Dynamic hybrid model of an electro-pneumatic clutch system," *Mechatronics*, vol. 23, no. 1, pp. 21–36, 2013.

[61] H. Langjord, G.-O. Kaasa, and T. A. Johansen, "Nonlinear observer and parameter estimation for electropneumatic clutch actuator," *IFAC Proc. Vol.*, vol. 43, no. 14, pp. 789– 794, 2010.

[62] J. Deur, M. Milutinovic, V. Ivanovic, and H. E. Tseng, "Modeling of a dry dual clutch utilizing a lever-based electromechanical actuator," *ASME J. of Dynamic Systems, Measurement and Control*, vol. 138, no. 9, pp. 091 012.1–091 012.11, 2016.

[63] J. J. Oh, J. Kim, and S. B. Choi, "Design of self-energizing clutch actuator for dual-clutch transmission," *IEEE/ASME Trans. on Mechatronics*, vol. 21, no. 2, pp. 795–805, 2016.

[64] F. Liu, L. Chen, D. Li, J. Zhang, and C. Yin, "Modeling and simulation study of a novel electromechanical clutch actuation system," *Advances in Mech. Eng.*, vol. 9, no. 8, pp. 1–21, 2017.

[65] H. Langjord, T. A. Johansen, S. R. Snare, and C. Bratli, "Estimation of electropneumatic clutch actuator load characteristics," *IFAC Proc. Vol.*, vol. 42, no. 2, pp. 152–157, 2008.

[66] B. Gao, H. Chen, Q. Liu, and H. Chu, "Position control of electric clutch actuator using a triple-step nonlinear method," *IEEE Trans. on Industrial Electronics*, vol. 61, no. 12, pp. 6995–7003, 2014.

[67] S. A. Haggag, "Sliding mode adaptive PID control of an automotive clutch-by-wire actuator," *SAE Int. J. of Passenger Cars - Mech. Systems*, vol. 9, pp. 424–433, 2016.

[68] F. Liu, L. Chen, C. Fang, C. Yin, and X. Zhang, "Modeling and experimental validation of lever-based electromechanical actuator for dry clutches," *Advances in Mech. Eng.*, vol. 9, no. 8, pp. 1–18, 2017.

[69] V. Ivanovic, M. Hoic, J. Deur, Z. Herold, and E. H. Tseng, "Design of test rigs for a dry dual clutch and its electromechanical actuator," *SAE Int. J. of Passenger Cars - Mech. Systems*, vol. 5, pp. 612–620, 2012.

[70] X. Song and Z. Sun, "Pressure-based clutch control for automotive transmissions using a sliding-mode controller," *IEEE/ASME Trans. on Mechatronics*, vol. 17, no. 3, pp. 534–546, 2012.

[71] L. Chen, F. Liu, J. Yao, Z. Ding, C. Lee, C. kuan Kao, F. Samie, Y. Huang, and C. Yin, "Design and validation of clutch-to-clutch shift actuator using dual-wedge mechanism," *Mechatronics*, vol. 42, no. Supplement C, pp. 81 – 95, 2017.

[72] A.-E. Balau, C.-F. Caruntu, and C. Lazar, "Simulation and control of an electro-hydraulic actuated clutch," *Mech. Systems and Signal Processing*, vol. 25, no. 6, pp. 1911–1922, 2011.

[73] J. Horn, J. Bamberger, P. Michau, and S. Pindl, "Flatness-based clutch control for automated manual transmissions," *Control Eng. Practice*, vol. 11, no. 12, pp. 1353–1359, 2003.

[74] H. Langjord and T. A. Johansen, "Dual-mode switched control of an electropneumatic clutch actuator," *IEEE/ASME Trans. on Mechatronics*, vol. 15, no. 6, pp. 969–981, 2010.

[75] R. Nouailletas, E. Mendes, and D. Koenig, "Hybrid modeling and identification of dry friction systems, application to a clutch actuator," *Control Eng. Practice*, vol. 18, no. 8, pp. 904–917, 2010.

[76] H. Langjord, G.-O. Kaasa, and T. A. Johansen, "Adaptive observer-based switched control for electropneumatic clutch actuator with position sensor," in *18th IFAC World Cong.*, Milano, Italy, Aug.-Sep. 2011, pp. 4791–4796.

[77] F. Meng, H. Zhang, D. Cao, and H. Chen, "System modeling and pressure control of a clutch actuator for heavy-duty automatic transmission systems," *IEEE Trans. on Vehicular Tech.*, vol. 65, no. 7, pp. 4865–4874, 2016.

[78] K. Tripathi, "A novel approach for enhancement of automobile clutch engagement quality using mechatronics based automated clutch system," *J. of The Inst. of Eng. (India): Series C*, vol. 94, no. 1, pp. 9–20, 2013.

[79] P. Qian, G. Tao, H. Liu, and X. Li, "Globally stable pressure-observer-based servo control of an electropneumatic clutch actuator," *Proc. of the Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 229, no. 11,

pp. 1483–1493, 2015.
[80] R. Prabel and H. Aschemann, “Backstepping-based sliding mode control of an electro-pneumatic clutch actuator,” in *22nd Int. Conf. on Methods and Models in Automation and Robotics*, Międzyzdroje, Poland, Aug. 2017, pp. 408–413.
[81] J. Thornblad, “Drivetrain modelling and clutch temperature estimation in heavy duty trucks,” Ph.D. dissertation, Linköping University, Linköping, Sweden, 2014.
[82] P. J. Dolcini, C. Canudas de Wit, and H. Béchart, *Dry clutch control for automotive applications*. London, UK: Springer-Verlag, 2010.
[83] E. Galvagno, M. Velardocchia, and A. Vigliani, “Dynamic and kinematic model of a dual clutch transmission,” *Mechanism and Machine Theory*, vol. 46, pp. 794–805, 2011.
[84] Y. Liu, D. Qin, H. Jiang, and Y. Zhang, “A systematic model for dynamics and control of dual clutch transmissions,” *J. of Mech. Design - Trans. of the ASME*, vol. 131, no. 6, pp. 610 121–610 127, 2009.
[85] G. Pica, C. Cervone, A. Senatore, and F. Vasca, “Frictional torque characteristic in automotive dry dual clutches and temperature influence,” in *5th AIT Workshop Tribologia e Industria*, Salerno, Italy, Apr. 2016, pp. 1–6.
[86] J. Rumetshofer, M. Bachinger, M. Stolz, and M. Horn, “A novel approach for model-based control of smooth and lossless gear shifts,” *IEEE Trans. on Vehicular Tech.*, vol. to appear, doi: 10.1109/TVT.2017.2759321, 2017.
[87] R. Amari, P. Tona, and M. Alamir, “Phenomenological model for torque transmissbility during dry clutch engagement,” in *18th IEEE Conf. on Control Applications*, Saint Petersburg, Russia, Jul. 2009, pp. 600–606.
[88] J. J. Oh, S. B. Choi, and J. Kim, “Driveline modeling and estimation of individual clutch torque during gear shifts for dual clutch transmission,” *Mechatronics*, vol. 24, no. 5, pp. 449–463, 2014.
[89] J. J. Oh and S. B. Choi, “Real-time estimation of transmitted torque on each clutch for ground vehicles with dual clutch transmission,” *IEEE/ASME Trans. on Mechatronics*, vol. 20, no. 1, pp. 24–36, 2015.
[90] V. Cerone, D. Regruto, and M. Abuabiah, “Set-membership identification of a dry-clutch transmission model,” in *IEEE Conf. on Control Tech. and Applications*, Kohala Coast, Hawai, Aug. 2017, pp. 1159–1164.
[91] B. Z. Gao, Y. Lei, A. Ge, H. Chen, and S. Kazushi, “Observer-based clutch disengagement control during gear shift process of automated manual transmission,” *Vehicle System Dynamics*, vol. 49, no. 5, pp. 685–701, 2011.
[92] Z. Zhao, L. H. Y. Yang, C. Wu, X. Li, and J. K. Hedrick, “Estimation of torque transmitted by clutch during shifting process for dry dual clutch transmission,” *Mech. Systems and Signal Processing*, vol. 75, pp. 413–433, 2016.
[93] P. Li, T. Jin, and X. Du, “Torque observer modelling for vehicle transmission shifting processing based on neural networks,” in *17th IFAC World Cong.*, vol. 41, no. 2, Seoul, Korea, Jul. 2008, pp. 16 039–16 044.
[94] J. Kim and S. B. Choi, “Nonlinear torque observer for a self-energizing clutch actuator using a reaction torque estimation approach,” *IEEE/ASME Trans. on Mechatronics*, vol. 20, no. 6, pp. 3071–3084, 2015.
[95] L. Li, X. Hu, Z. Chen, J. Song, and F. Muhammad, “A modified predictive functional control with sliding mode observer for automated dry clutch control of vehicle,” *ASME J. of Dynamic Systems, Measurement and Control*, vol. 138, no. 6, pp. 61 005–61 005–10, 2016.
[96] J. Wu, P. D. Walker, J. Ruan, and N. Zhang, “Target torque estimation for gearshift in dual clutch transmission with uncertain parameters,” *Applied Mathematical Modelling*, vol. to appear, doi: 10.1016/j.apm.2017.06.030, 2017.
[97] R. Losero, J. Lauber, T. M. Guerra, and P. Maurel, “Dual clutch torque estimation: Based on an angular discrete domain takagi-sugeno switched observer,” in *IEEE Int. Conf. on Fuzzy Systems*, Vancouver, Canada, Jul. 2016, pp. 2357–2363.
[98] H. Chu, N. Tang, B. Gao, and H. Chen, “Friction coefficient identification of dry clutch in automated manual transmissions,” in *Int. Conf. on Mechatronics and Control*, Jinzhou, China, Jul. 2014, pp. 1115–1119.
[99] A. Tarasow, G. Wachsmuth, J. Lemieux, R. Serway, and C. Bohn, “Online estimation of time-varying torque characteristics of automotive clutches using a control oriented model,” in *American Control Conf.*, Washington, DC, USA, Jun. 2013, pp. 6752–6757.
[100] A. Tarasow, C. Bohn, G. Wachsmuth, and R. Serway, “Online estimation of 3d-torque characteristics of dual clutches using control oriented models,” *IFAC Proc. Vol.*, vol. 46, no. 21, pp. 452–457, 2013.
[101] Z. Zhong, C. Wang, H. Wu, and X. Chen, “A study on self-learning control strategy for AMT clutch key control parameter,” in *ITEC Asia-Pacific*, Beijing, China, Aug. 2014, pp. 1–4.
[102] F. Garofalo, L. Glielmo, L. Iannelli, and F. Vasca, “Smooth engagement for automotive dry clutch,” in *IEEE Conf. on Decision and Control*, Orlando, FL, USA, Dec. 2001, pp. 529–534.
[103] C. Bostwick and A. Szadkowski, “Self-excited vibrations during engagements of dry friction clutches,” *SAE Tech. Paper, doi: 10.4271/982846*, 1998.
[104] A. Crowther, N. Zhang, D. K. Liu, and J. K. Jeyakumaran, “Analysis and simulation of clutch engagement judder and stick-slip in automotive powertrain systems,” *Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 218, no. 12, pp. 1427–1446, 2004.
[105] I. R. S. Gregori, I. Sanches, and C. E. Thomaz, “Clutch judder classification and prediction: A multivariate statistical analysis based on torque signals,” *IEEE Trans. on Industrial Electronics*, vol. 64, no. 5, pp. 4287–4295, 2017.
[106] G. J. L. Naus, M. A. Beenakkers, R. G. M. Huisman, M. J. G. van de Molengraft, and M. Steinbuch,

"Robust control of a clutch system to prevent judder-induced driveline oscillations," *Vehicle System Dynamics*, vol. 48, no. 11, pp. 1379–1394, 2010.

[107] A. Serrarens, M. Dassen, and M. Steinbuch, "Simulation and control of an automotive dry clutch," in *American Control Conf.*, Boston, MA, USA, Jun. 2004, pp. 4078–4083.

[108] C. Ni, T. Lu, , and J. Zhang, "Gearshift control for dry dual-clutch transmissions," *WSEAS Trans. on Systems*, vol. 8, no. 11, pp. 1177–1186, 2009.

[109] M. Zoppi, C. Cervone, G. Tiso, and F. Vasca, "Software in the loop model and decoupling control for dual clutch automotive transmissions," in *3rd IEEE Int. Conf. on Systems and Control*, Algiers, Algeria, Oct. 2013, pp. 1–6.

[110] L. Glielmo, L. Iannelli, V. Vacca, and F. Vasca, "Speed control for automated manual transmission with dry clutch," in *IEEE Conf. on Decision and Control*, Atlantis, Bahamas, Dec. 2004, pp. 1709–1714.

[111] ——, "Gearshift control for automated manual transmissions," *IEEE/ASME Trans. on Mechatronics*, vol. 11, no. 1, pp. 17–26, 2006.

[112] Y. Liu, D. Qin, H. Jiang, and Y. Zhang, "Shift control strategy and experimental validation for dry dual clutch transmissions," *Mechanism and Machine Theory*, vol. 75, pp. 41–53, 2014.

[113] J. Kim, K. Cho, and S. B. Choi, "Gear shift control of dual clutch transmissions with a torque rate limitation trajectory," in *American Control Conf.*, San Francisco, CA, USA, Jun. 2011, pp. 3338–3343.

[114] L. Glielmo and F. Vasca, "Optimal control of dry clutch engagement," *SAE Tech. Paper, doi: 10.4271/2000-01-0837*, 2000.

[115] ——, "Engagement control for automotive dry clutch," in *American Control Conf.*, Chicago, IL, USA, Jun. 2000, pp. 1016–1017.

[116] J. David and N. Natarajan, "Design of an optimal clutch controller for commercial trucks," in *American Control Conf.*, Portland, Oregon, Jun. 2005, pp. 1599–1606.

[117] P. Dolcini, C. Canudas de Wit, and H. Béchart, "Improved optimal control of dry clutch engagement," *IFAC Proc. Vol.*, vol. 38, no. 1, pp. 25–30, 2005.

[118] ——, "Lurch avoidance strategy and its implementation in AMT vehicles," *Mechatronics*, vol. 18, no. 5-6, pp. 289–300, 2008.

[119] P. Dolcini, H. Bechart, and C. Canudas de Wit, "Observer-based optimal control of dry clutch engagement," in *44th IEEE Conf. on Decision and Control*, Seville, Spain, Dec. 2005, pp. 440–445.

[120] B. Wang, B. Gao, and H. Chen, "Adaptive optimal control of starting-up of automated manual transmission," in *11th World Cong. on Intelligent Control and Automation*, Shenyang, China, Jul. 2014, pp. 1723–1728.

[121] L. Glielmo, P. O. Gutman, L. Iannelli, and F. Vasca, "Robust smooth engagement of an automotive dry clutch," *IFAC Proc. Vol.*, vol. 39, no. 16, pp. 632–637, 2006.

[122] G. J. L. Naus, M. A. Beenakkers, R. G. M. Huisman, M. J. G. van de Molengraft, and M. Steinbuch, "Robust control to suppress clutch judder," in *8th Int. Symp. on Advanced Vehicle Control Control*, Kobe, Japan, Oct. 2008.

[123] A. Bemporad, F. Borrelli, L. Glielmo, and F. Vasca, "Hybrid control of dry clutch engagement," in *European Control Conf.*, Porto, Portugal, Sep. 2001, pp. 635–639.

[124] A. C. Van Der Heijden, A. F. A. Serrarens, M. K. Camlibel, and H. Nijmeijer, "Hybrid optimal control of dry clutch engagement," *Int. J. of Control*, vol. 80, no. 11, pp. 1717–1728, 2007.

[125] M. Pisaturo, M. Cirrincione, and A. Senatore, "Multiple constrained MPC design for automotive dry clutch engagement," *IEEE/ASME Trans. on Mechatronics*, vol. 20, no. 1, pp. 469–480, 2015.

[126] X. Lu, P. Wang, B. Gao, and H. Chen, "Model predictive control of AMT clutch during start-up process," in *23rd Chinese Control and Decision Conf.*, Mianyang, China, May 2011, pp. 3204–3209.

[127] R. Amari, P. Tona, and M. Alamir, "Experimental evaluation of a hybrid MPC strategy for vehicle start-up with an automated manual transmission," in *European Control Conf.*, Budapest, Hungary, Aug. 2009, pp. 4271–4277.

[128] B. Gao, J. Hong, T. Qu, B. Wang, and H. Chen, "Linear-quadratic output regulator for systems with disturbance: Application to vehicle launch control," in *31st Youth Academic Annual Conf. of Chinese Association of Automation*, Wuhan, Hubei Province, China, Nov. 2016, pp. 135–140.

[129] L. Li, X. Wang, X. Qi, X. Li, D. Cao, and Z. Zhu, "Automatic clutch control based on estimation of resistance torque for amt," *IEEE/ASME Trans. on Mechatronics*, vol. 21, no. 6, pp. 2682–2693, 2016.

[130] J. Zhang, L. Chen, and G. Xi, "System dynamic modelling and adaptive optimal control for automatic clutch engagement of vehicles," *Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 216, no. 12, pp. 983–991, 2002.

[131] T. Jin, P. Li, and G. Zhu, "Optimal decoupled control for dry clutch engagement," in *American Control Conf.*, Washington, DC, USA, Jun. 2013, pp. 6740–6745.

[132] Z. Zhao, L. He, Z. Zheng, Y. Yang, and C. Wu, "Self-adaptive optimal control of dry dual clutch transmission (DCT) during starting process," *Mech. Systems and Signal Processing*, vol. 68-69, pp. 504–522, 2016.

[133] S. Kim, J. Oh, and S. Choi, "Gear shift control of a dual-clutch transmission using optimal control allocation," *Mechanism and Machine Theory*, vol. 113, pp. 109–125, 2017.

[134] F. Mesmer, S. Hentzelt, T. Szabo, and K. Graichen, "Optimal control of dual-clutch transmissions with multiple groups," in *IEEE Int. Conf. on Advanced Intelligent Mechatronics*, Munich, Germany, Jul. 2017, pp. 1749–1754.

[135] M. X. Wu, J. W. Zhang, T. L. Lu, and C. S. Ni, "Research on optimal control for dry dual-clutch en-

gagement during launch," *Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 224, no. 6, pp. 749–763, 2010.

[136] Z. Zhao, X. Li, , L. He, C. Wu, and J. K. Hedrick, "Estimation of torques transmitted by twin-clutch of dry dual clutch transmission during vehicle's launching process," *IEEE Trans. on Vehicular Tech.*, vol. 66, no. 6, pp. 4727–4741, 2017.

[137] J. Liu, B. Ma, H. Li, M. Chen, and G. Li, "Control strategy optimization for a dual-clutch transmission downshift with a single slipping clutch during the torque phase," *Proc. of the Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. to appear, doi: 10.1177/0954407017704783, 2017.

[138] X. Song, M. A. M. Zulkefli, and Z. Sun, "Automotive transmission clutch fill optimal control: An experimental investigation," in *American Control Conf.*, Baltimore, MD, USA, Jun. 2010, pp. 2748–2753.

[139] J. Fredriksson and B. Egardt, "Nonlinear control applied to gearshifting in automated manual transmissions," in *39th IEEE Conf. on Decision and Control*, Sydney, Australia, Dec. 2000, pp. 444–449.

[140] L. Xiaohui, C. Dong, L. Qifang, and G. Yuan, "A new nonlinear method for launch control of AMT vehicles," in *34th Chinese Control Conf.*, Hangzhou, China, Jul. 2015, pp. 8154–8158.

[141] Y. Dong, X. Jiao, and D. Chen, "Adaptive control for clutch engagement on starting up of vehicle," in *World Cong. on Intelligent Control and Automation*, Beijing, China, Jul. 2012, pp. 1230–1234.

[142] R. Knoblich, C. Guhmann, and J. Beilharz, "Sliding mode position control for automotive dry clutch system," in *9th Int. Multi-Conf. on Systems, Signals and Devices*, Chemnitz, Germany, Mar. 2012, pp. 1–6.

[143] J. J. Oh and S. B. Choi, "Design of hybrid position/force engagement controller for dry dual clutch transmission without diaphragm spring," in *American Control Conf.*, Portland, OR, USA, Jun. 2014, pp. 2618–2623.

[144] Z. Zhao, J. L. Jiang, Z. P. Yu, and T. Zhang, "Starting sliding mode variable structure that coordinates the control and real-time optimization of dry dual clutch transmissions," *Int. J. of Automotive Tech.*, vol. 14, no. 6, pp. 875–888, 2013.

[145] Z. Zhao, H. Chen, and Q. Wang, "Sliding mode variable structure control and real-time optimization of dry dual clutch transmission during the vehicle's launch," *Mathematical Problems in Eng.*, vol. 2014, pp. 1–18, 2014.

[146] S. Haggag and F. Ibrahim, "Dual mode control of an automotive clutch-by-wire system," *SAE Tech. Paper, doi: 10.4271/2013-01-0484*, 2013.

[147] J. Kim and S. B. Choi, "Control of dry clutch engagement for vehicle launches via a shaft torque observer," in *American Control Conf.*, Baltimore, MD, USA, Jun. 2010, pp. 676–681.

[148] J. Oh and S. B. Choi, "Design of dry dual clutch transmission actuator controller for simultaneous kissing point identification using multiple sliding surfaces," in *Int. Conf. on Control Automation and Systems*, Gwangju, Korea, Oct. 2013, pp. 726–732.

[149] H. Tanaka and H. Wada, "Fuzzy control of clutch engagement for automated manual transmission," *Vehicle System Dynamics*, vol. 24, no. 4-5, pp. 365–376, 1995.

[150] L. Li, Z. Zhu, X. Wang, Y. Yang, C. Yang, and J. Song, "Identification of a driver's starting intention based on an artificial neural network for vehicles equipped with an automated manual transmission," *Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 230, no. 10, pp. 1417–1429, 2016.

[151] L. Li, Z. Zhu, Y. Chen, X. Li, and X. Wang, "Engagement control of automated clutch for vehicle launching considering the instantaneous changes of driver's intention," *ASME J. of Dynamic Systems, Measurement and Control*, vol. 139, no. 2, pp. 021 011(1–12), 2017.

[152] Y. Y. Yang, Z. B. Zhu, X. Y. Wang, Z. Chen, and Z. L. Ma, "Optimal launching-intention-aware control strategy for automated clutch engagement," *Int. J. of Automotive Tech.*, vol. 18, no. 3, pp. 417–428, 2017.

[153] Z. G. Zhao, H. J. Chen, Z. X. Zhen, and Y. Y. Yang, "Optimal torque coordinating control of the launching with twin clutches simultaneously involved for dry dual-clutch transmission," *Vehicle System Dynamics*, vol. 52, no. 6, pp. 776–801, 2014.

[154] V. N. Tran, J. Lauber, and M. Dambrine, "$H^\infty$ launch control of a dry dual clutch transmission based on uncertain ts models," in *IEEE Int. Conf. on Fuzzy Systems*, Hyderabad, India, Jul. 2013, pp. 1–8.

[155] R. Losero, T. Guerra, J. Lauber, and P. Maurel, "Electro-mechanical clutch actuator control based on output switched Takagi-Sugeno controller," *IFAC-PapersOnLine*, vol. 49, no. 5, pp. 85–90, 2016.

[156] M. Przybilla, C. Kunze, S. Celik, and S. Dongaonkar, "Combined simulation approach for dry clutch systems," *SAE Tech. Paper, doi: 10.4271/2011-01-1232*, 2011.

[157] G. Qin, A. Ge, and H. Li, "On-board fault diagnosis of automated manual transmission control system," *IEEE Trans. on Control Systems Tech.*, vol. 12, no. 4, pp. 564–568, 2004.

[158] Y. Wang, B. Gao, and H. Chen, "Data-driven design of parity space-based FDI system for AMT vehicles," *IEEE/ASME Trans. on Mechatronics*, vol. 20, no. 1, pp. 405–415, 2015.

[159] S. J. H. Trask, G. J. Jankord, A. A. Modak, B. M. Rahman, G. Rizzoni, S. W. Midlam-Mohler, and G. R. Guercioni, "System diagnosis and fault mitigation strategies for an automated manual transmission," in *ASME Dynamic Systems and Control Conf.*, Tysons, VA, USA, Oct. 2017, pp. 1–10.

[160] H. Huang, S. Novoisky, R. Knoblich, and C. Guhamann, "Multi-domain modelling and simulation of an automated manual transmission system based on modelica," *Int. J. of Simulation and Process Modelling*, vol. 10, no. 3, pp. 253–264, 2015.

[161] X.-Y. Song, Z.-X. Sun, X.-J. Yang, and G.-M. Zhu, "Modelling, control, and hardware-in-the-loop simula-

tion of an automated manual transmission," *Inst. of Mech. Eng., Part D: J. of Automobile Eng.*, vol. 224, no. 2, pp. 143–160, 2010.

[162] H. Quanan, S. Jian, and L. Lei, "Research on rapid testing platform for TCU of automated manual transmission," in *3rd Int. Conf. on Measuring Tech. and Mechatronics Automation*, vol. 3, Shanghai, China, Jan. 2011, pp. 67–70.

[163] D. Olivi, "Development of control-oriented models of dual clutch transmission systems," Ph.D. dissertation, University of Bologna, Bologna, Italy, 2013.

[164] N. Cavina, D. Olivi, E. Corti, and G. Mancini, "Development and implementation of hardware in the loop simulation for dual clutch transmission control units," *SAE Int. J. of Passenger Cars, Electronic and Electrical Systems*, vol. 6, no. 2, pp. 458–466, 2013.

[165] M. Bachinger, M. Stolz, and M. Horn, "Fixed step clutch modeling and simulation for automotive real-time applications," in *American Control Conf.*, Portland, OR, USA, Jun. 2014, pp. 2593–2599.

[166] ——, "A novel drivetrain modelling approach for real-time simulation," *Mechatronics*, vol. 32, pp. 67–78, 2015.