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On the significance of operating temperature to the durability of a wet clutch

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1. Introduction

An important question for machine designers is the service life of individual components and how it relates to the service life of the entire machine. Many different aspects can be taken into consideration. The operating conditions will have a large impact on the service life. In the case of heavy duty equipment drivelines, a current development interest is the optimization of the driveline durability with regards to power losses in the driveline.

The wet clutch is one of the components in the driveline where a lot of energy is lost due to viscous shear of the lubricant when the clutch is not engaged. Decreasing clutch size will lower drag losses, but will lead to higher specific loads on the components. For this reason it is necessary to have intimate knowledge about the clutch component ageing behavior. This investigation is focusing on the wear resistance of the wet clutches in the powershift gearbox. The wear behavior of the clutch is of great importance when it comes to determining the necessary dimensions of a clutch for reaching proper service life.

The wet clutch is used to achieve smooth gear changes. The gear changes are achieved through controlled engagements and disengagements of different clutches in the gearbox. As a clutch engages, the difference in rotational speed between input and output shaft is synchronized, and kinetic energy is transformed to heat energy in the sliding interface between friction discs and steel discs in the clutch. Thus, the operating conditions in a wet clutch could be characterized in terms of three different but related quantities: dissipated energy, energy dissipation rate (power) and the operating temperature.

A few different temperatures could be considered when characterizing the wet clutch. The temperature of the lubricant controls the possibility of heat transfer between friction interface and oil sump. High lubricant temperature will also result in accelerated ageing of the clutch lubricant. The temperature in the friction interface is the temperature affecting the top layer of the friction material. There is also a flash temperature in the instantaneous contact between asperities of the friction material and the separator disc. The flash temperature, ΔT, is however, small [1], meaning that it should not be of great interest in terms of clutch durability.

The clutch temperature has been shown to have a correlation to the wear rate of a wet clutch [2]. The effects of energy input, power, inertia and speed was investigated for one material by Lloyd et. al. [3] even though only short term wear was measured. How the power and energy interact with temperature to affect the clutch wear behavior over a longer time span has not been explored. Lubricant temperature has a large influence on the overall clutch operating temperature. A higher lubricant temperature leads to a higher initial temperature and a smaller cooling effect. In turn, this means that a certain interface temperature can be reached at different engagement powers and energies depending on the lubricant temperature.

2. Method

The investigation was conducted by running a number of clutch engagements with different energy, power and oil temperature levels in a wet clutch test rig. Analysis of the results were performed by applying multiple linear regression to the results to estimate the effects.

The friction discs used in this investigation are standard friction discs used in powershift transmissions in heavy duty equipment. The grooves are cut in a waffle pattern and the friction material is a paper based friction material with some synthetic fibers in addition to the cellulose fibers.

2.1. Wear measurement

The equipment used was a custom built wet clutch inertia dynamometer test rig. The test clutch is a two friction disc - three separator disc clutch mounted in an original powershift gearbox shaft from a Volvo CE wheel loader. Temperatures are measured through thermocouples in the separator discs. Torque is measured through a torque arm mounted on the clutch shaft pressing on a load cell. The rotational speed is obtained from the electric motor driving the clutch rotation, allowing for calculation of the engagement power

$$P = M \cdot \omega. \tag{1}$$

The clutch wear is measured by a position sensor measuring the length of stroke of the hydraulic piston used to engage the clutch. The change in piston stroke over a test series is equivalent to the change in clutch pack thickness over said test series. The clutch pack thickness can then be assumed to be the change in friction material thickness due to the low wear of the steel surfaces in comparison. More details about the test setup can be found in Lingesten et. al. [4].

2.2. Clutch wear framework

Earlier work by the authors has shown that the wear of a wet clutch with paper based friction material can experience a two stage wear process [4, 5] as shown in Figure 1. The wear progression can be characterized by an initial wear rate (k_1), a second wear rate (k_2) and a number of engagements until wear rate transition (c). Wear rates are given in terms of mm of material removed from each friction material surface every engagement.

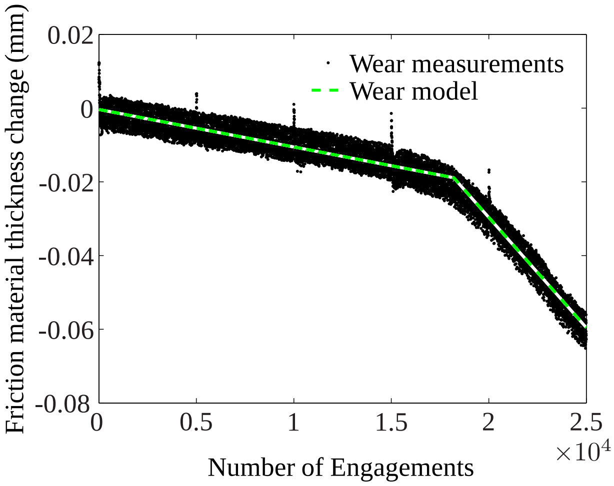

Figure 1: Example wear progression

The test series in Figure 1 shows one measurement value for friction material thickness for every engagement resulting in a cloud of measurement points. To these measurement points the wear model with initial and second wear rates and a transition point is fitted. The scalar values of the wear rates and engagements to the transition point are the numbers of interest in this investigation. Showing how these coefficients change when energy, power and temperature are varied during the engagement is the main objective.

2.3. Model assumption

A model for how the wear rate values depend on the running conditions, i.e. temperature (T), power (P) and energy (E) is formulated. An initial assumption is that the wear rate is linearly dependent on the factors and the products of the factors as

$$k_1 = \alpha_d + \alpha_1 P + \alpha_2 E + \alpha_3 T + \alpha_4 ET + \\ + \alpha_5 PT + \alpha_6 EP \quad (2)$$

$$k_2 = \beta_d + \beta_1 P + \beta_2 E + \beta_3 T + \beta_4 ET + \\ + \beta_5 PT + \beta_6 EP \quad (3)$$

$$c = \gamma_d + \gamma_1 P + \gamma_2 E + \gamma_3 T + \gamma_4 ET + \\ + \gamma_5 PT + \gamma_6 EP. \quad (4)$$

The k_1, k_2 and c parameters are measured at different P, E and T levels and coefficients α_x, β_x and γ_x are fitted through least squares to obtain a linear model for the wear parameter. Subscript d indicates a model constant.

2.4. Test matrix

The test series consists of between 25000 and 80000 engagements. The reason for the different test cycle lengths is the fact that under some conditions the wear rate transition occurs much later than under other conditions. For some of the test conditions the wear rate transition was not reached before the end of the test series while for other test conditions all the friction material was worn off before the end of the test series.

The test matrix was a two level factorial experiment design with three factors (energy E, power P and lubricant temperature T). However, as it is difficult to match temperature and power levels at different energy levels, the experiment allows for some variation in the levels. The power and energy is measured as specific values, that is power per unit area and energy per unit area. The temperature factor controlled was the lubrictant temperature. The factor levels are summarized in Table 1.

Table 1: The factor levels for the experiments

	High	Low
Maximum power	8.6 MW/m^2	6.14 MW/m^2
Energy	1.6 MJ/m^2	1.1 MJ/m^2
Oil temperature	80 °C	45 °C

Since there are large differences in the order of magnitude for the different factors it is necessary to code the variables before an attempt at model fitting is done. The power variable is coded in such a way that the low power level has the value -1 and the high power level has the value +1:

$$P_{Co} = \frac{P - (P_{max} + P_{min})/2}{(P_{max} - P_{min})/2}. \quad (5)$$

The energy and temperature variables are scaled in the same way.

3. Results

Figures 2-4 show the measured wear coefficients and how they compare individually against the abolute values of the different parameters. Some trends can be discerned from these figures. In general it seems like increased energy input generally lead to higher wear rates and quicker wear rate transition.

Similarly, increased power leads to increased wear rates even though it is harder to asess a difference due to the large scatter in power levels over the parameter range.

The temperature-wear rate plots are rather constant, but we can see that the higher wear rates occur at the highest temperatures while the lowest wear rates are found at low temperatures.

There are also fewer measurement points available for the second wear rate and point of transition. The reason is that at the trend is that higher amount of engagements is required to achieve wear rate transition at low energies. Therefore some of the earlier experiments were not extended enough to enter into the high wear rate regime, only yielding an initial wear rate value.

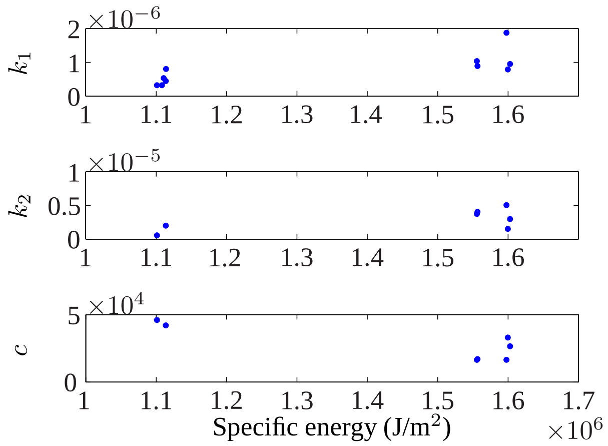

Figure 2: Wear coefficients plotted against energy

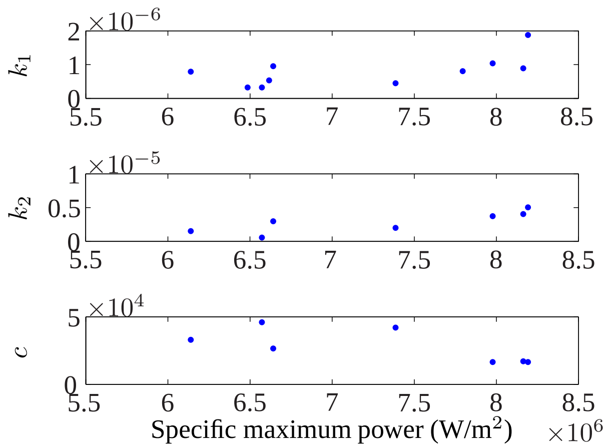

Figure 3: Wear coefficients plotted against power

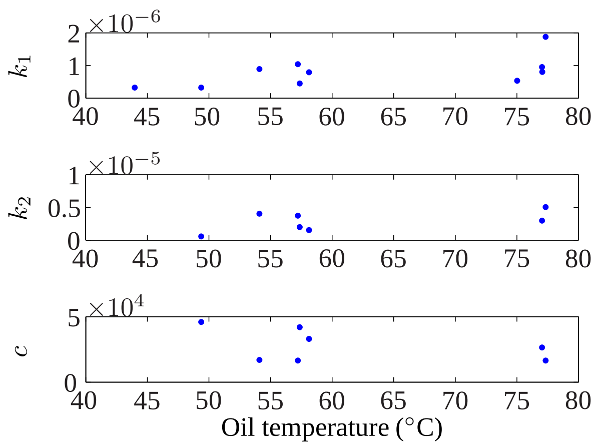

Figure 4: Wear coefficients plotted against lubricant temperature

Fitting the model coefficients from equation (2)-(4) to the result matrix as plotted above yields.

$$\alpha = \begin{bmatrix} 0.7039 \\ 0.1228 \\ 0.2992 \\ 0.2736 \\ 0.0111 \\ 0.3072 \\ 0.1274 \end{bmatrix} \cdot 10^{-6}, \beta = \begin{bmatrix} 0.0031 \\ 0.5445 \\ 0.2717 \\ -0.6004 \\ 0.6975 \\ 0.0374 \\ -0.4391 \end{bmatrix} \cdot 10^{-5}$$

$$\gamma = \begin{bmatrix} -0.0091 \\ 5.1213 \\ 1.9732 \\ -9.4061 \\ 9.7896 \\ 0.7730 \\ -6.4864 \end{bmatrix} \cdot 10^{4}$$

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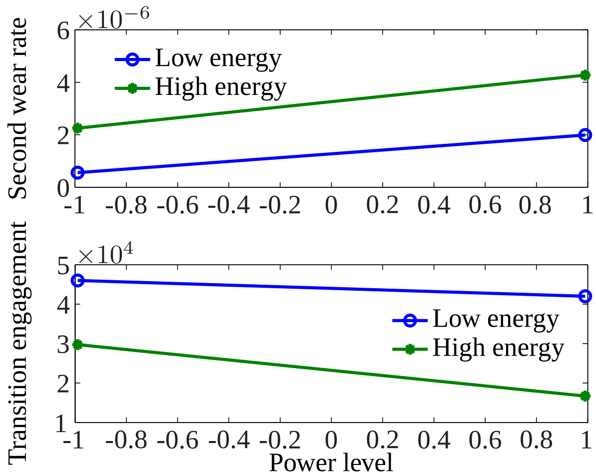

Figure 5: Interaction plot for power and energy with regards to second wear rate and wear rate change

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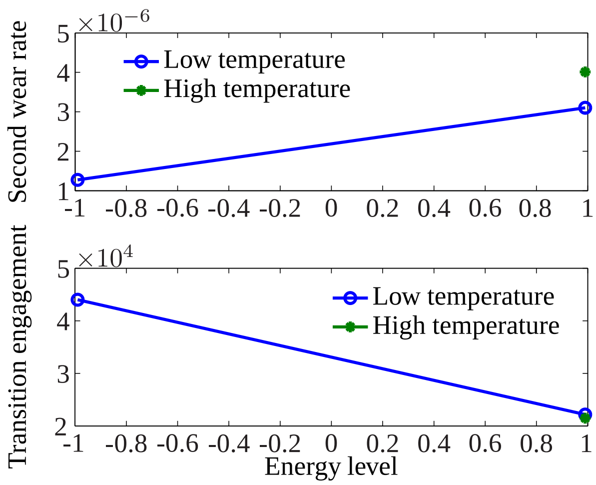

Figure 6: Interaction plot for temperature and energy with regards to second wear rate and wear rate change

Figures 5-6 illustrate the interactions between energy and temperature and power and temperature. The results shows that there is an interaction between engagement energy and power in terms of when the wear rate transition takes place. The interaction between temperature and energy is hard to assess but the what can be said is that the oil temperature does not influence the wear rate transition point when operating at the high energy level. The interaction between power and oil temperature on the second wear rate k_2 and wear rate transition point c is weak as suggested by the value for the interaction coefficients β_5 and γ_5.

For the interactions at the initial wear rate stage it is possible to say that the interaction between oil temperature and energy is weak. The interactions between power and oil temperature as well as the interaction between engagement energy and power are significant.

4. Discussion

The effects of temperature, energy and power are quite straightforward. Higher energies, power and temperatures lead to higher wear rates. These results are in line with the results shown in previous investigations. Higher load on the material yields a faster material degradation and wear.

Several interaction effects can be said to be significant when looking at the results. They all have in common that increasing one engagement parameter value will also increase the effect of the remaining engagement parameters. For the initial wear, the energy and temperature parameters are however nearly decoupled with very low interaction. For the second wear regime however, it seems like there is an interaction between the energy and oil temperature parameters. On the other hand the interaction between power and oil temperature is weak in the second wear regime. These results are important to consider when looking at how to set the operating parameters for an optimized transmission.

Due to the high amount of time required to measure the wear rate and wear rate transition behavior for one measurement point (5000 clutch engagements per day maximum), only one measurement per factor level has been performed so far. A higher number of tests would consolidate the results, giving a better understanding of the interactions. In addition to this, analysis of the significance of the effects and estimation of the random measurement errors can be done when tests are repeated.

5. Acknowledgements

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