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CUSHION SPRING SENSITIVITY TO THE TEMPERATURE RISE IN AUTOMOTIVE DRY CLUTCH AND EFFECTS ON THE FRICTIONAL TORQUE CHARACTERISTIC

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ABSTRACT

In automotive dry clutch systems the cushion spring plays an important role in terms of gradually torque transmitted during the engagement phase. In fact, it strongly influences the clutch torque transmission from engine to driveline through its non-linear load-deflection curve. Therefore, to improve the gearshift performance in automated manual transmission is fundamental the knowledge of the cushion spring compression behaviour. Furthermore, the cushion spring compression behaviour is influenced by the temperature due to the frictional heat generation of clutch facings with the flywheel and the pressure plate surfaces during the engagement phase. In this paper an analysis on the load-deflection curve, taking into account the thermal load to which it is subjected, of a typical passenger car cushion spring is proposed. Four temperature load values, in addition to the room one, have been analysed to investigate how the cushion spring load-deflection curve depends from temperature.

Keywords: Cushion spring, dry clutch, thermal effects, finite element analysis

1. INTRODUCTION

Recent market forecasts in automotive emphasize the increment of the production of cars equipped with Automated Manual Transmissions (AMTs) especially to the detriment of cars equipped with manual transmissions. This is due to the advantages that AMTs have respect the manual ones in terms of improvement of safety, comfort, reliability, shifting quality, and driving performance, together with reduction of fuel consumption and pollutant emissions [1].

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). The operating modes of AMTs are usually two: semiautomatic or fully automatic. In both cases, after the gear shift command, the TCU manages the shifting steps according to current engine regime, driving conditions and selected program. An AMT is directly derived from a manual one through the integration of actuators; then, development and production costs are generally lower than other automatic transmissions.

In automotive drivelines, the goal of the clutch is to smoothly connect two rotating masses, the flywheel and the transmission shaft, that rotate at different speeds, in order to allow the transfer of the torque generated by the engine to the wheels through the driveline [2]. A limit of the AMTs is the lack of traction during gear shift actuation as for manual ones. For solve this problem modern designs involve a dual-clutch (DCT) system between the engine and the transmission. DCT brings as outcome comfortable, jerk-free gear changes with the same relaxed driving style found in an AT combined with the efficiency of a manual transmission. It is as smooth as the most sophisticated automatic, but more economical than a conventional automatic; it is as easy to drive as a standard auto, faster and more responsive than manual gearbox on high performance cars [3].

For AMT and DCT systems very important are the strategies of the control unit, because they influence the vehicle propulsion perceived by the driver [4-7]. The model proposed in [8] describes the influence of the load-deflection curve of the cushion spring to the torque transmissibility by the clutch. In fact the torque transmissibility, as the load-deflection curve of the cushion spring, is non-linear and this issue affects the effectiveness of the control strategy implemented into TCU. Furthermore, as shown in [9, 10], after repeated clutch engagement the temperature on the clutch material facings, and consequently on the cushion spring, could reach very high values.

For these reasons, in this paper the influence of the temperature on the cushion spring load-deflection characteristic has been investigated for a given dry clutch cushion spring.

The results of this analysis could be useful for designers of automated clutches and control engineers to recast the cushion spring conception for overcoming some poor engagement performances exhibited by AMTs based on traditional manual schemes with “retrofit” automation through actuators.

2. DRY CLUTCH SYSTEM

In this section the dry clutch engagement system has been analysed. It consists of a steel clutch disk connected to a hub by means of torsional dampers springs which damp out torsional vibrations. The cushion springs and two or more friction pads (also called clutch facings) are clamped on the clutch disk by rivets. Diaphragm spring (or washer spring or Belleville spring), instead, is riveted to the cover and to the pressure plate (or push plate). This latest three parts are torsionally clamped to flywheel while clutch disk are torsionally clamped to main shaft. The diaphragm spring transforms a throwout bearing position into a corresponding pressure plate position. The pressure plate presses the clutch disk against the flywheel or keeps it apart. The friction between the external pads on the two sides of the clutch disk and the flywheel and pressure plate respectively, generates the torque transmitted from the engine to the driveline through the clutch [8]. When the pressure plate and the clutch disk have the same speed of the flywheel because they are constrained by the diaphragm spring the clutch is locked-up. In such operating conditions the engine is directly connected to the driveline. So the cushion spring (also called flat spring) is compressed and when it is completely compressed the clutch is closed. Conversely when the pressure plate position is such that the cushion spring is not compressed the clutch is open. When the clutch is going from open to locked-up it is in engagement phase [11]. In Fig. 1 a scheme of a push-type dry clutch system is shown.

The cushion spring is a thin and waved steel disk placed between the clutch friction pads and it is designed with different radial stiffness in order to ensure the desired smoothness of engagement.

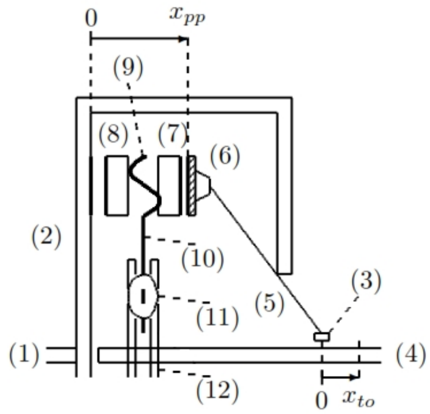


Fig. 1. Scheme of a dry clutch system in the open position: (1) crankshaft, (2) flywheel, (3) throwout bearing, (4) main shaft, (5) diaphragm spring, (6) pressure plate, (7) friction pad on the pressure plate side, (8) friction pad on the flywheel side, (9) cushion spring, (10) clutch disk, (11) torsional damper spring, (12) hub

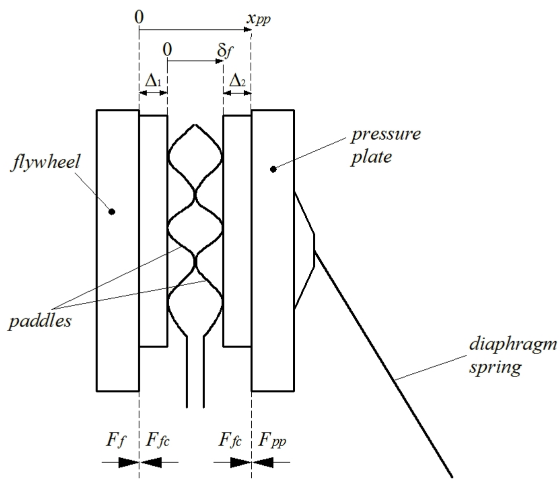


Fig. 2. Clutch engagement forces equilibrium

The knowledge of the clutch engagement scheme allows to correlate the throwout bearing displacement, the cushion spring compression and the clutch torque transmissibility. In Fig. 2, a zoom of the clutch engagement scheme is depicted.

In Fig. 2: Δ_1 and Δ_2 are the thicknesses of the two friction pads and Δ is their sum; the cushion spring compression is measured by $\text{¢}_f \in [0, \Delta_f]$; $F_{pp}(x_{to})$ is the pressure plate clamp action; $F_{fc}(\text{¢}_f)$ is the cushion spring axial reaction; F_f is the flywheel reaction. The cushion spring compression ¢_f is equal to zero when the clutch is closed and equal to

Δ_f value when the clutch is open. The throwout bearing position x_{to} is the clutch variable directly controlled by TCU through the electro-hydraulic actuator and its position results in given pressure plate position and cushion spring deflection. When x_{to} has zero value the clutch is open, when it has x_{to}^{cnt} value the friction pad and flywheel come in contact (also called "kiss point") and when its value is $x_{to} \in \left[x_{to}^{cls}, x_{to}^{\max} \right]$ the clutch is closed. The kiss point is the stroke point of the throwout bearing where the clutch disk comes in contact with the push plate and on the other side with the flywheel.

In [8] a novel model of dry clutch torque transmissibility has been proposed; it explains the role of clutch springs and how they influence the torque transmissibility. The connection between torque transmissibility and cushion spring force obtained in [8] is given by equation (1).

$$T_{fc}\left(x_{to}\right) = n\mu R_{eq} F_{fc}\left(\Delta_f\left(x_{pp}\left(x_{to}\right)\right)\right) \tag{1}$$

where n is the number of pairs of contact, μ is the dynamic friction coefficient and R_{eq} is the equivalent radius of the contact surface. The analysis of the equivalent radius and the experimental procedure for measuring the frictional behaviour of the clutch facings are presented in [1, 8].

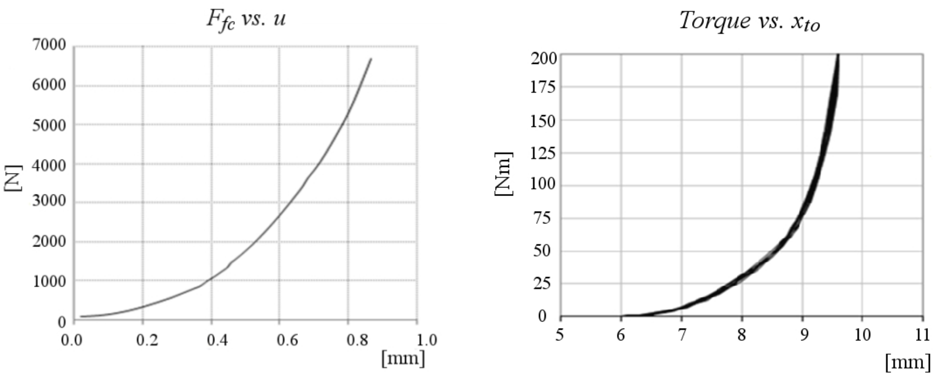


Fig. 3. Comparison between $F_{fc}(u)$ and $Torque(x_{to})$

In the Fig. 3a the cushion spring load-deflection curve is depicted while, in Fig. 3b the torque characteristic is shown. It is evident that the shape of the cushion spring curve has a strong inference on the torque characteristic. In fact, for a given clutch architecture with n frictional surfaces, μ and R_{eq} vary in narrow ranges or under opportune hypotheses they could be considered as constants [1, 8]. Thus, bearing in mind the equation (1), the clutch torque characteristic is substantially provided by the cushion spring load-deflection curve multiplied by a constant value.

The purpose of this paper is to investigate how the temperature influences the cushion spring behaviour in terms of the its load-deflection curve and in terms of the axial thermal expansion. These two phenomena have direct implication on the dry clutch transmissibility curve. In fact, it could be useful to implement in the TCU a set of load-deflection curves which are parameterized by the temperature. In this way, on the base of measured or estimated clutch/pressure plate interface temperature, the TCU could dynamically take into account the thermal distortion of the torque characteristic and the displacement of the kiss point in order to control the clutch actuator. The neglecting of the temperature inference might lead to a lack of torque during the engagement phase since the TCU calculates a not accurate throwout bearing position or result in uncomfortable gear shift.

Numerical Analysis Validation

Firstly it is necessary to implement a FE model which simulates the cushion spring compression and to compare the results with experimental tests to validate this kind of simulations. The implementation of a FE model to simulate the cushion spring compression requires not trivial effort due to the complex wavy shape and the continuously variable contact surface.

3. EXPERIMENTAL TESTS

In order to obtain the characteristic $F_{fc}(u)$, experiments have been realized on a eighth of clutch disk. The Instron 4200 series IX automated material testing system, suitable for compression and traction testing, has been used. Appropriate load cell detects the cushion spring reaction force on the sample in consequence of a certain compression. The upper pressure plate speed used was 0.5 mm min^{-1} simulating a quasi-static test. The tests were performed on a new clutch disk; in Fig. 4, clutch disk and test specimens are shown.

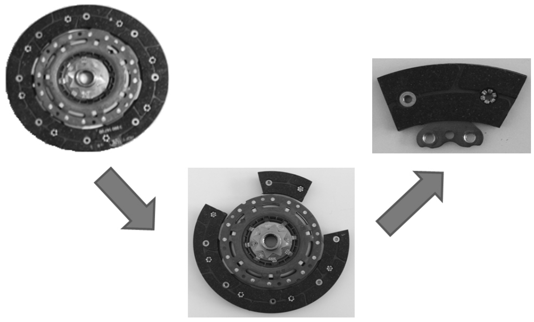


Fig. 4. From new clutch disk to compression test specimen

The zero displacement was assumed when the cushion spring reaction force is around 10 N to cancel out the resolution uncertainty due to the testing system. In Fig. 5, the results of the compression test are shown.

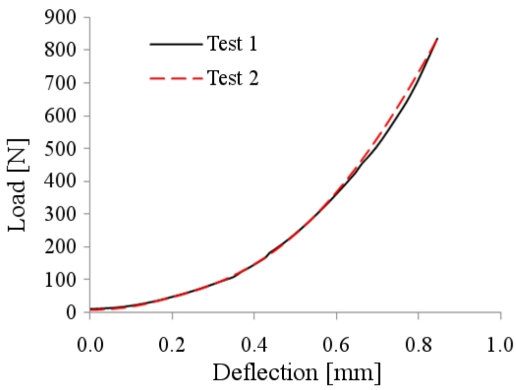


Fig. 5. Load-deflection curve from experiments

Each cushion spring subsystem is a series of two paddles, as already shown in Fig. 2. The cushion spring considered in this paper is composed by eight pairs of paddles which work how parallel springs. The whole characteristic has been obtained by considering that all the paddles have the same load-deflection characteristic. In fact, the tests have been repeated on more cushion spring paddles and their repeatability has been checked. Therefore it has been confirmed that the entire characteristic can be obtained simply multiplying by 8 the characteristic of each pair of paddles.

4. FINITE ELEMENT MODEL

An ANSYS® FE model has been realized to simulate cushion spring compression and in Fig. 6 is shown the scheme. Only one subsystem has been modelled according to the symmetry assumption described above.

Reverse engineering method has been used to import tested cushion spring geometry into CAD file. The acquiring was performed by means of a digital scanner at Department of Industrial Engineering of University of Salerno. In Fig. 7 the acquired cushion spring segment (a) and the obtained result (b) are shown.

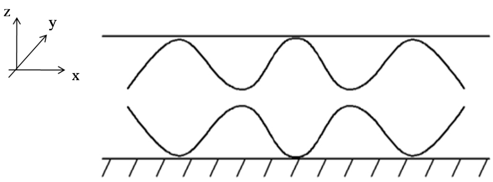
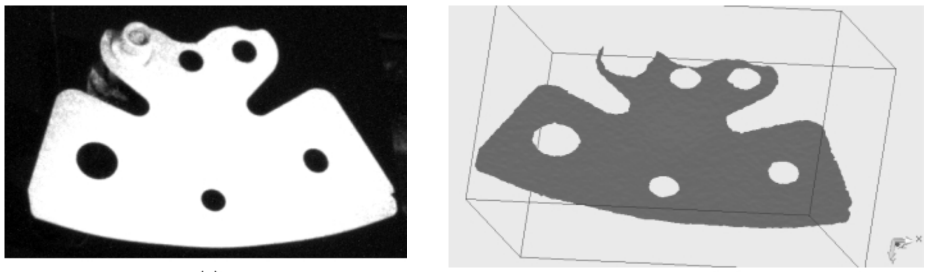


Fig. 6. Scheme of the system to model



(a) (b)

Fig. 7. The acquired cushion spring paddle (a) and the mathematical 3D representation (b)

The achieved result is a cloud of points and, after appropriate adjustments, it has been used for Finite Element (FE) analysis. The cushion spring geometry has been meshed with shell elements (SHELL 181 in the ANSYS library) according to its thin thickness (0.4 mm) and computed using a linear elastic material law. The thickness of the cushion spring has been measured both with digital scanner after the acquiring and with digital calibre, in different points, and with both methods the same thickness value has been obtained. The clutch facings and pressure plates have been modelled by flat rigid plates. The axial compression of the clutch disk during the gear re-engagement has been simulated in this way.

Surface-to-surface contact has been used to simulate cushion disk phenomena compression. Each contact pairs needed of a master (contact element “CONTA 174”) and slave (target element “TARGET 170”) surface. The augmented Lagrangian algorithm has been chosen to solve the contact problem. In this work the only real constants (RC) used are: normal contact stiffness factor named FKN and penetration tolerance factor named FTOL. FKN represents the stiffness of a fictive springs disposed between the two surfaces in contact [12]. Its default value is 1 but it could range between 0.01 and 100 [13]. A high value of FKN gives good solution accuracy but can lead to ill-conditioning of the global stiffness matrix causing convergence difficulty [12, 13, 14]. On the other hand a too low contact stiffness value can lead to a bad solution [12]. FTOL default value is 0.1 and if it is too small, i.e. small penetration, the FE analysis can fail. On the other hand if FTOL is too large, i.e. high penetration, can lead to a bad solution [12, 13]. Lower pressure plate is fixed in all directions while the upper one moves axially compressing the paddles. The plate motion is controlled by displacement in order to limit convergence problem. The cushion spring paddles are clamped on the clutch disk by rivets; in the FE model this condition has been simulated by locking the nodes of the holes where they are housed. In Fig. 8, the ANSYS model used is shown.

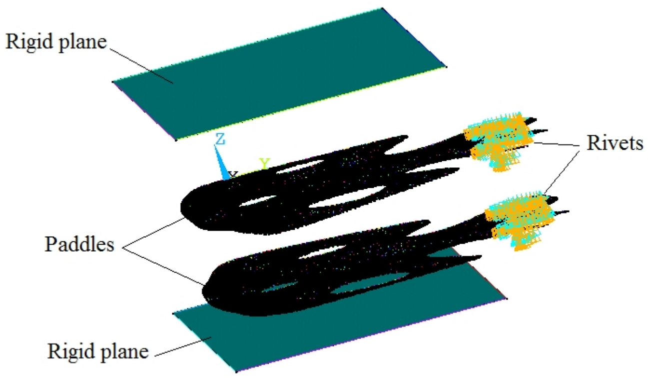


Fig. 8. ANSYS model for the simulation of the paddles compression

5. MODEL VALIDATION

Two different mesh densities have been used to validate the model: 0.5 mm and 2.0 mm. Furthermore, the ANSYS default convergence criterion has also been used. The best results have been achieved with this factor values: FKN=0.9; FTOL=0.1. The zero position was assumed when the cushion spring reaction force is equal to 10 N to compare FE results with experimental ones. In Fig. 9, the experimental load-deflection curve is compared with the result of the simulation.

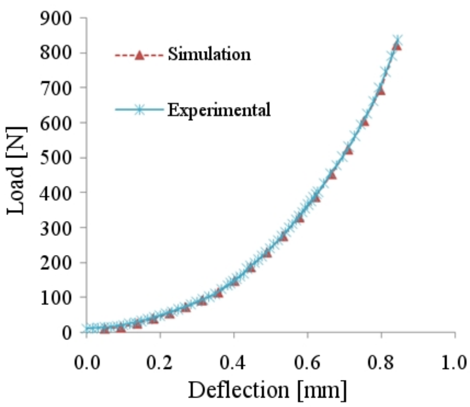


Fig. 9. Load-deflection curve: experimental and simulation results

6. CUSHION SPRING CURVE: THERMAL EFFECTS

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 behaviour 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 [9].

A simplified heat transfer model has been assumed in order to calculate the average temperature of the cushion spring $\bar{~}_{cs}$. The two clutch facings on flywheel side and pressure plate side have been assumed at the same temperature levels. Say C 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 modelled as the conductance U times the temperature difference $(\theta_{cm} - \theta_{cs})$. 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 θ_a is modelled by way of the transfer coefficients H.

On the base of these hypotheses, the simplified thermal dynamics of the cushion spring is provided by a 1st order differential equation where U has value of 0.1 W K^{-1}, H of 0.04 W K^{-1} and C of 1.0 J K^{-1}:

$$U\left(\theta_{cm}(t)-\theta_{cs}(t)\right)+H\left(\theta_{a}-\theta_{cs}(t)\right)=C\dot{\theta}_{cs} \tag{2}$$

$$C\dot{\theta}_{cs}(t)+(U+H)\theta_{cs}(t)=U\theta_{cm}(t)+H\theta_{a} \tag{3}$$

$$\frac{C}{U+H}\dot{\theta}_{cs}(t)+\theta_{cs}(t)=\frac{U}{U+H}\theta_{cm}(t)+\frac{H}{U+H}\theta_{a} \tag{4}$$

The temperature of the facing material has been simulated aiming at reproducing literature results [9, 10], by considering repeated clutch engagements with 60 seconds period. The previously described model has been implemented in Matlab/Simulink®; the results are depicted in the Fig. 10.

In addition to these phenomena the temperature also changes the load-deflection curve of the cushion spring and consequently also the clutch transmission characteristic is affected by the temperature.

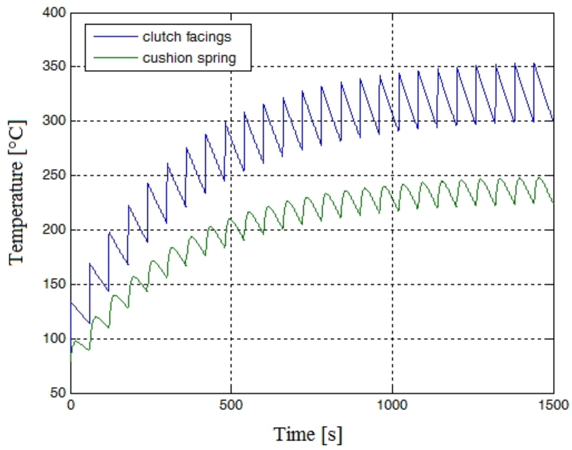


Fig. 10. Clutch facings and paddles temperature after repeated starts

Problem description

In order to predict the influence of the temperature on the cushion spring load-deflection characteristic a finite element analysis has been performed, using above mentioned ANSYS® FE model. The analysis aimed to investigate the temperature influence on the cushion spring characteristic modification and the consequent torque transmissibility curve. The knowledge of these effects could provide useful data for a real-time adaption of TCU control through the measurement of the estimation of clutch system average temperature.

As first simulation step, an uniform temperature field has been applied to the cushion spring paddle to calculate the only thermal distortion and the modified material properties. As second step, a controlled displacement of rigid planes to induce axial compression has been simulated in order to investigate how the temperature affects the load-deflection curve.

Both the steps involve nonlinear shape modifications since the analysis faces with contact problems. The first analysis, in fact, is nonlinear because during the thermal strain the paddles get in contact both each other and with the rigid plane that simulate the clutch facings.

In this analysis an isotropic linear elastic material, characterised by a Young's module depending on the temperature and a constant expansion coefficient, has been used to model the material behaviour. The Table 1 shows the material proprieties used for the analysis.

Table 1. Materials properties

Parameters	*Values*			
Temperature [°C]	20	80	200	350
Young's module [GPa]	206	202	196	186
Expansion coefficient [°C^{-1}]	1.12 10-6			

Finite element results

In this section the results of the above described FE analysis are shown. The analysis was based on the cushion spring paddle geometry acquired through the reverse engineering system. The FE model is described in the above sections.

The results of this analysis are shown in Fig. 11: it has been assumed a reference load-deflection curve at 20°C which starts at 0 mm. The zoom of the Fig. 11 shows how at higher temperatures the curves start before of the reference curve at 20°C. This effect is due to thermal expansion which produces axial size increase and consequently a change of the kiss point position.

The waves of a single cushion spring paddle has a maximum height of about 0.7 mm; therefore to flatten completely a pairs of paddles, see Fig. 6 for details, the maximum compression value should be around 1.4 mm.

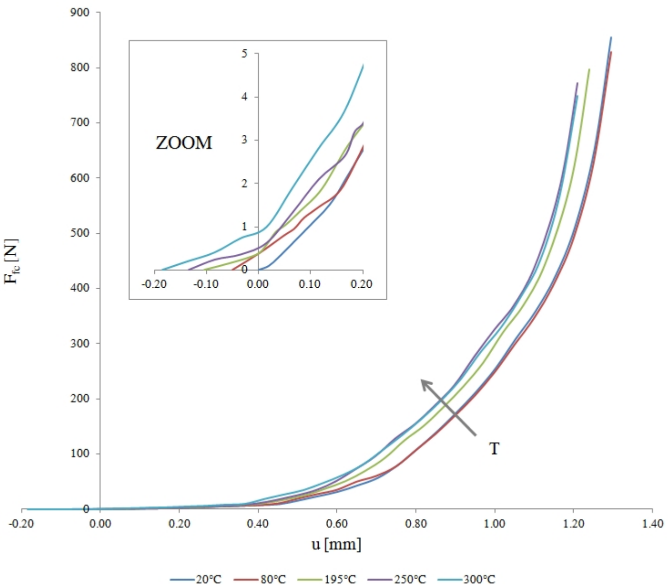


Fig. 11. Paddles load-deflection curves depending on the temperature

Because the clutch system, during its life, could work in a wide range of temperatures how shown previously, in this analysis the cushion spring behaviour has been analysed at four temperature levels (80°C; 195°C; 250°C; 300°C), in addition to the room one.

The second effect is that, increasing the temperature level, the material stiffness changes and this results in modification of the load-deflection slope [15].

In Fig. 12 the clutch torque transmissibility depending on the temperature on the cushion spring is shown. The clutch torque transmissibility curves have been obtained by using the equation (1) and by considering $n = 2$, $\mu = 0.28$ and $R_{eq} = 0.089$ m. Also in this case the reference curve is the curve at 20°C while the reference value of the kiss point is 4 mm. It is evident that the diaphragm spring lever ratio amplifies the change of the kiss point as perceived by the actuator on the throwout bearing. For example, at a temperature value of 195°C there is an axial thermal expansion of around 0.1 mm which result in around 0.3 mm of kiss-point modification.

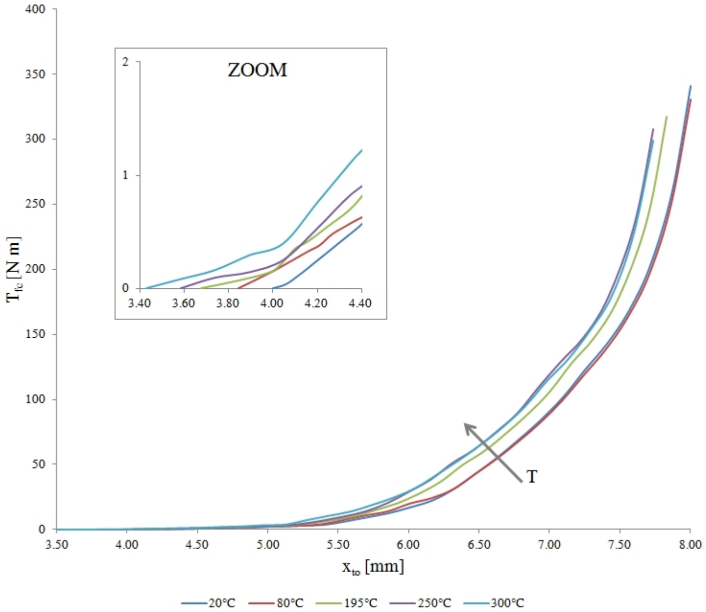


Fig. 12. Clutch torque characteristic at different temperature of the cushion spring

After these results the equation (1) can be rewritten by considering the thermal effects on the cushion spring:

$$T_{fc}\left(x_{to}, \theta_{cs}\right) = n\mu R_{eq} F_{fc}\left(\delta_f\left(x_{pp}\left(x_{to}, \theta_{cs}\right)\right), \theta_{cs}\right) \tag{5}$$

where the relationship $x_{pp}\left(x_{to}, \theta_{cs}\right)$ represents the change of the kiss point and the relationship $F_{fc}\left(\delta_f, \theta_{cs}\right)$ represents the change of the material stiffness and, consequently, the modification of the load-deflection characteristic slope.

7. CONCLUSIONS

This study aimed to evaluate the influence of the temperature on the static load-deflection curve of a cushion spring in an automotive dry clutch system by means of a finite element analysis. This analysis has highlighted that the temperature influences mainly in two ways the cushion spring load-deflection characteristic. In fact, an increment of the temperature level result in a decrease of the material stiffness and this is underlined by a curve slope modification. On the other hand, also the change of the kiss point location is due to the thermal effects which, in fact, induce an axial thermal expansion.

It is known that the main contribution to the shape of the clutch torque characteristic is given by the cushion spring load-deflection characteristic. For this reason, the influence of the temperature on this fundamental component is an important issue to improve the control algorithms implemented in the transmission control unit (TCU) of the automated manual transmissions. In fact, by not taking into account of the thermal effects on the cushion spring and, consequently, its influence on the kiss point location during the engagement phase could lead to a wrong position of the throwout bearing assigned by the TCU.

Concluding, the outcome of this analysis could be useful for designers and control engineers of automated clutches to recast the cushion spring conception in order to obtain an improvement of the performance exhibited by AMTs after repetitive gear shifting.

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