

Experimental Characterization and Modeling of Dry Dual Clutch Thermal Expansion Effects

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ABSTRACT

Thermal expansion of a clutch pack with position-controlled actuation can affect the accuracy of clutch normal torque control, because it causes an increase of the clutch normal force for the given actuator position. The paper presents an experimental characterization and mathematical modeling of the dry dual clutch thermal expansion effects. The experimental data have been collected by using a clutch/transmission test rig. The acquired data point to two separate, mutually opposite thermal expansion effects. The first effect relates to increase of the clutch clearance with temperature growth, while the second one includes decrease of press plate and engagement bearing positions for a given clutch torque and a rising temperature (i.e. the clutch torque rises with temperature growth and a constant actuator position). In order to explain and describe these two effects, a geometry analysis of the clutch, focused on thermal expansion, is carried out. Based on the analysis results, a simplified thermal expansion model is developed, extended with temperature dependence of clutch friction coefficient, and experimentally validated.

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INTRODUCTION

Dual clutch transmission (DCTs) systems are being increasingly introduced into modern vehicles as a solution which combines the driving comfort of conventional torque converter transmissions with the fuel efficiency of a manual transmission [1]. DCTs with wet clutches have found applications in numerous vehicles so far (e.g. [2, 3]). The dry-clutch based solutions have emerged on the market recently [4, 5] aimed at the fuel economy improvement due to absence of drag associated with the clutch viscous friction and transmission oil pump drive. They are however, limited to application in smaller vehicles due to limitations related to the durability, thermal capacity, and stability of dry friction characteristics [6].

Disregarding the type of clutch, the accurate torque control is desired for the purpose of development of vehicle control systems [7,8,9]. It has been demonstrated in [10] that the torque of a dry dual clutch equipped with an electromechanical actuator can significantly vary with the friction interface temperature. This can be a consequence of (i) temperature-dependence of clutch friction coefficient

(COF) and/or (ii) thermal expansion of clutch assembly components causing the change of clutch normal force.

This paper presents results of an experimental characterization and mathematical modeling of the influence of clutch thermal effects on the torque control accuracy for the electromechanically actuated dry dual clutch. The experimental data for a wide range of operating conditions have been collected by using the clutch/transmission test rig described in [10]. The developed thermal expansion effect model represents an extension of the previously developed thermal dynamics model [11]. The overall thermal model is to be integrated in the previously proposed and experimentally validated physical clutch model [12], which was focused on the clutch actuator and clutch axial dynamics.

SYSTEM DESCRIPTION

The considered dual clutch system schematic is presented in Fig. 1. Torque from the engine shaft is transferred over the clutch yoke (YO) to the double-sided flywheel (FW). Onwards, the torque is transferred to the pressure plates (PP1 and PP2) by means of leaf-type return springs. Additionally, these leaf-springs disengage/open the clutch when the clutch

actuator is not engaged. The friction plates FP1 and FP2 are connected the transmission input shafts; FP1 to the full shaft which rotates inside a hollow shaft connected to the FP2. An electrical servomotor engages each clutch through a lever-based mechanical actuator and an associated clutch diaphragm spring. The diaphragm spring lever boosts the actuator engagement bearing force by a diaphragm spring lever transformation ratio (equals 4 for clutch 2) to generate the axial force on the press plate which represents the clutch normal force. Details on the clutch design and clutch dynamics modeling can be found in [12, 13].

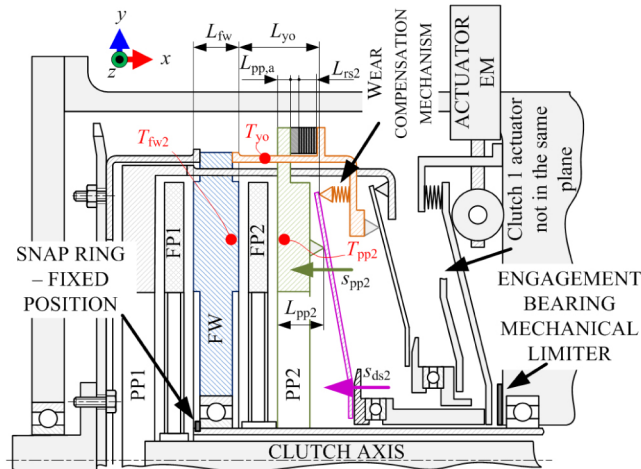


Figure 1. Dry dual clutch schematics.

TEST RIG

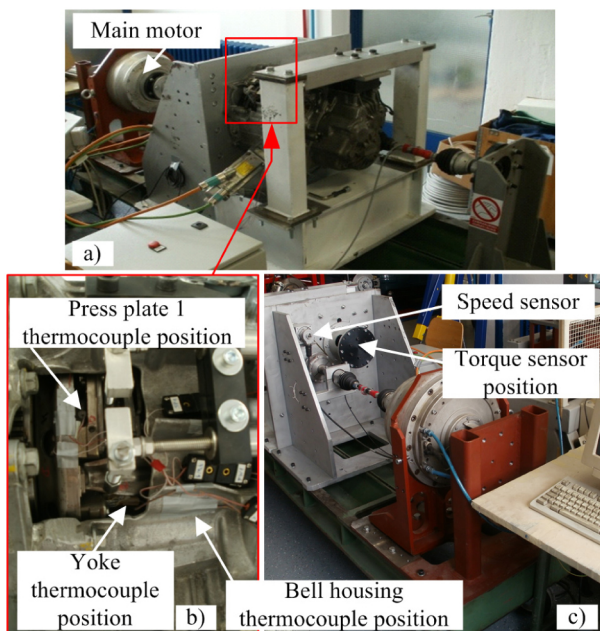


Figure 2. Photographs of clutch/transmission test rig [10].

The experimental investigation and model validation are conducted by using the clutch/transmission test rig [10] shown in Fig. 2. In the given rig configuration, which is aimed at precise clutch characterization experiments, the flywheel (and the clutch press plates) is grounded by means of a strain-gauge reactive torque sensor (see Fig. 2c). A high-torque electric servo-motor drives the clutch friction plates through one of the half-shafts (second half-shaft is grounded) and the transmission. The chosen gear and the given motor speed/torque range provide the full clutch torque range (approx. 150 Nm) and the slip speed range of 150 rpm.

TEST PROCEDURE

Experimental characterization of the thermal expansion effect was conducted by recording the clutch torque vs. actuator position static curves throughout the whole clutch temperature region (Fig. 3). Note that all values on the plots are given in the normalized (per unit, p.u.) form, where e.g. the clutch torque equal to 1 corresponds to the nominal torque.

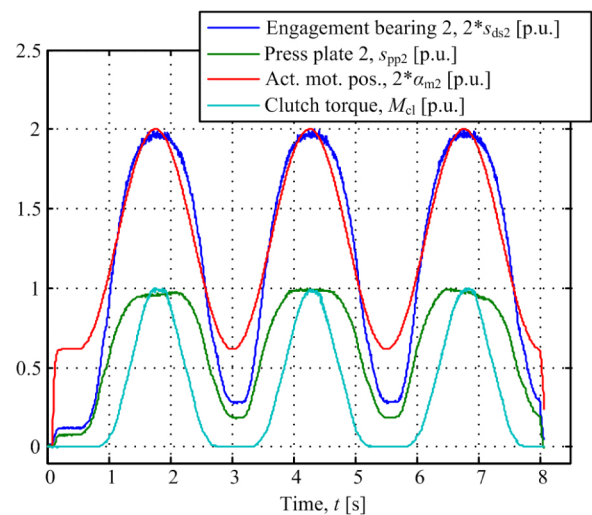


Figure 3. Main parameters variables recorded during a single identification run.

The recording was carried out during the clutch cooling period, because the clutch temperature then decreases gradually and slowly. The clutch was initially self-preheated up to 200°C while rotating at constant slip speed and torque. Starting from 200°C, and down to 40°C, the clutch static curves were recorded at the equidistant temperature intervals of approximately 10°C. An additional recording was conducted before the heating up procedure to cover the room temperature region (approx. 25°C).

In each recording, the actuator motor position was changed in a sinusoidal manner (3 periods, see Fig. 3), while the clutch slip speed was kept constant. While the main recorded static curve corresponded to the clutch torque vs. actuator position, other influencing variables such as press

plate and engagement bearing positions were recorded, as well.

Two-side (zero-phase lag), fourth-order Butterworth filter with the cut-off frequency of 5 Hz has been used to effectively filter the recorded signals. The obtained static curves include a hysteresis caused by friction effects, which can be removed by averaging the torque values obtained at the same, rising and falling actuator positions (see Fig. 4). As each run lasts 8 s (see Fig. 3), the temperature drop during a single run is sufficiently small to justify using the mean press plate temperature as a temperature value which can represent the entire run.

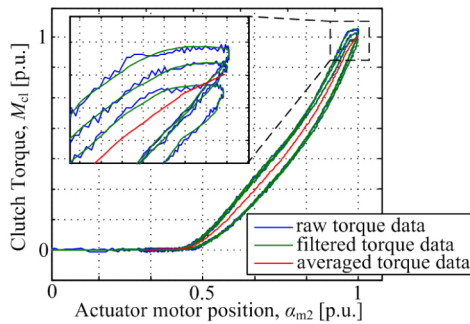


Figure 4. Recorded, filtered, and averaged clutch static curves.

TEST RESULTS

The main experimental results are shown in Fig. 5. The engagement bearing position (Fig. 5 c and d) is reduced by diaphragm spring lever transformation ratio to press plate position. It can be clearly seen that both press plate- and engagement bearing-related torque static curves (as well as the actuator position-related one) are consistently shifted up with the clutch temperature growth. That is, at higher temperatures the same torque is achieved for lower press plate displacement. Or, for the same displacement the torque grows with temperature. This effect can be caused by thermal expansion of the clutch pack, i.e. the normal force grows as the clutch pack expands.

However, the clutch clearance displacement behaves in the opposite manner. That is, the clearance zone is wider for higher temperatures (Fig. 5 b, c and d), i.e. at higher temperatures the torque starts to be developed at greater press plate positions. This effect is in an apparent collision with the basic clutch pack thermal expansion manifestation.

Between these two distinctive phases (clearance and torque generation), a crossover point can be observed. It will be shown in the next section that the thermal effect may indeed manifest itself in different ways for the two distinct phases.

Furthermore, a detail of the static curves from Fig. 5, which is shown in Fig. 6a, shows that the initial position of the press plate (for full disengaged actuator) significantly varies with the temperature, where the initial position shift

direction and size is in accordance with the shift of the torque build-up point (the “touchdown” point; cf. Figs. 5b and 6a). However, unlike the press plate position (Fig. 6a), the initial engagement bearing position (see Fig. 6b) shows virtually no change with respect to temperature, which is apparently because of a stiff contact at an engagement bearing mechanical limiter [13].

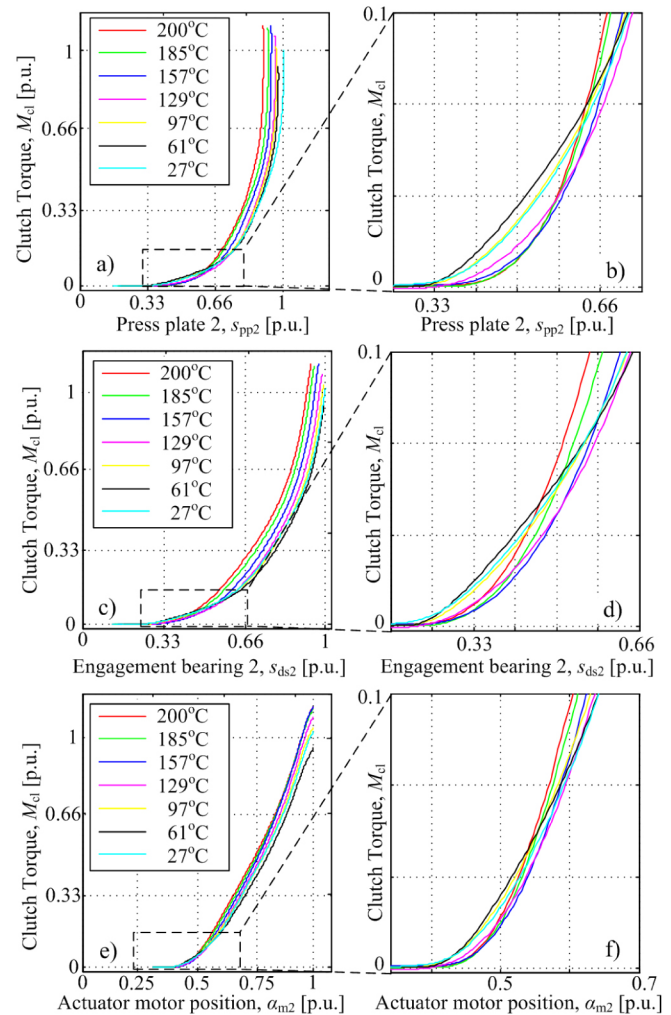


Figure 5. Experimentally recorded clutch torque vs. position static curves for full range of (press plate) temperatures.

Finally, it should be observed that, independently of the clutch temperature, the press plate and engagement bearing positions are linearly related to each other up to certain torque value after which the press plate position exhibits some sort of saturation albeit both clutch torque and the actuator motor position increase further (cf. subplots in Fig. 5 and cf. Figs. 7). Consequently, the fixed diaphragm spring lever transformation ratio no longer applies for higher torques. This leads to the conclusion that at higher normal forces (for higher positions), a significant elastic deformation occurs, whether on the diaphragm spring lever or on one of its support points (see Fig. 1). Based on a visual inspection of

the clutch components, it has been hypothesized that the majority of compliance is associated with the yoke support point.

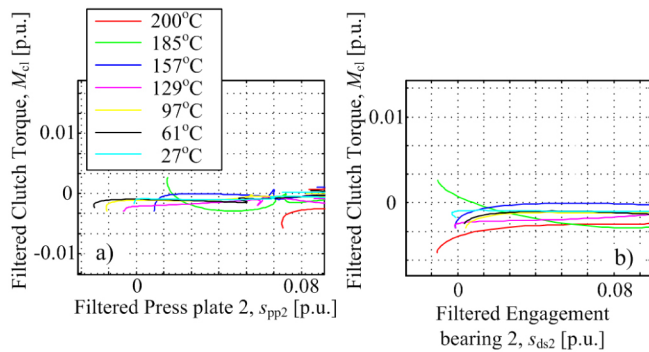


Figure 6. Very initial part of press plate (a) and engagement bearing (b) position-related static curves from Fig. 5.

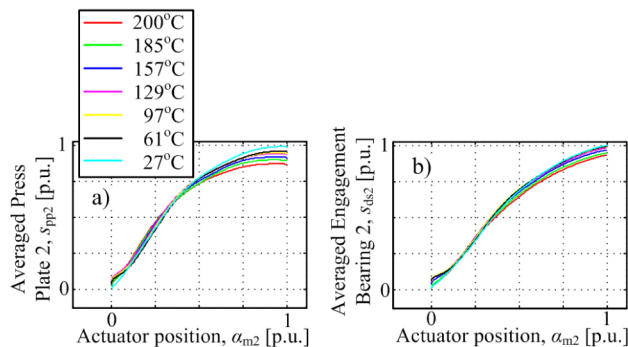


Figure 7. Experimentally recorded, filtered and averaged, pressplate (a) and engagement bearing (b) vs. actuator position static curves for full range of (press plate) temperatures.

The above results clearly indicate that there is a considerable influence of the thermal expansion effect on the clutch behavior both in the clearance and torque transfer phases. Namely, if the temperature dependence of transferred clutch torque [10] were solely due to the COF temperature dependence (and not due to the normal force temperature dependence), there would be no reason for any temperature “drift” of engagement bearing/press plate position for the constant actuator position.

THERMAL EXPANSION ANALYSIS

The experimental data have pointed to two separate, mutually opposite thermal expansion effects. In order to explain and describe these two effects, a geometry analysis of the clutch, focused on thermal expansion, has been carried out.

The modeling is first conducted separately for the two distinctive phases: (i) the clutch clearance phase and (ii) the torque transfer phase including the effect of temperature

dependent coefficient of friction, in order to try to predict the opposite character of thermal expansion for the two cases (see previous section). Once both effects are modeled and the models are validated, a crude combined model is derived and validated, as well.

There are two possible fixed points on the clutch, which are related to the thermal expansion model (see Fig. 1). The definite fixed point is the snap ring which prevents axial movement of flywheel towards the “engine” side. The engagement bearing position (s_{ds}) may also be initially treated as a fixed point, at least in the case of disengaged actuator when the engagement bearing is leaned to the mechanical limiter.

Analysis of Thermal Expansion in Clearance Mode

The clutch clearance is the total distance between the press plate 2 (PP2) and the flywheel (FW), which is not occupied by friction plate (Fig. 1). The test results have indicated that the clearance increases with increasing clutch temperature (Figs. 5 and 6). It will be shown below that the temperature dependence of clearance can be explained by a bending effect in the return spring, which occurs when the return spring is exposed to the temperature difference between the press plate and flywheel connection points.

The analysis is based on two assumptions. The first one is that, as already mentioned, both flywheel and initial engagement bearing positions are independent of the temperature (both are leaned against fixed points). The second assumption is that the return spring bending axial force should be large enough to overcome the support point reaction/stress force, thus compressing the yoke support point. This assumption is based on the fact that the initial engagement bearing position does not change with the temperature (Fig. 6b), while the press plate moves towards the yoke support point as a consequence of the bending effect (Fig. 6a).

Consequently, in the clearance mode the clutch pack may be considered as an isolated “floating” system disconnected from the diaphragm spring and support point, as illustrated in Fig. 8. The return spring bending force is balanced by the engagement bearing limiter reactive force, which needs to be overcome by the actuator when the clutch is started to be engaged (a transition from the clearance mode to the torque transfer mode).

Since the friction plate has a relatively low thermal conductivity, its internal body temperature is relatively low compared to the friction interface temperature. It is, therefore, assumed that friction plate does not exhibit significant thermal expansion.

Therefore, the total thermal shift of the clearance is defined as a relative change of the press plate 2 position with respect to snap ring (the fixed point of the system). From this fixed point, the expected thermal expansion of flywheel (ΔL_{fw}) and yoke (ΔL_{yo}) pushes the press plate 2 away, thus