高性能樹脂歯車の摩耗予測に関する取り組み

Predicting wear for high performance plastic gears

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1. Introduction

One of the materials within the portfolio of Envalior is Stanyl® polyamide 4,6 resin. This polyamide-based material has a long-standing track record in a large variety of high performance gear applications. To be successful when developing applications, it is necessary to provide directions to the end-user concerning the application's long-term performance in operation. For a gear application this resolves in statements about the tribological behavior of the system. For example, to predict wear of gears it involves the tribological behavior of the material, such as a wear rate. This paper describes and discusses the ongoing work on a predictive framework, shown in Figure 1, which combines Computed Aided Engineering output (CAE) and experimentally determined wear rates on a tribological set-up, the Disc-to-Disc setup (D2D). Here we present our main learnings from the attempt to connect the experimental results on the D2D setup to the gear tester.

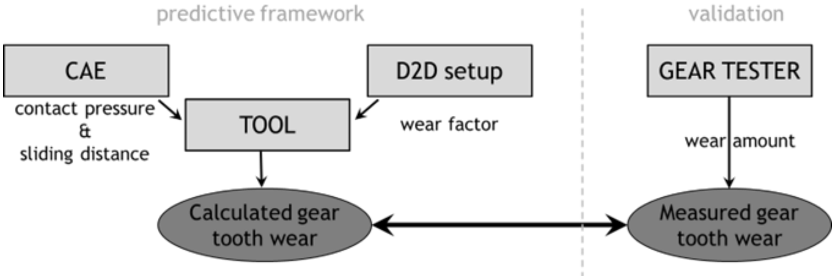

Fig. 1　Model scheme of the predictive framework and the validation step

2. Background and Theory

The wear model that is used is Archard's wear equation[1)], which states that the total wear h is linear dependent on pressure p, sliding distance s and a wear factor k via:

$$h = k \cdot p \cdot s \tag{1}$$

The wear factor k is regarded as a material constant, the pressure and sliding distance as a result of geometry and material that can be computed either analytically or using CAE. The total cumulative wear depth, Δh, after N cycles follows from:

$$\Delta h(N, p, s) = k \cdot p \cdot s_{cycle} \cdot N \tag{2}$$

In this equation often the maximum pressure and total sliding distance per mesh cycle is used. However, sliding contact between two tooth flanks happens at high speed and includes a rotational displacement, causing local time-dependent pressure peaks in combination with accumulating sliding distances. As a result, also the product of pressure and distance will vary in time, and therefore integration in time is preferred instead:

$$\Delta h(N, p, s) = k \cdot \int_a^b p\Delta s dt \cdot N \tag{3}$$

In equation (3) a is the time in a mesh cycle when a local material point encounters its mating tooth flanks and b the moment when exiting the contact. Δs is the increase in sliding distance during one time increment, and p the pressure at that increment. In the remainder of this paper $p\Delta s$ is referring to $\int_a^b p\Delta s dt$.

3. Methods

A polyamide 4,6 resin without glass fibers was used. All tests were done by mating two geometries (discs or gears) of the same material under dry-running conditions. Gears were injection molded with a 1 mm module, consisting of 30 teeth each. The addendum was 1 mm, the dedendum 1.25 mm and a face width of 6 mm. For both the driven and driver gear the same geometry is used. For Disc-to-Disc (D2D) test, Discs were milled from an injection molded plate with a 9 mm thickness and the thickness was reduced to 8 mm. The diameter of the discs was 35 mm. Sliding wear was measured on a custom-built machine, based on a twin-disc test rig, the so-called Disc-to-Disc (D2D) setup. Two motors rotate the two discs at a fixed, but different speed. On simulations, 2D plane stress FEM approach was used to model the gear and D2D setup to predict and respectively validate the contact pressures and sliding distances. Abaqus was used as solver.

4. Results

Fig. 2 shows the average wear, including the standard deviation from all 30 teeth after 5 million cycles at 0.2 Nm and 0.3Nm, compared to the position on the gear flank. The variation in wear between the teeth is caused by variations in diameter along the circumference of the gear due to the injection molding process.

From the D2D-experiments the wear factor can be derived from the mass loss of both discs or by using the derivative of the displacement of the motors. The latter one is plotted in Fig. 3 as the dimensional change of the discs as a function of run time, indicating a linear decrease of the diameter after an initial run-in period at a constant temperature of 130°C.

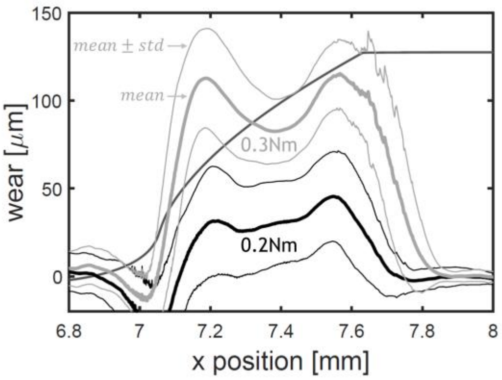

Fig. 2 Measured absolute wear of the gear after 5 million cycles and loaded with 0.2 Nm and 0.3 Nm

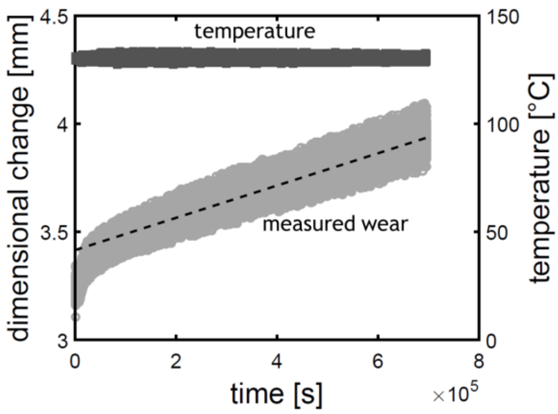

Fig. 3 Measured diameter change (wear) on the D2D setup at a constant temperature of 130°C, including a trend line(dashed) that shows a linear wear rate

5. Discussion

The similarity between the wear factors in the D2D experiments and the FEM computations and gear experiments is promising, however, it does not explain the large amount of wear at the pitch point where $p\Delta s$ is very small as shown in Fig. 4. The framework assumes the wear factor is a material property and $p\Delta s$ depends on both material and geometry. To investigate the effect of wear on $p\Delta s$, the worn shapes are taken from the gear experiments, and meshed in the FEM setup accordingly, both for the driver and driven gear. An important learning is demonstrated in Fig. 5b, which visualizes the VonMises equivalent stresses for a gear with initial dimensions (not worn) and a gear with a severely worn profile, both loaded at 0.3 Nm at the same moment during the mesh cycle. One can see that the pressure distribution along the flank is different for the initial gear shape and the worn shape. Instead of one contact location, the contact is at two either sides of the flank, causing the material at the original pitch point not even being in contact anymore.

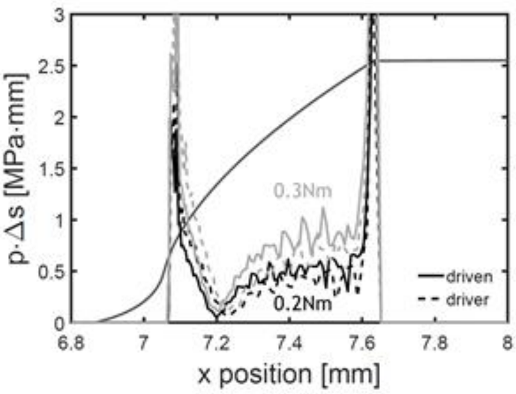

Fig. 4 $p\Delta s$ results obtained from FEM simulations for 0.2 Nm and 0.3 Nm

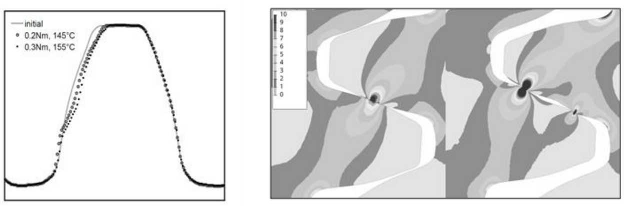

Fig. 5 a) Measured tooth profiles, initial and tested for 5 million cycles for two torque levels, b) VonMises stress [MPa] contour plots with the initial geometry and a measured worn geometry (0.3 Nm).

6. Conclusions and Outlook

The D2D setup can provide relevant wear factors for unfilled systems, however, quantitative prediction of gear wear was not possible at the time of writing this paper. With current framework and FEM approach it was shown that, due to changing contact conditions in the application, a representative wear factor from gear experiments can only be obtained using iterative wear analysis, albeit a highly impractical procedure. Future work will focus on closing the gap between experimental wear factors and apparent wear factors with possibly other calculation methods using dedicated gear design software.

References

1) J.F. Archard, J. Appl. Phys. 24 (1953) 981–988.