Accelerated Testing and Temperature Calculation of Plastic Gears

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1. Introduction

Plastic gears have been in use since the 1950s, and their popularity has increased significantly in the last few years. The mass production of plastic gears using injection moulding and new plastic materials, have additionally increased the application of plastic gears in automotive and medicine areas. The primary advantages of plastic gears are low manufacturing costs for serial production, no need for external lubrication and good noise damping properties [1, 2]. There are also some disadvantages that limit the use of plastic gears: inferior mechanical and thermal properties compared to typical gear materials, lower operating temperatures and lower manufacturing tolerances [3-5].

A wide variety of different types of polymer materials (PA, POM, etc.), different reinforcements (fibres, nanoparticles, etc.) and internal lubricants (PTFE, etc.) can be used to tailor a polymer for a specific application. However, due to the large number of different material combinations possible, it is difficult to make an optimal material selection for a certain gear drive.

In the literature and guidelines, the allowable gear endurance limits (for root and flank) are mainly given for PA and POM materials [2, 6]. Only a few attempts have been made to compare the allowable endurance limits of the standards with the results obtained from gear testing [5, 7]. The common conclusions are, that great discrepancies can exist.

In order to appropriately design gears, it is in most cases advisable to perform gear testing in order to evaluate material selection, gear temperatures as well as allowable root and flank stresses. As this can be very expensive and time consuming, it is recommended to combine standard test with accelerated test procedures.

1.1 Problem definition

To determine the polymer gear endurance limits with acceptable testing times and costs, an accelerated testing procedure is proposed. The method was tested on polymer gears made of unreinforced and reinforced polyamide 6 (PA6, PA6-30) and polyacetal (POM) materials under different testing conditions (up to 2500 rpm and 0.82 Nm). The test results from lifetime testing (max. allowable root stress and gear bulk temperature) were compared also with the values described in the VDI 2736 guideline.

2. Accelerated testing method for gears

To have an efficient testing procedure, a balance needs to be found between reliability of the results and the testing time. A proposed testing method for polymer gears can have 2 types of tests: tests with step increased load and lifetime tests. Such testing can lead to a reduced number of tests and also provide reliable test results for different applications.

2.1 Testing with step increased load

When running tests with increased step load, the tangential speed in the test must be similar to the tangential speeds expected in the final application. The load during test is increased in steps. Starting load (or equivalent tooth root stress) should be determined with preliminary tests. For PA6 and POM materials, the tooth root stress for the starting load should be between 15 MPa and 20 MPa. This is selected based on expected allowable tooth root stresses that are presented in the VDI 2736. The torque increase in each step test should be around 20% of the initial load. The duration of the test at 1 load level is determined based on the temperature of the tested gears. The duration must ensure that the gear temperature is stable before moving on to higher loads. We suggest testing for at least $2 \cdot 10^5$ cycles before increasing the load level. The load level is increased until one of the gears fails, either by temperature overload, fatigue or excessive wear.

In case that the first result does not fulfil the requirements (temperature is not stable, gears immediately fail, high wear …), the step load test is repeated using a pair of gears made from different materials or with some other modification (changed load step procedure, modified centre distance …) [8].

The purpose of the tests with step increased load is:

- to determine tribological compatibility of the selected materials for application;
- to check theoretical temperature calculations acc. to the VDI 2736;
- to calculate back the corresponding coefficient of friction based on the temperature measurements;
- to determine the max. allowable operating temperature of the materials tested.

This type of test can be used to efficiently check new material combinations, for which tribological properties (coefficient of friction, wear rate and wear mechanisms) are not known.

2.2 Lifetime tests

If more data is needed about a certain material combination (or material), lifetime test should be performed. Lifetime tests are performed at constant load and constant speed. Usually at least 3 different speeds and load levels are used. With this type of testing, S-N curves are generated. Compared to the tests with increased load, the time to complete lifetime tests is significantly longer.

The purpose of the lifetime tests is:

- to determine the temperature dependant S-N curves (gear temperature must be controlled – either by climate chamber or by adjusting rotational speed);
- to adjust temperature calculation formula using a measured COF (from tribological tests) – to get a better match between measured and calculated gear temperatures;
- to determine gear failure modes and gear temperature behaviour.

3. Experiments

3.1 Gear test rig

Tests were performed on a purpose built, open-loop testing machine, which is schematically shown in Figure 1. The test rig consist of a brake shaft and torque shaft. Braking torque is provided by hysteresis brake (max. 1 Nm), while the speed is controlled with servomotor (max. 4000 rpm). Motor and brake are connected (via coupling) to the shafts on which the driver/driven gear is mounted. The design of the test rig allows precise positioning of the test gears in x and y directions.

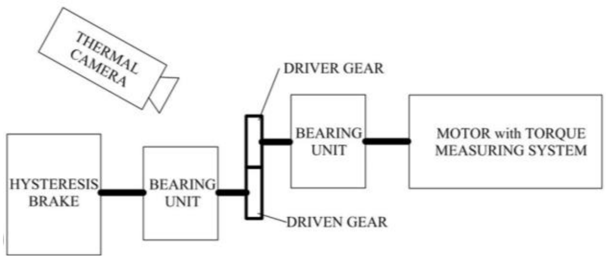

Fig. 1: Schematic of a polymer gear test rig.

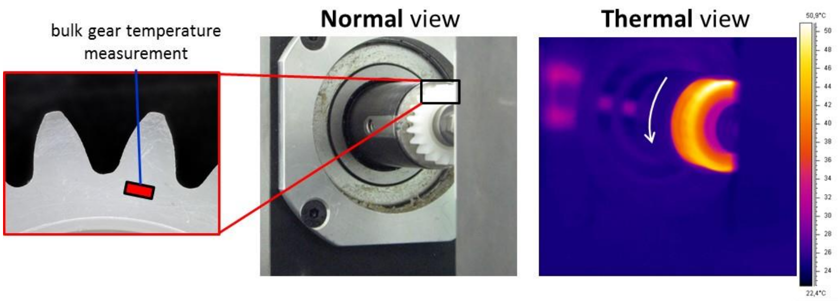

Fig. 2: Driver gear temperature measurement.

A thermal camera Flir A320 (Flir, USA) was used to measure the temperature of the driver gear. The bulk gear temperature was measured from the side of the gear, as shown on Figure 2. The area of temperature measurement was 2 mm x 1 mm. The emissivity of the test gear materials was measured and was set to a constant value of 0.95. The same gear body temperature was assumed for driver and driven gear.

3.2 Test gear and materials

Due to its common use in gears, an involute test gear geometry was selected. A standard pressure angle of α = 20° was used with no profile shift applied to the geometry (x = 0 mm). A detailed specification of test gear can be found in Table 1. Because of the thermal expansion of the polymer materials, the operating centre distance was set to 20.05 mm.

Table 1. Specification of the test gear.

Module	1 mm
Number of teeth $z_1 = z_2$	20
Diameter of tip circle	22 mm
Pressure angle	20°
Face width	6 mm
ISO gear quality	10-11

Three different polymer materials were used for the injection moulding of the test gears. Based on tribological tests [9], a preferred combination POM/PA6 material was selected. The first material always refers to the driver gear (POM) and the second to the driven gear (PA6). Selected materials for testing were Polyamide 6 (PA6, Ultramid® B3S, BASF), Polyamide 6 + 30% GF (PA6-30, Zytel® 73G30 HSLNC, DuPont) and Polyacetal (POM, Delrin® 500P, DuPont).

3.3 Testing conditions for step test

The rotational speed of 1176 rpm was selected based on the results from our previous tests. This can also represent an approximate rotational speed of the final gear application. The starting load level was 0.30 Nm and increased every $2 \cdot 10^5$ cycles by a value of 0.05 Nm. Several preliminary measurements were also conducted to confirm parameter adequacy. The starting load level of 0.30 Nm represents the low load level for the selected materials and gear geometry. Different material combinations were tested (PA6/PA6, POM/POM, PA6-30/POM) at room temperature (23°C) and without lubrication.

3.4 Testing conditions for lifetime testing

The torque and speed for the lifetime tests are shown in Table 2. The torque and speed are increased by a factor of 1.4 in order to enable better comparison of the results. The speed was between 600 rpm and 2305 rpm, while the torque load was varied between 0.30 Nm and 0.82 Nm. The tests were performed at 23°C and without lubrication.

Table 2. Testing conditions for lifetime tests (POM/PA6 combination).

		Test torque			
Rotational speed, rpm	**Tang. speed,** m/s	0.30 Nm	0.42 Nm	0.59 Nm	0.82 Nm
600	0.63			37	52
840	0.88		37	52	72
1176	1.23	37	52	72	101
1646	1.72	52	72	101	142
2305	2.41	72	101	142	
		Transmitted power, W			

4. Results

4.1 Temperatures at tests with increased step load

Figure 3 shows driver gear temperature at step test for different material combinations. Depending on the material combination, completely different gear behaviour was observed. Combinations POM/PA6-30 and POM/PA6 had the most stable operation; the temperatures were moderate and without any spikes, resulting in the highest cycles to failure. For combination PA6/PA6, high temperatures were observed as a result of high coefficient of friction for this material combination. The operation was inconsistent (high temperature oscillations), which resulted in failure at 0.45 Nm. Combination POM/POM had the lowest operating temperatures, but had the highest wear. Other two combinations (PA6-30/PA6 and PA6-30/PA6-30) also resulted in higher temperatures and had semi-stable operation.

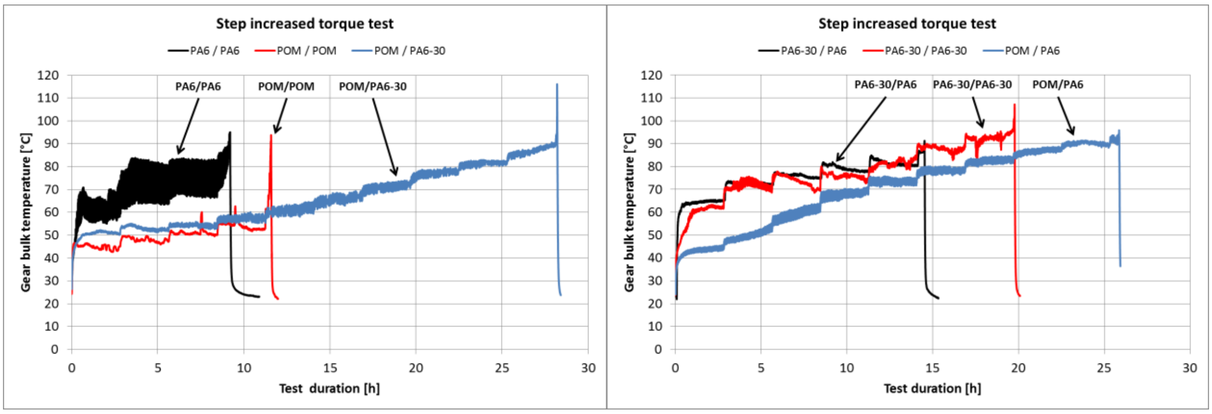

Fig. 3: Driver gear temperature measurements at step tests.

Measured gear temperatures from step test (Figure 3) were used to calculate the coefficient of friction (COF) for different material combinations. Calculation was performed using equation *(1)*. For each step of the test, corresponding COF was calculated. At the end, an average COF was calculated for every material combination. Results are shown in Table 3. The lowest COF was calculated for combinations POM/PA6, POM/POM and POM/PA6-30. On the other hand, PA6/PA6 combination had the highest COF. This is also in agreement with tribological investigations of the same materials, were similar tribological phenomena was observed [10]. POM/POM combination had very high wear coefficient both at gear and tribology testing. Table 3 also shows the load level, at which certain material combination failed.

Table 3. Measured step test gear temperatures and calculated COF.

		Torque					COF	
	Torque at failure, Nm	**0.30 Nm**	**0.40 Nm**	**0.50 Nm**	**0.60 Nm**	**0.70 Nm**	**avg. COF from calculation**	**acc. to VDI 2736**
PA6/PA6	0.45	**71** *0.47**	**83** *0.43**	/	/	/	**0.45**	**0.40**
POM/PA6	0.75	**46** *0.22**	**64** *0.29**	**75** *0.30**	**84** *0.29**	**92** *0.28**	**0.28**	**0.18**
POM/POM	0.50	**47** *0.23**	**61** *0.27**	/	/	/	**0.25**	**0.28**
PA6/PA6-30	0.55	**66** *0.42**	**77** *0.39**	**85** *0.36**	/	/	**0.39**	**/**
POM/PA6-30	0.75	**52** *0.28**	**55** *0.23**	**65** *0.24**	**74** *0.25**	**83** *0.25**	**0.25**	**/**
PA6-30/PA6-30	0.65	**63** *0.38**	**78** *0.40**	**85** *0.36**	**92** *0.33**	/	**0.37**	**/**
		Gear bulk temperature, °C *Calculated COF**						

**COF – calculated coefficient of friction according to the VDI 2736*

If we compare the calculated COF values with the recommendations in the VDI 2736, we can see the biggest difference for material combination POM/PA, while for the other two combinations, COF values are close together. With a reverse calculation of COF from the temperature calculations, a more realistic COF can quickly be determined and used for a more precise gear temperatures, even for materials not listed in the VDI 2736.

4.2 Lifetime testing of POM/PA6 material combination

Full lifetime testing was only performed for POM/PA6 material combination. The resulting cycles to failure are shown in Table 4. It can be seen, that both speed and load decrease the number of cycles to failure. Gears tested at 0.59 Nm/2305 rpm and 0.82 Nm/1176 rpm resulted in immediate failure as a result of temperature overload. For the other testing conditions, gears failed as a result of root fatigue on driven gear PA6.

Table 4. Cycles to failure of PA6 material for POM/PA6 gear combination with measured tooth root temperatures.

	0.30 Nm	**0.42 Nm**	**0.59 Nm**	**0.82 Nm**
600 rpm			**5.43** *61°C*	**0.45** *80°C*
840 rpm		**10.98** *59°C*	**3.42** *65°C*	**0.30** *87°C*
1176 rpm		**7.30** *60°C*	**1.59** *78°C*	**0.01**
1646 rpm	**15.56** *61°C*	**3.62** *73°C*	**0.76** *89°C*	
2305 rpm	**11.75** *76°C*	**2.35** *87°C*	**0.01**	
	Cycles to failure [10^6, 90% survival rate]			

4.3 Comparison of allowable root stresses and operating temperatures with VDI 2736

Calculation of gear operating temperatures according to the VDI 2736 and comparison with measured values is shown in Table 5. Values in Tables *b*-*d* present the differences of calculated temperatures compared to the measured values (Table 5*a*).

Eq. 1 presents the gear tooth root temperature calculation from the VDI 2736. The standard values in the VDI 2736 for POM/PA combination are: $\mu = 0.18$, $k_{\theta,Fuß} = 2100$ and $c = 0.75$.

$$\vartheta_{root} \approx \vartheta_0 + P \cdot \mu \cdot H_V \cdot \left(\frac{k_{\vartheta,root}}{b \cdot z \cdot (v_t \cdot m_n)^c}\right) \cdot ED^{0.64} \qquad (1)$$

Table 5b shows a temperature calculation with factors specified in the VDI 2736. It can be seen, that there are significant differences between measurements and calculation. A calculated standard deviation was 20°C. Table 5c shows results with modified coefficient of friction (COF). The COF value of 0.36 for PA6/POM combination was taken from the literature [10]. When using a COF of 0.36, the calculated temperatures better match the measured data. The calculated standard deviation was 12°C.

In order to get a better correlation between measured and calculated temperatures, a VDI 2736 calculation factors were modified. Root heat transfer coefficient $k_{\theta,Fuß}$ was modified from 2100 to 1505 and the coefficient c was reduced from 0.75 to 0.36. With that corrections, we were able to get much better correlation between measurements and calculations. The calculated standard deviation in that case was only 4°C. The biggest difference was at high load and high speed, probably as a result of hysteresis heating and high tooth deformations, which are not taken into consideration by the VDI 2736 calculation.

Table 5. Comparison between measured and calculated gear temperatures.

a) Measurement with thermal camera

	0.30 Nm	0.42 Nm	0.59 Nm	0.82 Nm
600 rpm			61	80
840 rpm		59	65	87
1176 rpm		60	78	x
1646 rpm	61	73	89	
2305 rpm	76	87	x	
	MEASURED temperature, °C			

b) Standard VDI 2736 calculation

(μ = 0.18, $k_{\theta\ Fuß}$ = 2100, c = 0.75)

	0.30 Nm	0.42 Nm	0.59 Nm	0.82 Nm
600 rpm			-7	-14
840 rpm		-12	-8	-17
1176 rpm		-11	-18	x
1646 rpm	-18	-21	-26	
2305 rpm	-31	-33	x	
	COMPARISON with measurements, °C			

Calculated standard deviation: ***20°C***

c) Modified VDI 2736 calculation

(μ = 0.36, $k_{\theta\ Fuß}$ = 2100, c = 0.75)

	0.30 Nm	0.42 Nm	0.59 Nm	0.82 Nm
600 rpm			+24	+29
840 rpm		+12	+26	+30
1176 rpm		+15	+19	x
1646 rpm	+3	+7	+14	
2305 rpm	-9	-2	x	
	COMPARISON with measurements, °C			

Calculated standard deviation: ***16°C***

d) Modified VDI 2736 calculation

(μ = 0.36, $k_{\theta\ Fuß}$ = 1505, c = 0.36)

	0.30 Nm	0.42 Nm	0.59 Nm	0.82 Nm
600 rpm			-1	-5
840 rpm		-3	+4	0
1176 rpm		+4	+2	x
1646 rpm	-2	+1	+5	
2305 rpm	-8	-1	x	
	COMPARISON with measurements, °C			

Calculated standard deviation: ***4°C***

*calculated temperature is bigger (**+**) or smaller (**-**) than the measured temperature*

Table 6 shows tooth root stresses, which were calculated from the load and gear geometry based on the VDI 2736.

Table 6. Torque load level and corresponding tooth root stress according to the VDI 2736.

Torque, Nm	**0.30**	**0.42**	**0.59**	**0.82**
σ_F, MPa	17.6	24.6	34.6	48.1

A comparison of measured allowable tooth root stresses with the VDI 2736 was also performed. The results are shown in Table 7. It can be seen, that a significant differences exist between standard and our measurements. The biggest differences are at small loads (at 0.30 Nm, difference for a factor of 2.6), however with increasing load, the differences get smaller (at 0.59 Nm, difference for a factor of 1.4). The differences are probably a result of different material type (PA6 in our tests and PA66 in the VDI 2736). In addition to that, it could also be true, that the allowed tooth root stresses according to the VDI 2736 are too optimistic.

Table 7: Allowable tooth root stresses for measured PA6 material compared to PA66 material from the VDI 2736.

Torque, Nm	M	0.30	0.42	0.42	0.59	0.59
Speed, rpm	v	1646	1176	1646	840	1176
Cycles to failure, 10^6	N	15.56	7.30	3.62	3.42	1.59
Gear temperature, °C	T	61	60	73	65	78
Allowable tooth root stress acc. to VDI 2736, N/mm^2	$\sigma_{FE(VDI)}$	**46**	**50**	**46**	**52**	**48**
Allowable tooth root stress acc. to tests, 90 % survival rate, N/mm^2	$\sigma_{FE(TEST)}$*	**17.6**	**24.6**	**24.6**	**34.6**	**34.6**
Difference $\sigma_{FE(TEST)}/\sigma_{FE(VDI)}$		**38%**	**49%**	**53%**	**66%**	**72%**

**Tooth root stress calculated based on torque and gear geometry according to VDI 2736.*

4.4 Gear failure mechanisms

POM/PA6 polymer gears typically fail as a result of fatigue. However, if improper material combination in selected (like PA6/PA6), polymer gears can also fail due to temperature overload. Fatigue can be measured using lifetime tests and is predictable. Temperature of the gears can also be calculated, but the results are questionable. In most cases, the melting failure occurred during the first hour of the gear test. If the gear pair survived the first phase and the gear body temperature is stabilised, fatigue will most often be the failure mode.

Wear was recognised as a damage mechanism only for some gear pairs, e.g., POM/POM. In this case, the wear rate was high even at acceptable load levels from the tooth root stress and temperature point of view.

For other combinations of materials, root fatigue was recognised as the main failure mechanism.

5. Conclusions

The paper presents an updated testing procedure for polymer gears, which enables a more accurate determination of gear temperatures and thus also a better prediction of gear lifetime. When using a preliminary step test, a combination with good tribological properties can be determined. In addition to that, COF can also be calculated and later used for a more precise temperature calculation.

A full lifetime testing (together with temperature measurements) was performed for POM/PA6 material combination. The results show a great effect of load and temperature on the lifetime of polymer gears.

Comparison of measured and calculated gear bulk temperatures showed great differences. However, with modification of the VDI 2736 temperature calculation formula (change of root heat transfer coefficient and factor c in the equation), we were able to get a good correlation between the measured and calculated gear temperatures.

When comparing measured allowable tooth root stresses with the VDI 2736, again significant differences occurred (up to 62%). The differences, however, decreased with increasing load. The differences are probably a result of different material type. In addition to that, it could also be true, that the results from the VDI 2736 are too optimistic.

In our research, polymer gears typically failed either as a result of root fatigue (POM/PA6) or as a result or temperature overload. Significant wear was only observed for POM/POM material combination.

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