

Experimental Characterization and Modeling of Dry Dual Clutch Wear

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ABSTRACT

Clutch wear is dominantly manifested as the reduction of friction plate thickness. For dry dual clutch with position-controlled electromechanical actuators this affects the accuracy of normal force control because of the increased clutch clearance. In order to compensate for the wear, dry dual clutch is equipped with wear compensation mechanism. The paper presents results of experimental characterization and mathematical modeling of two clutch wear related effects. The first one is the decrease of clutch friction plate thickness (i.e. increase of clutch clearance) which is described using friction material wear rate experimentally characterized using a pin-on-disc type tribometer test rig. The second wear related effect, namely the influence of the clutch wear compensation mechanism activation at various stages of clutch wear on main clutch characteristics, was experimentally characterized using a clutch test rig which incorporates entire clutch with related bell housing. Finally, the previously proposed and experimentally validated physical clutch model, which was focused on the clutch actuator and axial dynamics, is extended to capture both wear related effects as a further step towards a more comprehensive overall clutch dynamics model.

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INTRODUCTION

Traditionally, increased driving comfort was reserved for vehicles with conventional torque converter automatic transmissions, while improved fuel efficiency was achieved by using manual transmissions. The goal of using dual clutch transmissions (DCTs) is to combine the best characteristics of both solutions [1, 2]. DCTs are based on standard friction plate clutches and can be found both as wet [3, 4] and dry [5, 6] clutch-based solutions, with dry dual clutches promising better fuel efficiency due to the absence of drag loss when in the released state [7]. However, due to limitations related to the durability, thermal capacity, and stability of dry friction characteristics [8], they are usually limited to application in smaller vehicles.

Disregarding the type of clutch, the accurate clutch torque control is desired for the purpose of development of transmission/vehicle control systems [9, 10, 11]. It has already been demonstrated in [12] that the torque of a dry dual clutch equipped with an electromechanical actuator significantly varies with clutch components thermal expansion due to its influence on the clutch normal force. Since maximum expected decrease of clutch friction material thickness caused by clutch wear exceeds the clutch components thermal expansion, it may be expected that the clutch friction plate wear would also

significantly influence the torque transfer characteristic throughout clutch exploitation. However, the clutch is equipped with wear compensation mechanism which is intended to compensate for the increase of clutch clearance caused by wear. It should be noted that, due to a teeth shaped trigger component, the mechanism compensates for the increase of clearance in a stepwise manner.

A direct, control-based compensation of various clutch wear effects is difficult due to the problem of inability to directly measure the friction plate thickness during clutch exploitation. Therefore, the wear is usually estimated using various models and used for various purposes including clutch temperature estimation during on-line clutch control [13] or determination of wear influence on the diaphragm spring lever transfer ratio during clutch design [14].

This paper presents results of experimental characterization and mathematical modeling of two clutch wear related effects. The first one is the decrease of clutch friction plate thickness (i.e. increase of clutch clearance) which is described using friction material wear rate as a function of the clutch dissipated energy and the friction interface temperature. Experimental characterization is conducted using a pin-on-disc type tribometer test rig, where the wear is measured from the

difference in friction material sample mass. The second wear related effect is the influence of the clutch wear compensation mechanism activation at various stages of clutch wear on main clutch characteristics including the torque static curve. The experiments were conducted on a clutch test rig which incorporates entire clutch with related bell housing and enables measurements of relevant variables such as clutch components axial positions and clutch torque.

The proposed wear model is integrated in the previously proposed physical clutch model [15, 16], which was focused on the clutch actuator and clutch axial dynamics, and the overall model is validated against experimental data.

SYSTEM DESCRIPTION

Fig. 1 presents the considered dry dual clutch system schematics. From the engine shaft, the torque is transferred over the clutch yoke (YO) to the double-sided flywheel (FW). Torque is onwards transferred to the pressure plates (PP1 and PP2) by means of leaf-type return springs. The leaf-springs also disengage/open the clutch when the clutch actuator is not active. The friction plates FP1 and FP2 are connected to the transmission input shafts, where the FP1 full shaft rotates inside a hollow shaft connected to the FP2.

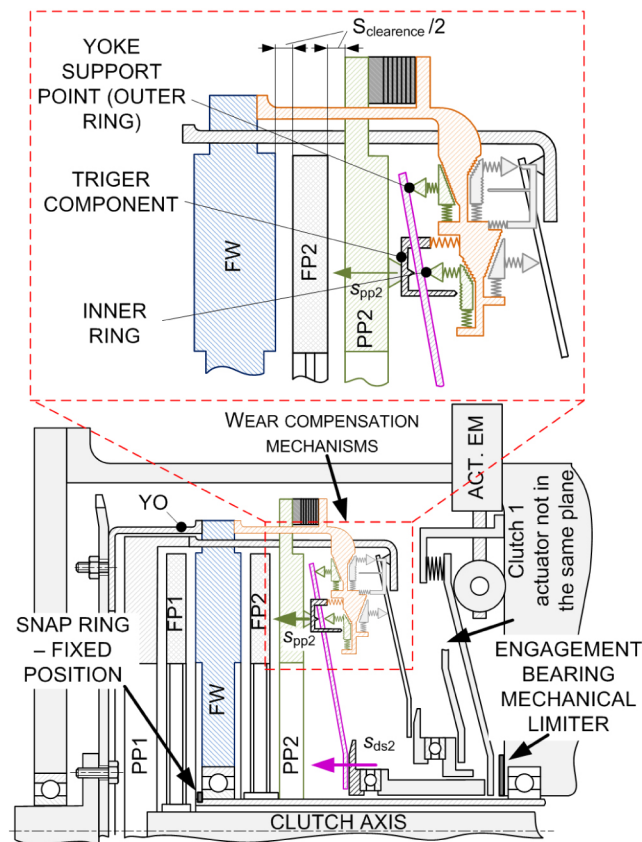


Figure 1. Dry dual clutch schematics.

Each clutch is engaged by means of an electrical servomotor through a lever-based mechanical actuator and an associated clutch diaphragm spring. The actuator engagement bearing force input is boosted through a diaphragm spring lever transformation ratio to generate the axial force on the press plate. Additional details on the clutch design and clutch dynamics modeling can be found in [15, 16].

Each clutch is equipped with a wear compensation mechanism which comprises of two separate rings sliding against the ramps formed on the clutch yoke in circular direction (presented on Fig. 1 as radial ramp for the sake of simplified description). The rings are being pressed by springs to shift forwards up the ramps (thus pushing the press plate towards the flywheel), and restrained by a trigger component acting as a position sensor on the clutch diaphragm spring lever press plate support point. In this way, the trigger has the same displacement as the press plate.

In the case of worn friction plate, during clutch engagement the press plate displacement is larger than normal (due to smaller thickness of compressed material, i.e. larger clutch clearance) which, for maximum actuator position, causes the trigger to move forward sufficiently to release the teeth on the inner ring. Consequently, the inner ring rotates until, due to its axial motion caused by the ramp, its support point again reaches the diaphragm spring.

Afterwards, when the clutch disengages, the diaphragm spring lever leans against the inner ring support point which is now positioned closer to the flywheel, thus releasing the yoke support point. This enables the outer ring spring to push the outer ring which sets the yoke support point to a new position closer to the flywheel thus reducing clutch clearance with the aim of restoring the initial clutch torque vs. press plate position characteristics.

FRICITION MATERIAL WEAR RATE MODELING

Dry clutch wear relates to an adhesive wear of two smooth dry surfaces which is commonly modeled using the Archard's law [17, 18, 19]. Archard's law postulates that the wear volume V_w is proportional to the material hardness H , the contact load F and the relative surface sliding distance L :

$$V_w = K \left(\frac{FL}{3H} \right) \quad (1)$$

where K is the wear coefficient. The contact load F is equal to the ratio of tangential force F_t and coefficient of friction μ :

$$F = \frac{F_t}{\mu}$$

while the tangential force multiplied by sliding distance equals the dissipated energy E_{dis} :

$$E_{\text{dis}} = F_t L \quad (3)$$

By incorporating Eqs. (2) and (3) into Eq. (1), the following expression is derived:

$$V_w = \frac{K}{3H\mu} E_{\text{dis}} \quad (4)$$

Wear coefficient, material hardness, and coefficient of friction are generally temperature dependent (see [19] and [12], respectively). Consequently, it is possible to replace them with a single temperature-dependent coefficient (including the scaling factor 3) named wear rate w . In that case, the volume which is worn off, V_w , may be calculated from the known value of wear rate (which is a function of contact surface temperature T) and the dissipated energy E_{dis} :

$$V_w = w(T) E_{\text{dis}} \quad (5)$$

Eq. (5) indicates that, for the known dissipated energy obtained from the clutch slip speed and estimated clutch torque, the total wear can be estimated once the wear rate temperature dependency is known. Due to the highly complex physical background, wear rate is usually experimentally characterized either by wearing out the entire friction plate or by running a friction sample on a tribometer machine [20]. In either case, wear can be measured either from the difference in thickness [20] or, more traditionally, from the difference in mass [21].

In this paper, the tribometer developed at the University of Zagreb [22] has been used to determine friction material wear rate, where the wear was measured from the difference in friction sample mass.

Tribometer Test Rig

The tribometer is of the pin-on-disc type, where the friction sample (20×20 mm size, see Fig. 2a) and the disc plate are cut from the real dry dual clutch friction plate and press plate, respectively.

The tribometer disc rotational motion and the sample vertical motion (Fig. 2b) are controlled by means of servo drive systems. A three-axis piezoelectric force sensor is used to measure the sample vertical force, which is fed to a vertical force feedback controller, as well as the sample tangential force that is used to calculate the COF. The tribometer includes a heater installed into the rotating disc and connected to a variable-voltage power supply through a slip ring system. Another slip-ring component is used to transfer the disc temperature signal to the temperature feedback controller.

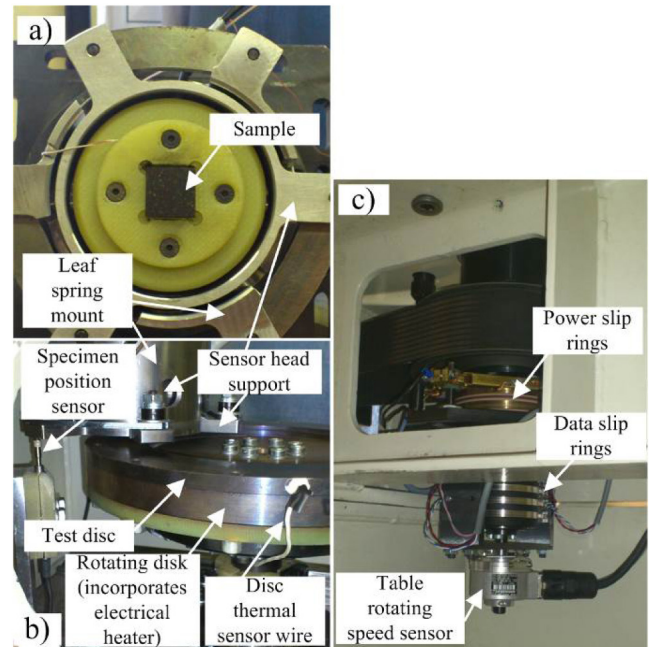


Figure 2. Photographs of pin-on-disc tribometer used to collect the dry friction material COF data.

Tribometer Test Procedure

After the sample was initially weighted, wear procedure was conducted at a constant normal force (70 N), slip speed (1000 rpm) and desired temperature for a planned test period (see Table 1). At higher temperatures, the wear period was decreased since it was expected that wear rate would increase with temperature thus generating the same wear for smaller dissipated energy. The slip speed was reduced to 750 rpm at the lowest temperature of 85°C, because at higher slip speed the friction heat was causing the increase of temperature above this particular target temperature. For the higher temperatures from Table 1, in addition to friction heat the installed electric heater was needed to maintain the target temperature (in a closed-loop control manner). The sample mass was measured after each wear procedure.

Table 1. Tribometer test procedure.

Target temperature [°C]	Total wear time [min]	Slip speed [rpm]
85	145	750
115	105	1000
155	75	1000
195	60	1000
235	30	1000

Worn volume was calculated from difference in sample mass and friction material density which was determined from the difference in mass/volume of water with and without

submerged friction material sample measured on the appropriate scale (in accordance with ASTM D 792 B standard).

Tribometer Test Results

Tribometer wear experiments results are presented in Fig. 3 in a normalized (per unit, p.u.) form. The numbers next to the dot symbols indicate the order in which the experiments were recorded. The experimental results are interpolated in Fig. 3 by a 2nd order polynomial curve. The point related to the wear rate estimate at 260°C is introduced for the purpose of later comparison with the actual clutch wear rate determined at the clutch test rig (see next section).

As can be seen in Fig. 3, wear rate experiments were conducted three times at each temperature level (except at lowest temperature, where two experiments were carried out). During the first three experiments and then again in the 12th experiment, wear was significantly higher compared to all later results. This might indicate that there is some difference in the surface layer of the material, as consequence of the surface break-in process.

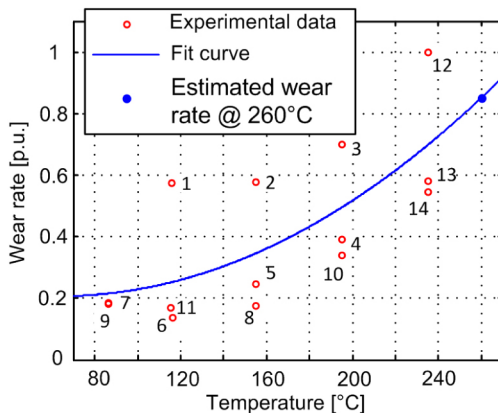


Figure 3. Normalized wear rate vs. temperature dependence recorded on pin-on-disc tribometer.

WEAR COMPENSATION MECHANISM MODELING

In order to experimentally characterize the influence of wear compensation mechanism on clutch characteristics, clutch experiments have to be conducted at various stages of clutch wear. One possibility for obtaining such data is to conduct measurements during clutch wear down, preferably over the entire lifespan.

Based on the wear rates obtained in the previous section, as well as on experimentally determined, relatively low value of clutch (dissipated) power required for reaching a maximum allowed steady-state friction interface temperature, the continuous operation time for total clutch wear was estimated to be equal to several months. To reduce the testing time to a

reasonable value, it was decided that only partial ("small") wear would be tested. For full ("deep") wear, the clutch would be disassembled and material would be removed by machining.

Since the particular wear compensation mechanism is such that the trigger component and ramps are tooth-shaped, activation of the mechanism is a discrete-event process. Consequently, clutch characteristics should show jumps in achieved values of clutch torque as well as engagement bearing and press plate position. This means that experiments related to wear compensation mechanism of this type have to include at least one such jump, i.e. at least one wear compensation mechanism activation.

Ideally, the clutch characteristics should be recorded just prior and post to the wear mechanism activation. Since precise moment of mechanism activation is difficult to predict, clutch characteristics should be recorded as often as possible. Therefore, precision of experimentally obtained data would increase with the frequency of clutch characteristics experiments. On the other hand, more frequent recording of clutch characteristics causes longer pauses between actual wear (note that they should be recorded at the room temperature to be comparable with the previous experimental work conducted on this clutch [15]), thus increasing the necessary total time of wear experiments.

Clutch Test Rig

Fig. 4 shows schematic and photographs of the transmission test rig in Configuration 2 [23] used for clutch small wear experiments. The flywheel is directly driven by using the high-torque electric motor while both shafts are grounded. In the absence of a rotary torque sensor, the clutch torque was estimated from the motor current required to maintain a constant target motor/clutch speed. The clutch friction interface temperature and the press plate positions cannot be measured in this rig configuration, because it imposes rotation of flywheel and press plates in rather tight bell housing.

Configuration 2 was chosen due to its suitability for long term experiments. Namely, while Configuration 1 characterized with non-rotating flywheel/press plates (see Fig. 5) does enable more precise clutch characterization, the unnatural powering of the clutch through one half shaft with the other half shaft being grounded presents significant nonsymmetrical load on differential, which should not be applied for long periods of clutch wear. Additionally, since the main clutch components rotate in Configuration 2, it was expected that increased cooling of clutch components would allow clutch wear at higher power levels, thus shortening the entire testing period.