WEAR INVESTIGATION OF WET CLUTCH FRICTION MATERIAL

B. Genbrugge[1], P. De Baets[2], W. Ost[2]

[1] Ghent University, Belgium
[2] Ghent University, Laboratory Soete, Belgium

Abstract Wear of friction plates for wet clutch applications can result in a decreasing transmittable torque and the occurrence of vibrations throughout the entire drive chain causing loss of performance and discomfort. The need for a simple wear model exists to predict the lifetime of the clutch and to give an insight in the combined influence of the operational parameters such as pressure and speed. In this paper wear of paper-based friction material is investigated on a simplified SAE#2 test-rig using only one friction plate and one spacer plate. During engagement torque, applied pressure and sliding velocity are continuously monitored. After a set number of engagement cycles the thickness change of the friction plate is measured and surface topography of the spacer plate is registered. Based on Archard's wear law a specific wear rate is derived.

Keywords wet clutch, wear, specific wear rate

1 INTRODUCTION

Wet friction clutches consist of a pack of alternating friction- and separator plates anchored to the in- and outgoing axle by in- and external teeth. A normal force applied to the clutch pack by a piston, enables the clutch to transmit a friction torque between both axles. Wet clutches are commonly used as shift clutches in automatic transmissions for passenger cars and heavy duty off-road vehicles. As the number of engagement cycles increases during the lifetime of the clutch, the friction material wears out which results into a change in thickness of the plate and alteration of the friction surface. This over time will lead to a drop of the maximal transmittable torque and the end of life of the clutch.

Paper-based friction plates for wet clutches are produced by inbedding cellulose fibres into a resin. Wear of such materials is known to be depending on a large number of parameters. These parameters include the operational conditions of the wet clutch as well as the material properties of the friction plate itself. Experimental results show that elastic friction material is able to make a more uniform contact with the opposing separator. This uniform contact leads to a better heat distribution and reduced wear of the surface [1]. By decreasing the E-modulus, the occurrence of so called hot-spots, places of increased surface temperature on the separator, is reduced [2]. In the same way as the E-modulus, non-flatness of the separator results in increased wear-rates due to temperature differences [1]. These uneven temperature distributions may lead to failure of the clutch by coning [3].

Although performance of the clutch is improved by appropriate material selection during design and careful manufacturing of the friction plates, wear of the clutch during its lifetime greatly depends on the conditions of use, such as relative velocity at the start of engagement, and operational parameters. The latter include the inertia of the drive line and the preset applied normal force. No strict conclusions can be drawn concerning the influence of pressure. On one hand increasing pressure leads to a shorter stop time thereby reducing sliding distance and probably wear. On the other hand shorter stop times lead to an increased power input enhancing elevated surface temperatures and possible additional wear. What can be concluded is that combining high power load with high energy load always results in extreme wear [3]. It is remarkable to notice that wear of paper-based friction material seems to be more sensitive to changes in energy input due to inertia, rather than change in energy load due to velocity [3].

Noticing that sliding distance is the product of instant velocity and time, both pressure and velocity are combined in Archard's wear law. This wear law predicts steady state thickness change using a specific wear rate coefficient. Here the wear behaviour of a paper-based friction plate was studied based on Archard's law and under fixed conditions. Knowledge on the combined influence of operational parameters and the presence of a predictive wear model can lead to an optimized maintenance scheme and could make contributions to the development of new friction materials and to wet clutch design.

2 TEST RIG

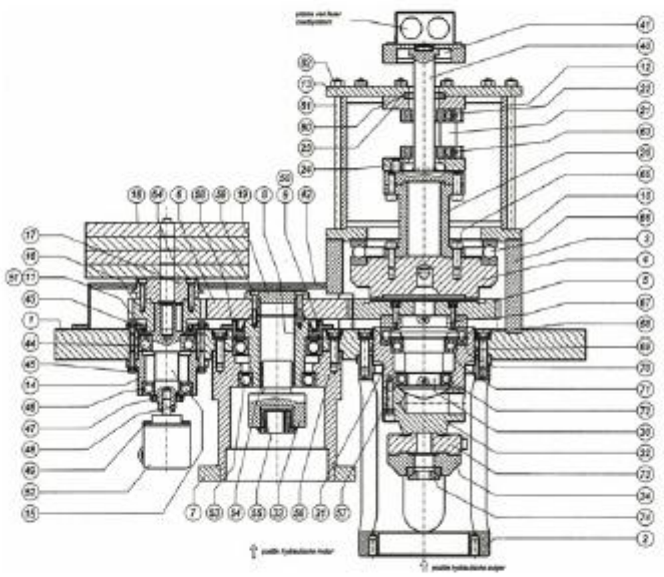

Fig 1: View of the test rig.

A test rig, similar to the apparatus described by Holgerson [4], is used with only one friction plate and one spacer plate. An overview is given in figure 1. Gears connect the inertia plates (18) to the test plate allowing for a 2:1 reduction. A hydraulic motor is connected to the middle gearwheel (6) through a one way clutch. A steel plate separator with inner diameter of 98 mm and outer diameter of 133 mm is mounted on the third gearwheel (5) and rotates at motor speed.
Two Rexroth-Bosch HACD-1 Digital Control Cards are programmed to accelerate the motor up to 700 rpm and to control linear movement of the hydraulic piston. Once the preset rotational speed is reached the motor stops. During one second the 0.1475 kgm^2 rotating inertia is allowed to spin freely as a result of the unidirectional coupling between the motor and middle gearwheel. After this small pause the hydraulic piston is send out to press the separator against the static friction plate. During engagement F_a is held constant at 11.5 kN until the kinetic energy of the rotating inertia is dissipated in the clutch and the inertia is stopped. After a pause of 5 sec this engagement cycle is repeated.
Automatic transmission fluid, Texametic 7045, is used at a rate of 0.85 l/min. No external cooling of the oil sump was used and inlet oil temperature never reached more than 35 °C.

2.1 Measurements

Two pairs of friction and separator material were submitted to approximately 4000 resp.3250 engagement cycles. Measurements were performed during and after the tests.

2.1.1 online measurements

During engagement normal force, torque, axial displacement of the piston, rotation speed of the inertia and inlet oil temperature were continuously monitored during the 1st minute of every 15 minutes. Signals were sampled at a rate of 600 Hz resp. 1000 Hz and processed with LabView 8.0.

2.1.2 offline measurements

To investigate wear of the friction plate, thickness measurements were performed at four points 90 degrees apart before, during and after completion of the test program. A digital micrometer was used with an accuracy of 0.01 mm.
For the second test specimen surface topography of the separator was inspected. Surface parameters R_t, R_{max}, R_z, R_a, R_{sk} and R_{ku} were acquired at six different positions (see figure 4) using a Hommel Tester T1000 Surface Scanner.

3 RESULTS

3.1.1 wear of paper-based friction material

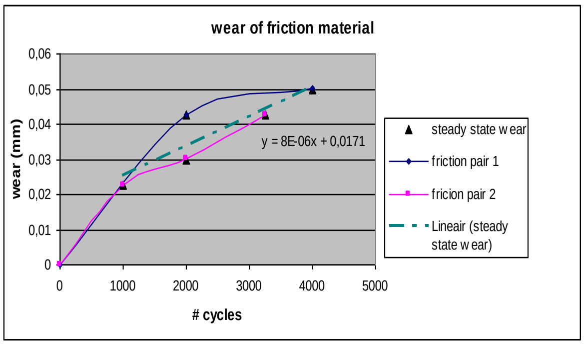

Fig 2: Wear of friction material with trendline.

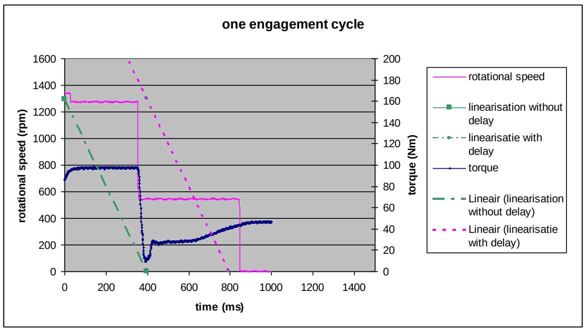

Fig 3: Rotational speed of inertia and torque curve during one engagement.

An average was calculated of the four measured thickness changes on the friction plate. Although wear of the paper-based friction material is very small, a drop in wear-rate can be seen approximatly halfway the testprogram. This suggest a running-in period of 2000 engagement cycles. Based on the wear results of the two test specimens and using Archard´s wear law (1), an estimation of the specific wear rate can be made for the subsequent steady state wear cycles.

$$\Delta h = K \cdot p \cdot s \tag{1}$$

Sliding distance can be calculated by integrating instant sliding velocity over engagement time. As seen in figure 3, integration of the velocity data would be inaccurate as steps in the velocity curve occur. These steps are probably due to delay in signal processing before these signals reached the data acquisition. Because of the short stop time of 0,397 s it is a reasonable to suggest that velocity decreases linear with time. Sliding distance per cycle can be estimated as 780 mm. For a total of 4000 resp. 3250 cycles that makes approximately 3121 m resp 2535 m.
Calculation of the specific wear rate based on a steady state wear rate of 8 10^{-3} mm/1000 cycles (see figure 2) results in 5,661 10^{-15} m^2/N.

3.1.2 surface inspection of the separator

pair 2	# cycles	*Rt*	*Rmax*	*Rz*	*Ra*	*Rsk*	*Rku*
position 1	*0*	1.72	1.62	1.21	0.13	-1.386	7.774
	1000	2.38	2.38	1.34	0.13	-2.658	21.733
	2000	3.3	3.3	1.64	0.15	-2.153	19.393
	3250	2.66	2.66	1.3	0.13	-2.898	23.435

pair 2	# cycles	*Rt*	*Rmax*	*Rz*	*Ra*	*Rsk*	*Rku*
position 4	*0*	2.32	2.3	1.51	0.19	-1.953	10.371
	1000	1.95	1.92	1.38	0.15	-1.999	10.237
	2000	1.61	1.54	1.23	0.15	-1.623	7.218
	3250	1.3	1.26	1.08	0.14	-1.504	6.453

Table 1: Roughness measurement on position 1 and 4 of the second friction pair.

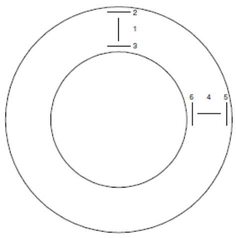

Fig 4: Positions of roughness measurement.

Based on experiments conducted on the same test rig under similar test conditions [5], it was expected that roughness parameter R_a would increase when measured perpendicular to the direction of sliding i.e. positions one and four. This could be explained by the presence of wear debris between friction plate and separator making wear grooves in the sliding direction. Although grooves appeared after testing, no such trends could be noticed in the results presented here. This is probably due to the short period of testing.

4 CONCLUSIONS

1. Under the specified test conditions the running-in period for the tested paper-based friction plate and steel plate separator takes approximately 2000 engagement cycles.
2. Experimental results have shown a steady state specific wear rate for the paper-based friction material of 5,661 10^{-15} m^2/N. Better estimations might be achieved by improving accuracy of the velocity measurements and extending test duration.

5 NOMENCLATURE

Δh	thickness change	m
F_a	axial force	N
K	specific wear rate	m^2/N
p	contact pressure	Pa
s	sliding distance	m

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