EXPERIMENTAL STUDY OF FRICTION AND WEAR OF PAPER BASED WET FRICTION CLUTCHES BASED ON SAE#II AND PIN-ON-DISK TESTS

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SUMMARY
The friction behaviour of wet clutches for automatic transmission applications strongly influences the dynamic behaviour of the entire machine or vehicle including the transmission. The wear, but also to a certain extent the friction curve, determines the lifetime of the clutch. The role of wear is obvious. The friction coefficient of the material couple friction plate/separator plate decreases with number of engagement cycles. As a result the possible torque of the transmission decreases with time. Under a certain threshold the clutch has to be revised. But because manufacturers tend to oversize their clutches the decrease in friction coefficient does not yield a limitation to the lifetime of the clutch.

Keywords: Wet clutch, friction, wear, simulation test

1 INTRODUCTION
Wet clutches often are used for transmission application in machine and vehicle drivetrains. A typical application is the automatic powershift transmission used in earth moving off-road vehicles [1]. In this case the clutch has to transfer energy continuously while synchronising. In its slipping state the clutch absorbs both inertial energy and prime mover energy. The automatic transmission connects the output of the motor and torque converter to the drive train of the vehicle. Several clutches are necessary to make different gear and direction changes.

The wet friction clutch investigated in this paper essentially consists of 9 friction plates, radially fixed by means of a spline on a central shaft, and 8 separator plates radially fixed in a drum. In order to transmit a certain torque, the friction and separator plates are pressed against each other by means of an axial hydraulic piston. For smooth coupling characteristic the hydraulic pressure is applied linearly with time (electronically controlled modulation). When the modulation is softer (smaller inclination of the pressure vs. time curve), the slipping time of the clutch and the heat dissipated in the clutch becomes larger, but as an advantage the jerk (the derivative of the acceleration) and torque peak remain acceptable. The friction surfaces are cooled by means of oil brought into the clutch through the hollow, central shaft and ejected between the friction surfaces.

2 EXPERIMENTAL

2.1 SAE # II Test rig

2.1.1 Test rig characteristics
The test rig schematically shown in Figure 1 (SAE#II machine) is used to investigate the friction characteristics and wear of the clutch plates.

A rotating inertia of 9.44 kgm² (1) is driven by an electric AC motor (2). At a certain moment (pre-set rotational speed) the motor is powered off and the clutch (3) is closed (hydraulic pressure is linearly increased), connecting the inertia to the fixed frame (4). The torque between the frame and the inertia is measured by means of a torque transducer (5). The clutch is lubricated and cooled by means of oil pumped with a hydraulic pump (6) into the clutch through a filter (7) and a cooling heat exchanger (8). The rotational speed of the inertia is measured with a magneto-resistive speed sensor on a 60 teeth gear (9); the pressure on the axial piston is measured (10) as well as the oil temperature and the temperature of the several separators in the clutch.

Figure 1: SAE # II test rig

All measuring signals are amplified, digitised and stored on a PC. The same PC controls the motor speed and the opening and closing of the clutch. The SAE#II tests consist of several cycles, in which the clutch is used to decelerate the rotating inertia. At the start of each cycle the clutch is opened and the electric motor accelerates the inertia up to a pre-set speed of 1300 rpm. Then the clutch is closed, the contact pressure between friction and separator plates increases up to 2.8 MPa, and the inertia decelerates to a standstill. The clutch is kept closed for 4 seconds after standstill is then opened and a new cycle is started.
Due to the small inclination of pressure ramp the clutch is already closed (lock-up) before the control pressure reaches its maximum.

Before and after the whole test, the thickness of the friction plates and the roughness of the separator plates are measured.

### 2.1.2 Clutch characteristics

#### Friction plate characteristics

A friction plate is made of a substrate steel plate (Euronorm 1C60) with hardened external spline teeth, coated with a friction material. Organic fibre matrix friction material (‘paper’, Raybestos S-7901-2) with groove pattern (waffle) is used.

#### Separator plates

The separator plates are made of stamped steel sheet (Euronorm 1C60, $\varnothing_{\text{out}} = 133.25 \text{ mm}, \varnothing_{\text{in}} = 92.66 \text{ mm}$). The surface is ground and tumbled to a mean surface roughness of $R_a 0.32 \mu\text{m}$. The hardness of the plates equals 215 HB.

#### Lubricant

For lubrication and cooling of the clutch a commercially available ATF-oil is used (Texamatic 7045, viscosity 81.1 cSt at 20 °C).

### 2.2 Pin-on-disk test rig

Because the full clutch tests are very time consuming (lot of preparation work, long testing time) and expensive it was decided to scale down the clutch to the contact of a coupon from a friction plate on a separator plate. The surface area of the friction coupon is chosen in such a way that although the normal force is limited, the apparent contact pressure on the coupon lies in the same range as in the clutch test (0 - 3 MPa). The contact pressure and the rotational speed can be varied separately.

#### 2.2.1 Pin-on-disk test rig characteristics

A separator plate (1) is placed on a turning Table (2) driven by a frequency controlled AC motor (3). A coupon from a friction plate (4) is pressed against the separator plate by means of a pneumatic piston (5). A ball-and-socket joint is used to avoid misalignment. The friction surface is lubricated with abundant oil flow (6) at 80 °C. The normal force and the friction force on the coupon are measured using resistive loadcells (11 res. 7). The rotational speed of the separator plate is measured with the tachometer (8), the temperature just above the contact surface with a thermocouple (9). The combined wear of the friction plate and the mating surface is measured with the contactless distance probe (10). All signals are amplified and stored on a PC hard disk. The PC also controls the rotational speed of the separator plate.

The pin-on-disk tests comprise several cycles in which the rotational speed of the separator plate is varied, but the normal force is kept constant. In each cycle the rotational speed is increased up to 1300 rpm in 4 seconds, remains constant for 5 seconds and is decreased to standstill in 4 seconds. The separator plate is halted for 2 seconds and then a new cycle is started. Before and after each test, the surface roughness of the separator plate and the thickness of the friction plate are measured.

#### 2.2.2 Specimen characteristics

A coupon with a surface of 310 mm$^2$, of the same friction plates as for the clutch tests is used. The external spline teeth are removed, and part of the friction material is removed by milling. The separator plates and the lubricant are the same as used in the clutch tests.

### 3 RESULTS AND DISCUSSION

#### 3.1 SAE # II tests

Three different tests were conducted with number of cycles 18590, 11140 and 30455 respectively. Test 1 was stopped at the moment that a change of the friction coefficient (occurrence of vibrations) was detected. Test 2 stopped accidentally and experiment 3 was stopped because of heavy vibrations.

The measured torque is the result of the friction force between the plates, generated by the axial force on the clutch plates. On each time the instantaneous friction coefficient can be calculated [2] as

$$\mu(t) = \frac{3M(t) \cdot (r_o^3 - r_i^3)}{2zF_a (r_o^2 - r_i^2)}$$

With:

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu(t)$</td>
<td>friction coefficient</td>
</tr>
<tr>
<td>$M(t)$</td>
<td>measured torque</td>
</tr>
<tr>
<td>$r_o$</td>
<td>outer radius of the friction plate</td>
</tr>
<tr>
<td>$r_i$</td>
<td>inner radius of the friction plate</td>
</tr>
<tr>
<td>$z$</td>
<td>number of friction faces</td>
</tr>
<tr>
<td>$F_a$</td>
<td>axial force on clutch plates</td>
</tr>
</tbody>
</table>
In deriving formula (1) the friction between the friction plates and the outer drum and between the separator and central shaft have been neglected. The axial force is calculated from the measured pressure and the surface area of the axial piston. A uniformly distributed contact pressure and friction shear force have been assumed.

For each test cycle the friction coefficient is defined as the time average of the friction coefficients calculated with equation (1), for the period of a cycle when there is contact and relative movement between the friction and separator plates.

This mean friction coefficient during test 3 is shown in Figure 3. The variation of friction coefficient over time in tests 1 and 2 are analogous. In Figure 3 a running in period of 1000 cycles with rising friction can be observed. The rise in friction coefficient at 12000 cycles is caused by an accidental standstill (some days) of the SAE#II machine. Torsional vibrations cause the larger scatter after 17000 cycles. These vibrations are induced either by dynamic friction instability (clutch shudder) caused by a negatively sloped friction vs. slip curve [3] or by stick-slip, which originates from the difference between static and dynamic coefficient of friction and can also occur when the friction vs. slip curve is positively sloped [4]. Because of the simultaneous variation of contact pressure and slip speed, the slope of the friction vs. slip curve (for a given contact pressure) cannot be determined from the SAE#II friction data, but pin-on-disk tests will show a negative slope of this curve. Considering the negative slope of the friction vs. slip curve, and the fact that the speed measurements are not detailed enough, so that the existence of “stick” periods cannot be detected, the origin of the vibrations remains unknown.

The wear rates calculated from the difference in thickness before and after the 3 performed SAE#II tests are shown in Table 1. From this Table it can be concluded that first a running-in wear with high wear rate or setting of material occurs, followed by a steady state wear with lower wear rate. The wear rates of the friction plates in each test are dependent upon their place in the pack. This is caused by differences in temperatures in the clutch pack (Table 2).

Figure 3: Mean friction coefficient during test 3

<table>
<thead>
<tr>
<th>Thickness change rate (10^-6 mm/cycle) of friction plates</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction plate</td>
</tr>
<tr>
<td>Test 1 (18590 cycles)</td>
</tr>
<tr>
<td>Test 2 (11140 cycles)</td>
</tr>
<tr>
<td>Test 3 (30455 cycles)</td>
</tr>
</tbody>
</table>

Table 1: Thickness change rates after different SAE#II wear tests

![Image](image.png)

From Table 2 it is clear that the maximum separator plate temperatures vary along the clutch pack. This is a result of a non-uniform cooling because the oil, which serves as a coolant is admitted near friction plate 9. Because of the friction in the splines of the clutch however the normal pressure is higher near the axial piston (friction plate 9) than near friction plate 1.

Nevertheless, the wear rates of the different friction plates in the clutch indicate that the effect of decreasing contact pressure along the clutch-pack on the wear of the friction plates is negligible compared to the effect of the variation of temperatures along the clutch-pack (see Table 1).

### 3.2 Pin-on-disk tests

Using the pin–on-disk test rig the influence of the contact pressure and separator plate hardness on the friction coefficient and the wear rate of the friction material was investigated. Pin-on-disk tests were performed at 5 different contact pressures: 1.0, 2.0, 2.3, 2.6 and 2.9 MPa.

Figure 4 shows the friction coefficient, defined as the measured friction force divided by the measured normal force, during 1 cycle after 10 hours of testing. This friction coefficient is decreasing with increasing sliding speed for all contact pressures, which could have yielded the vibrations occurring in the SAE#II tests. Also the friction coefficient is decreasing with increasing contact pressure.

![Image](image.png)

**Table 2: Maximum separator plate temperatures**

<table>
<thead>
<tr>
<th>Separator plate number</th>
<th>Maximum Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>241</td>
</tr>
<tr>
<td>2</td>
<td>296</td>
</tr>
<tr>
<td>3</td>
<td>287</td>
</tr>
<tr>
<td>4</td>
<td>289</td>
</tr>
<tr>
<td>5</td>
<td>279</td>
</tr>
<tr>
<td>6</td>
<td>285</td>
</tr>
<tr>
<td>7</td>
<td>223</td>
</tr>
<tr>
<td>8</td>
<td>240</td>
</tr>
</tbody>
</table>

**Table 2: Maximum separator plate temperatures**

(separator plate 1 is situated between friction plates 1 and 2, plate 8 is situated near the axial piston between friction plates 8 and 9)
At the end of each test (100 hours), the friction coefficients are lower than at the start, but their dependence upon sliding speed and contact pressure remains unaltered.

![Figure 4: Friction coefficients in pin-on-disk tests with different contact pressures vs. rotational speed](image)

<table>
<thead>
<tr>
<th>Contact pressure (MPa)</th>
<th>1.0</th>
<th>2.0</th>
<th>2.3</th>
<th>2.6</th>
<th>2.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>3.36</td>
<td>6.50</td>
<td>2.38</td>
<td>6.09</td>
<td>10.70</td>
</tr>
<tr>
<td>Test 2</td>
<td>--</td>
<td>7.52</td>
<td>3.54</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Test 3</td>
<td>--</td>
<td>6.43</td>
<td>--</td>
<td>--</td>
<td>--</td>
</tr>
</tbody>
</table>

Table 3: Thickness change rate of friction plates coupons \((10^{-5}\text{µm/m})\)

The wear rates for the different contact pressures are shown in Table 3. Wear rates are increasing with contact pressure, though low values occur for 2.3 MPa.

However, the values of Table 3 are an order of magnitude lower than those recorded in the SAE#II test. The wear rates for the SAE#II tests are situated between 4.9 and 17.4 \(10^{-6}\) mm/cycle or (7 m sliding distance per cycle) 124 and 35 \(10^{-5}\) µm/m, while those of the pin-on-disk tests are situated between 2.83 and 7.10 \(10^{-5}\) µm/m.

The influence of the hardness of the separator plate on the wear rate and friction was checked during a test in which hardened separator plates were used with hardness 370 HB but the same initial surface roughness. The test was conducted at an apparent contact pressure of 2 MPa, while the rest of the conditions remained the same as in the previous tests.

Contrary to previous experiments performed by Fish and Lloyd [5] the result was that though changes in surface roughness and friction coefficients remained the same as for the normal hardness, the wear rate was 50% higher i.e. 9.65 \(10^{-5}\) µm/m.

4 CONCLUSIONS

As could be expected, the wear rates in SAE#II test and pin-on-disk tests do not match, mainly because a large difference in surface temperatures exists between the two tests and the sensitivity of the wear rate to temperature. However, the pin-on-disk test indicate that it could be possible to raise the contact pressure used in the clutches (2 MPa) with very little effect on wear rate of the friction material, if care is taken to restrain the surface temperatures of the plates. This would leave a possibility to reduce the size of the clutch.

The pin-on-disk tests point out that hardening of the separator plates does not yield any benefits since it raises the wear rate of the friction material and leaves friction characteristics unchanged.

5 ACKNOWLEDGEMENTS

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6 REFERENCES