Comparison of The Fatigue Life of PA 6 Gears

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Abstract
Plastic materials are increasingly used as a gear wheel material. The advantages of plastic gears are such that they can be used under unlubricated conditions and also they are light and corrosion resistant. Thermal fatigue in plastic gears is a unique type of damage. The generated heat should be considered during performing theoretical calculations of plastic gears. In this study, the BS 6168 standard and other important methods used in the analytical calculations of plastic gears were investigated. The tooth root stresses and fatigue life of the gears with a 6 mm module were determined using the finite element method according to the BS 6168 standard and the fatigue life of specimens were compared.

Keywords: Polyamide Gears, Calculation Methods of Plastic Gears, BS6168, ANSYS

1. INTRODUCTION
Gears are power and motion transmission elements. As well as in the home and office tools, plastic gears are frequently used in the food industry where high level of noise is dominant, because they are corrosion resistant, lightweight, and easily producible and they can function without lubrication [1].

Damage mechanisms in plastic gears are crack formation from the tooth root, crack formation on the base circle and wear. The damage formed on the base circle is unique for polymeric gears. The fatigue in most of the plastics does not occur by crack propagation, but plastics are damaged by softening in the material because of the lack of transmission of the heat caused by internal friction due to viscoelastic behavior of plastic. This damage, called thermal fatigue, is defined as the strength decrease due to high local temperature formation at the contact point on the base circle. The formation of softening damage depends on the frequency of the applied load, the stress amplitude and loss compliance. Plastics that have high loss compliance (J_2>0.5*10^{-10} m^2/N) exhibit softening damage. Polypropylene (PP), polyethylene (PE) and nylon (polyamide) fall into this group of plastics [2-3].

It was observed that thermal fatigue on the base circle occurs especially in gears operating under unlubricated conditions. In the standards related to the calculations of plastic gears, the damage on the base circle is ignored and these standards take into account other three damage types, namely, the fracture from the tooth root, wear and flaking. There is limited information available about glass-reinforced gears, however it was determined in various studies that there is an increase in tensile strength for brittle damage. Furthermore, the wear with glass reinforcement is a primary problem [4].

In a study performed by Can, PP specimens were studied at a frequency of 0.75 Hz, which did not cause softening and experiments were carried out by allowing a deformation that was 25% of the tooth root cross-section. As a result of the experiments, plastic deformation was observed in some specimens while tooth root fatigue fracture occurred and progressed in others [5].

Considering a plastic gear/steel gear pair, the heat caused by friction is more easily removed due to the good conductivity characteristics of metals. Because the average surface roughness cannot be greater than 2 μm in surface-hardened gears, the wear on plastic gears will be less. The acetal copolymer gears with improved properties have better sliding properties than standard plastic gears and therefore they wear less. Plastic gears can be coupled with gears made of steel or non-ferrous metals (like copper and aluminum alloys) in mechanisms where the total operating time is short [6].

The plastic gear wheels are either lubricated once during assembly or have to operate without lubrication. When boundary friction or dry friction occurs, the material pair to be used has a great significant effect on the amount of friction and wear. During forming the material pair, low friction, smooth slip, minimum wear and high thermal conductivity is desired for rapid removal of the friction-induced heat [5].

It is a known fact that gear wheels become unusable more likely due to severe abrasions rather than tooth fracture. Wear is caused by deformations due to high surface pressure between the gripping tooth surfaces. Determination of the approximate amounts of elastic deformation formed under those conditions to the plastic limit is very important for the forced machine elements such as gear wheels. It was observed that Hertz pressure and deformation increases in external and internal gear wheels with increasing the number of gears and profile sliding factor of the module; the increase in the ratio of the cycles number decreases the Hertz pressure and elastic deformation in external gear wheels while it increases those values in internal gear wheels [7].

In coupling of polymers with metal counterface, one of the most important factors affecting wear is the surface roughness of counterface. As a result of the wear tests of extrusion PA 6 with stainless steel discs, it was seen that PA 6 formed a continuous transfer film on the counterfaces, the wear rate increased when the axial surface roughness value was between 0 and 1.5 μm, and the wear rate decreased when it was between 1.5 and 3 μm [8].

The contact behaviors of polymeric gears were analyzed using analytical calculations performed with the ABAQUS software package according to the finite element method and BS ISO 6336

standard. As a result, the simulated values and calculations made according to the BS ISO 6336 standard were compared and it was determined in particular that the analytically derived values were independent of the contact hardness and substantially different for bending stresses. Differences between the stresses obtained from simulations and those analytically calculated indicate that the effects of the load distribution and friction (especially in the dry operation) due to the excessive deformation formed in the teeth should be taken into consideration in preparing polymeric gear standards [9].
In various lubrication conditions, thermal analyses were performed by finite element method to estimate the bulk temperature and flash temperature of the gear surface and it has been determined that the results were consistent with those obtained from the literature studies. It was found that the finite element model can be used with high accuracy to predict the load carrying capacities of polymeric gears, whose mechanical properties depend on the critical temperature [10].

2. CALCULATION METHODS OF PLASTIC GEARS

Standards, such as BS 436, AGMA 218 and ISO 6336 were developed for the calculation of metallic gear wheels. The basic equations for calculating tooth bending and surface contact tension values are Lewis equation for tooth flank fracture and Hertzian contact equation for bending stress and contact corrosion. These equations are also used for plastic gears with modifications. In the Lewis and Hertz equations for metallic gears, various coefficients are used for stress intensity, load distribution and dynamic effect of load. There are some adaptations in the Lewis and Hertz equations for plastic gears for these cases, but they revealed less improvement. When the tangential force F_t acts on the tooth, it creates the maximum bending moment. The equation developed by Lewis for the tooth root bending stress by considering the tooth geometry is as follows [5]:

$$\sigma_e = \frac{F_t}{b \cdot m} \cdot K_d \quad (1)$$

The most important factor affecting the performance of plastic gears is temperature. Because the plastic gears are generally used without lubrication, the temperature generated during operation increases. Lubrication has a significant effect on increasing the heat transfer from the tooth and reducing the sliding friction, and it greatly affects the performance of the gear. The correction factors used for metallic gears do not have the same effect on the stress in plastic gears and additional factors, such as temperature and humidity, should be considered in plastic gears in addition to these factors. In a study performed using the spur gear combinations operating under lubricated and dry conditions and fiber reinforced nylon gears, it was experimentally determined that the wear was less in the gears operating with lubrication and their fatigue strength increased [11].
For the calculations of plastic gears, it is necessary to consider the modulus of elasticity and Poisson’s ratio, which vary with temperature, in the Hertz equation. BS 6168, Polypenco, ESDU 68001, VDI 2545 (Verein Deutscher Ingenieure) are the most commonly used methods and standards for plastic gear calculations. Moments of nylon 66 gears with 30 teeth that can be transmitted according to the module were found by the calculations made according to BS 6168, Polypenco and ESDU 68001 standards. In modified form, the temperature and other factors were also considered. It has been found that there are large variations between calculation methods [12].

Various standards have been developed for the calculation of plastic gears. It is difficult to provide a general equation covering all correction factors suggested for metallic gears as there are many factors that affect the fatigue behavior of plastic gears. Yelle proposed the equation as defined by the allowable tooth root bending stress (S_{La}) [13].
Experiments were carried out under 60°C lubrication conditions for two different types of nylon 6 gears with 40 teeth and a module of 2 and their calculations were performed according to VDI 2545 and then the results were compared. The fatigue life of the gears designed to have a smaller core diameter were experimentally determined to be higher and a similar result was obtained with the calculations made with the correction coefficient in the VDI 2545 [14].

2.1. Bending Stress Calculation based on the BS 6168 Standard

In the BS 6168 standard, the effects of temperature and lubrication are taken into account when making calculations of plastic gears according to the tooth root fracture. The converted contact ratio is calculated according to the change of humidity and temperature.
The fatigue stress is calculated as follows:

$$\sigma_F = \frac{F_t}{b.m_n} Y_F Y_\varepsilon K_A \tag{2}$$

The temperature to be generated on the surface and inside the bulk during operation of the gear is calculated using the equation given below:

$$\theta_1 = \theta_0 + \frac{136 P_t \mu (1+u)}{z_2 + 5} \left\{ \frac{1{,}71 \cdot 10^4 K_a}{b z_1 (v m_n)^{K_m}} + \frac{7{,}33 K_b}{A} \right\} + 5 \tag{3}$$

For bending stress calculation in the BS 6168 standard, there are Wöhler curves only for PA 66 and POM with a steel spur gear, a module of 3 and a base circular speed of 10 m/s in the unlubricated condition. The maximum operating temperature for PA 66 and PA 6 is given as 90°C in unlubricated condition. The bulk temperature is used in the calculation of tooth root fracture and the external surface temperature is used in the calculation of the contact stress. According to the temperature obtained with the equation (4), $\sigma_{F\lim}$ of the gear material is determined from the Wöhler graph and the safety factor is expected to be greater than 1.25 in the equation given below [15].

$$S = \frac{\sigma_{F\lim}}{\sigma_{Fh}} Y_x \tag{4}$$

The symbols used in equations were given in Table 1.

Table 1. Symbols used in equations

Symbols	Meaning	Units
σ_e	Bending stress	MPa
K_d	Correction factor	---
b	Tooth width	mm
m	Module	mm
F_t	Tangential tooth load	N
σ_F	Actual tooth root bending stress	MPa
m_n	Normal module	---
Y_F	Tooth form factor for bending stress	---
Y_ε	Contact ratio factor for bending stress	---
K_A	Application factor	---
θ_0	Initial temperature of gears	°C
θ_1	Temperature of pinion/whell (surface and bulk)	°C
z_1	Number of teeth of pinion	---
z_2	Number of teeth of wheel	---
μ	Friction coefficient	---
K_a	Experimental factor for material temperature	---
K_b	Correction coefficient for material temperature	---
K_M	Gear material constant	---
P_t	Transmitted power	kW
u	Conversion rate	---
v	Pitch line velocity	m/s
A	Surface area of gearbox	---
σ_{Flim}	Endurance limit for contact stress for gears	MPa
Y_x	Size factor for bending stress	---

3. THEORETICAL FINDINGS AND EXPERIMENTAL STUDIES

3. 1. BS 6168 Standard

According to the calculation method given in the BS 6168 standard, calculations for PA66 were carried out according to two different tangential forces, considering that the gearbox was open in various loads and that there was no change in humidity during operation and the results in Table 2 were obtained. The reason for the calculation for PA66 is that the Wöhler curve has been given for this material in the relevant standard.

Table 2. The number of cycles based on tangential loads obtained from the BS 6168 standard

m (mm)	Z	n (min^{-1})	d (mm)	F_t (kN)	σ_{Fh} (MPa)	σ_{Flim}(MPa)	The number of cycles
6	17	600	102	1500	15.32	22.53	10^6
				2000	20.53	30.19	50.000*

*In the Wöhler curve given in the standard, a value is available for minimum 10^5 cycles.
The fatigue life for 2000 N is an approximate value by extrapolating the graph.

3.2. Experimental Results

The gear specimen produced as module 6 from PA6 is shown in Figure 1. Gears produced by injection molding were tested in the Mechanical Laboratory of Faculty of Engineering in Pamukkale University by using Instron 8801 device at a constant frequency of 10 Hz and 25% position change. The test setup and load distribution are shown in Figure 2.

The life values determined in the experiments performed for tangential loads of 2000 kN and 1500 kN until reaching a 25% position change are given in Table 3. The tooth root stresses in Table 3 were calculated according to the bending equation.

Table 3. Experimental results for tangential loads

Tangential load (kN)	Tooth root stress (MPa)	The number of cycles	The number of cycles at 50% safety
		49.573	
		52.341	
2000	25.9	58.099	53.893
		53.167	
		56.284	
		1.201.567	
		1.151.552	
1500	19.43	1.223.348	1.184.899
		1.282.987	
		1.065.040	

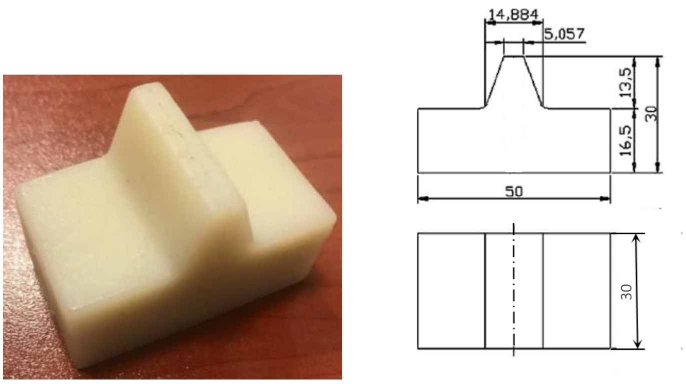

Figure 1. PA6 gear specimen and its dimensions

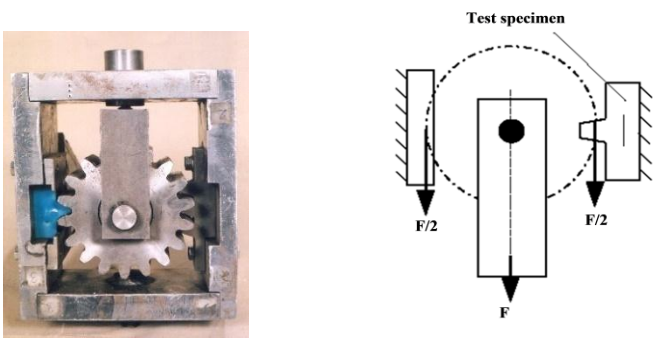

Figure 2. Test setup and schematic load distribution

3.3. ANSYS Analysis Results

The modeling was done in the dimensions of the specimen, and as in the case of test setup, the specimen was fixed from the lower surface and the ends of the upper surface and tangential load was lineally applied over the tooth. When a tangential force of 2000 kN was applied, the lifetime of the unreinforced specimen was minimum 58,147 cycles; when it was 1500 kN, the tooth root stress was found to be 17.6 MPa (Figure. 3 and 4).

The calculations made for tangential loads of 1500 kN and 2000 kN according to the BS6168 standard, experimental results and the results obtained from ANSYS finite element method are shown in Table 4.

Table 4. The results obtained for tangential loads of 1500 kN and 2000 kN

	Tangentiall oad (kN)	BS 6168	Experimental Results(25% position change)	ANSYS
Tooth root stress (MPa)	2000	20.53	25.9	23.1
The number of boundary cycles		50.000	53.893	58.147
Tooth root stress (MPa)	1500	15.32	19.43	17.6
The number of boundary cycles		1.000.000	1.184.899	1.343.871

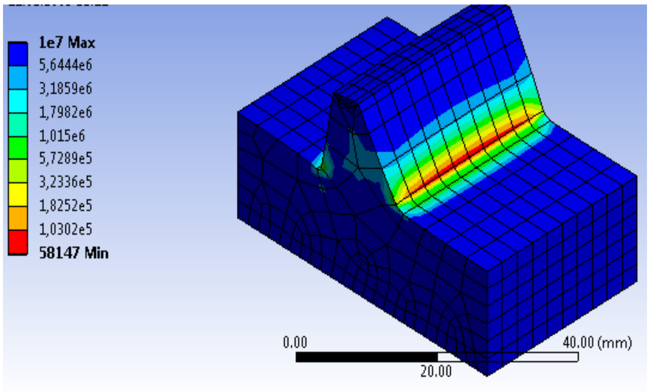

Figure 3. The life under a tangential load 2000 kN

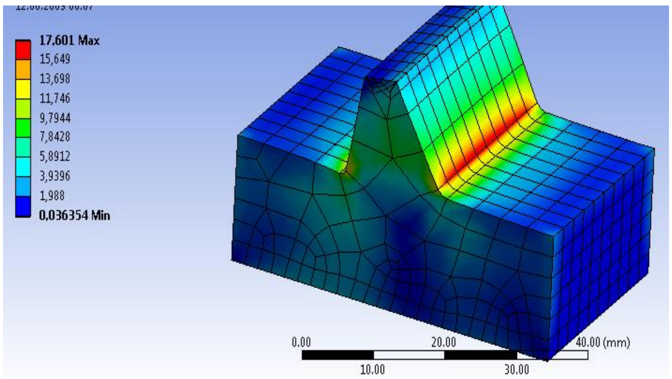

Figure 4. Tooth root stresses under a tangential load of 1500 kN

The material used in the experiments was PA6 whereas PA66 was used for theoretical calculations. Experimental studies have shown that under tangential force of 1500 kN, the life was 1.184.899 cycles while the minimum life of 2000 kN was found to be 53.893 cycles. For the tangential force of 1500 kN obtained with the BS 6168 standard, the number of cycles was 10^6 cycles and it was 50.000 cycles for a tangential load of 2000 kN and the results were consistent with experimental results. Higher number of boundary cycles were obtained for the simulations performed by the finite element method rather than those obtained by the BS 6168 standard and experimentally (Figure 5).

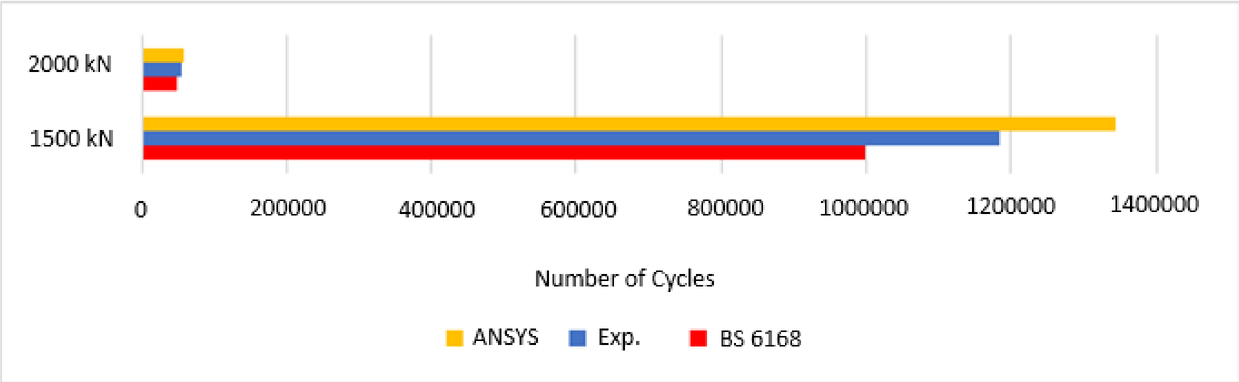

Figure 5. Comparison of analytical, BS6168 standard and ANSYS simulation results in terms of the number of cycles

In ANSYS, when the tangential load was 1500 kN the tooth root stress was found to be 17.6 MPa, whereas it was 23.1 MPa when the load was 2000 kN. Evaluating the tooth root stress in terms of theoretical calculation, the tangential load was 1500 kN in the specimen, the tooth root stress was 19.43 MPa while it was found to be 25.9 MPa for the tangential load of 2000 kN. Based on the BS 6168 standard, when the tangential load was 1500 kN, the tooth root stress was 15.32 MPa while it was determined as 20.53 MPa for the tangential load of 2000 kN. The highest values for tooth root stresses were obtained for bending stress (Figure 6).

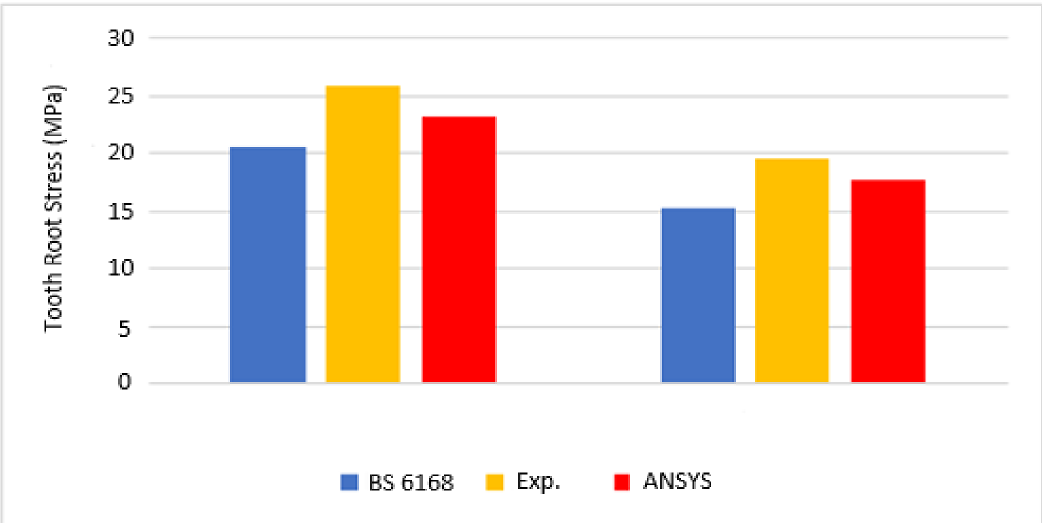

Figure 6. Comparison of analytical, BS6168 standard and ANSYS tooth root stress results

4. CONCLUSION

- The standards are lack of Wöhler curves for certain values, such as the number of teeth, module, spur gear combination, circumferential speed. The lack of these data is the most fundamental factor that limits the use of non-metallic gears.
- When the calculations and test results for Module 6 gears are evaluated, it was found that both tangential loads and life values are close to each other.
- The tooth root stresses given theoretically were calculated with general bending stress formula.
- It was found that the theoretical calculations made according to the BS 6168 standard were compatible with the lifetime values of experimentally tested module 6 gears. Examining the results according to the tangential loads, it was determined that the values found in the theoretical calculations were lower than the life values obtained in practice. In this respect, the calculations made with BS 6168 are found to be reliable.

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