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MODELS FOR PRESSURE CONTROL OF AUTOMATED DRY CLUTCHES: TEMPERATURE INFLUENCE ON FRICTIONAL AND ELASTIC BEHAVIOUR

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ABSTRACT: Automated Manual Transmissions (AMTs) systems are generally constituted by a dry or wet clutch assembly and a multi-speed gearbox, both equipped with electro-mechanical or electro-hydraulic actuators, which are driven by a control unit, the transmission control unit (TCU). In this transmission type the quality of the vehicle propulsion as perceived by the driver is largely dependent on the quality of the control strategies. This paper aims at investigating the influence of the temperature on the engagement performance of an actively closed dry clutch.
KEYWORDS: Automated manual transmissions, Dry clutch transmissibility, Frictional coefficient map, Temperature effect.

INTRODUCTION

Nowadays, the development of the Automated Manual Transmission (AMTs) has led to remarkable increment of their performance respect the manual ones. In fact, AMTs allow strong improvement in terms of safety, comfort, reliability, shifting quality, and driving performance, together with reduction of fuel consumption and pollutant emissions [1-6]. In automotive drivelines, the goal of the clutch is to smoothly connect two rotating masses, the flywheel and the transmission shaft, which rotate at different speeds, to allow the torque transfer generated by the engine to the driveline [7]. In the Figure 1, an actively closed dry clutch system suitable for pressure-controlled engagement is shown.

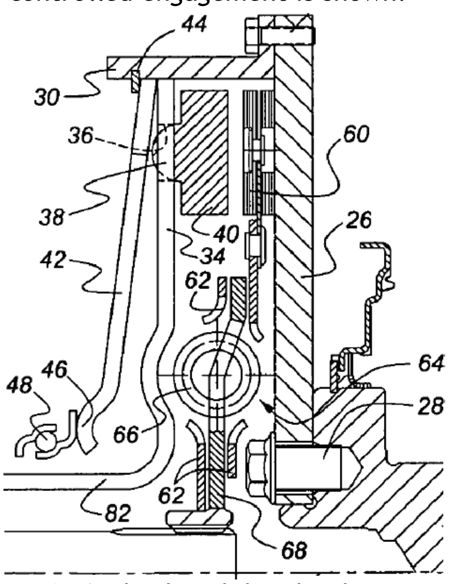

Figure 1: Actively closed dry clutch system (open position) [3]: 48, throwout bearing; 42, lever spring; 40, pressure plate; 60, clutch disc; 26, flywheel.

In AMTs the clutch engagement during gearshift is managed by an actuator which is driven by a control unit, the transmission control unit (TCU). Consequently, high importance is assumed by the control scheme implemented in the TCU. In fact, to reach high performances is fundamental implement a dependable control scheme in order to obtain reduction of fuel consumption, short gearshift time, reduction of facing wear and comfort.

In order to achieve these objectives on clutch engagement, several models of control strategies for dry clutches in AMTs have been recently proposed in the literature, e.g., classical controller [1], optimal control [8, 9], predictive control [10, 11], decoupling control [7], and robust control [12, 13]. However, effective AMTs controllers are difficult to be designed without having a physical model of the clutch-torque transmissibility characteristic [6].

A model of dry clutch torque transmissibility is given by equation (1). It explains the relationship between the torque transmitted by the clutch T_{fc} and the pressure plate clamping load F_{pp}.

$$T_{fc}(F_{to}) = n\mu_d R_m F_{pp}(F_{to}) \tag{1}$$

In (1): n is the number of clutch disc frictional sides; μ_d is the dynamic coefficient of friction, R_m the mean radius, F_{to} the force applied by the actuator to the throwout bearing.

It is well known the importance to evaluate how both the friction coefficient and the cushion spring characteristic affect the performance of the dry clutch assembly during operation, along with their sensitivity to the clutch working temperature [11-15].

For this purpose, this paper aims at investigating the engagement performance of an actuated dry clutch by taking into account the inference of the contact *pressure and the sliding speed on the friction coefficient.*

Firstly, the inference of the temperature, of the contact pressure and of the sliding speed on the friction coefficient have been analyzed in order to obtain useful information on the behavior of the clutch facing material during the working conditions. Secondly, the inference of the temperature on the cushion spring characteristic has been analyzed in order to obtain information on the pressure values which operate on the clutch facing material during the engagement phases. Thirdly, it has been analyzed the relationship between $F_{pp}(F_{to})$ at various temperature, by assigning a bilinear relationship between $x_{pp}(x_{to})$.

TEMPERATURE EFFECT

Temperature plays an important role on the engagement phase in a dry clutch system. In fact, repeated gear shifting induce a rise of the temperature due to the friction between the flywheel and a clutch facing on one side and between the push plate and the second clutch facing on the other side. This thermal effect has a strong influence on the behavior of the main components of the dry clutch assembly. Therefore, if they have not taken into account accurately, they could lead to a poor engagement. For example, after repeated clutch engagement the temperature on the clutch facing could attain very high values, around 300-350 °C and, above 350-400 °C the friction system starts to suffer permanent damage [15].

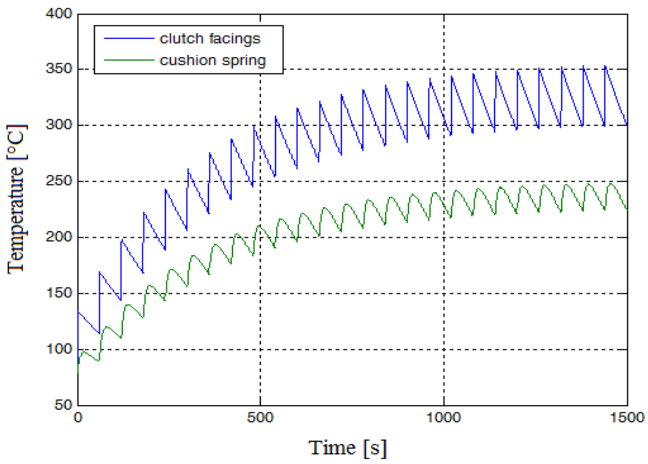

Figure 2: Clutch facings and paddles temperature after repeated starts

A simplified heat transfer model has been assumed in order to calculate the average temperature of the cushion spring. The two clutch facings on flywheel side and pressure plate side have been assumed at the same temperature levels. Say the thermal capacity of the cushion spring, the heat transfer mechanism through the facing materials is mainly given by conductive flux; the heat transfer can be modeled as the conductance times the temperature difference. A more detailed model should take into account the actual heat patterns through the rivets or other metal joints between the facings and the cushion spring. Furthermore, a convective radial heat flux toward an ambient at room temperature is modeled by way of the transfer coefficients. On the base of these hypotheses a simplified thermal dynamics of the cushion spring is provided by a 1st order differential equation [4]. The temperature of the facing material has been simulated through this differential model aiming at reproducing literature results [15, 16], by considering repeated clutch engagements with 60 seconds period (Fig. 2).

FRICTION COEFFICIENT

The friction coefficient has a strong effect on the clutch torque characteristic as shown in the equation (1). For this reason a deep analysis on its variation during the engagement phase is fundamental to improve the performance of an actuated dry clutch. In fact, effective controllers are difficult to be designed without having a dependable frictional map of the clutch-torque transmissibility characteristic [5].

The modeling of friction variation during the clutch engagement process has been studied by numerous authors. The relationship between the static and/or dynamic friction coefficient and the sliding speed has been extensively studied in [17-19]. Raghavan and Jayachandran [20] considered that the coefficient of friction varies with the sliding speed as well as with the generated contact pressure and the number of clutch engagements due to the thermal effect. The thermal effect was also approached through FE analysis for ceramic clutch in [16]. In [14], instead, tests on automotive clutch facings have been carried out in order to analyze the temperature effect. Poser et al. [21] investigated the dependence of friction coefficient of clutch conventional and innovative facings on the sliding speed. In order to ascertain the rise up of the judder phenomenon during the engagement process, Centea et al. [22] and Maucher [23] investigated the gradients of the friction coefficient with slip speed.

FRICTION COEFFICIENT VS. TEMPERATURE

In this section it has been analyzed the thermal impact on the frictional behavior of the clutch disc facings. In fact, during the engagement phase the friction between the clutch facings and the flywheel on one side and the push plate on the other side brings about a rise of the temperature, especially after repeated engagements, see Figure 2. In [14] tests on automotive clutch facings have been carried out in order to analyze the temperature effect. In these tests the friction coefficient exhibits a smooth variation within the temperature range from 25 °C to 250°C, whereas it begins to decline at 250 °C, more sharply after 340 °C [14].

This effect is due to the decomposition of the phenol resin of the clutch facings at high temperature. In fact, when the temperature is higher than 330 °C, a severe thermal decomposition produces fluids and gas emissions. This effect induces not expected phenomena transition from dry friction to lubricated friction. For this reason the friction coefficient drops [14].

FRICTION COEFFICIENT VS. SLIDING SPEED

In this section the influence of the sliding speed, at different contact pressure, on the friction coefficient has been analyzed. In [5] tests on commercial clutch facings have been carried out in order to investigate how these parameters affect the friction coefficient.

The results as function of the sliding speed and for two contact pressure levels are shown in Figure 4. It is evident that for both contact pressure level the friction coefficient tends to an asymptotic value at higher sliding speeds. The friction coefficient asymptotic value is higher for higher contact pressure. Moreover, the contact pressure has a nearly linear influence on the friction coefficient [5]. The relationship between the friction coefficient μ_d, the temperature θ_{cm} and the contact pressure p is given by the equation:

$$\mu_d\left(p,\theta_{cm}\right)=\alpha\left(p-p_0\right)+\mu_0\left(p_0,\theta_{cm}\right) \quad (2)$$

a has been identified and the corresponding value is: $a=0.333\ MPa^{-1}$, $p_0=0.50$ MPa and μ_0 (p_0,θ_{cm}) is derived [14].

Based on the experimental results in Fig. 3, the function that better fits the experimental data, for strictly positive slip speed, is given by the equation:

$$\mu\left(v\right)=\mu_s+\Delta\tanh\left(\gamma v\right) \quad (3)$$

$$\Delta=\mu_d-\mu_s \quad (4)$$

$$\mu_s\left(p,\theta_{cm}\right)=\mu_d\left(p,\theta_{cm}\right)-\Delta \quad (5)$$

where v is the local sliding speed, Δ and γ have been identified and the corresponding value are 0.09 and 2 s m^{-1}, respectively.

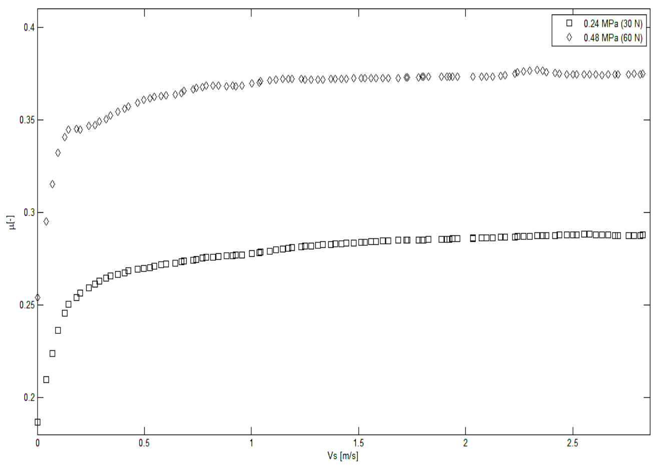

Figure 3: Friction coefficient vs. sliding speed at different contact pressures [5]

INFERENCE ON ELASTIC CHARACTERISTICS

***Cushion Spring**. The cushion spring is a thin steel disc placed between the clutch friction pads and is designed with different radial stiffness in order to ensure the desired smoothness of engagement [2]. When the cushion spring is completely compressed by the pressure plate we say that the clutch is closed, whereas when the pressure plate position is such that the cushion spring is not compressed we say that the clutch is open. We say that the clutch is in the engagement phase when is going form open to locked-up. The temperature influences the local stiffness in the cushion spring load-deflection characteristic [4]: it results in a modification of the function $F_{pp}(F_{to})$ in the equation (1). Thus, the frictional torque transmissibility curve is influenced. In order to evaluate the influence of the temperature on the cushion spring characteristic a finite element analysis has been carried out by using commercial software. The temperature yields also the axial thermal expansion of the cushion spring leading to a variation of the first contact point ("kiss point") between pressure plate and clutch disc.*

***Lever Spring**. The lever spring is the component driven by the electro-hydraulic actuated throwout bearing in order to engage/disengage the clutch (element No. 42 in the Fig. 1). The FEA model allowed exploring the temperature influence on the relationship between the throwout bearing force F_{to} and the pressure plate force F_{pp}. In Figure 4, the results of this analysis are shown.*

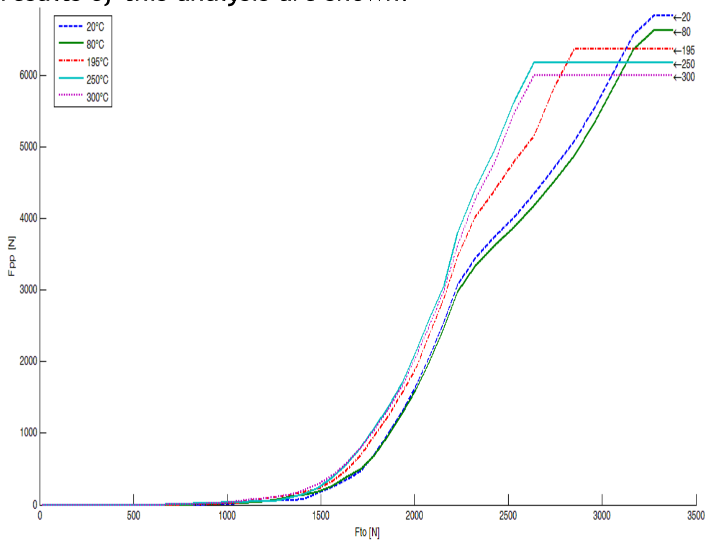

Figure 4: Pressure plate clamping load F_{pp} vs. throwout bearing force F_{to} at different temperatures

CONCLUSIONS

In this paper the multi-variable frictional map and the thermal effect on the cushion spring load-deflection curve have been analyzed in order to find out how they influence the relationship between the pressure plate clamping load and the friction torque uncertainty in a actively closed dry clutch for automated manual transmission (AMT).

After repeated engagements the temperature of the facings rises and this influences the friction coefficient behaviour. In fact, the friction coefficient exhibits a smooth variation within the temperature range from 25°C to 250°C, whereas it begins to decline at 250°C, more sharply after 340°C. This effect is due to the decomposition of the phenol resin of the clutch facings at high temperature. In fact, when the temperature is higher than 330°C and this mechanism leads to friction coefficient drop.

The sliding speed and the contact pressure also influence the friction coefficient. In particular the friction coefficient tends to an asymptotic value at higher sliding speed, and this value is higher for higher contact pressure level. This means that the contact pressure has a nearly linear influence on the friction coefficient.

The temperature affects in two ways the cushion spring load deflection curve. In fact, by increasing the temperature level, the material stiffness changes and this results in point-to-point modification of the load-deflection characteristic slope. Conversely, the thermal load induces a thermal expansion with axial size increasing. These effects result in a modification of the relationship between the pressure plate force and the throwout bearing force.

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