UDC 62-233.3/.9

RELIABILITY ANALYSIS OF ORIENTING TOOL GEARBOXES

O.E. Polyntsev
Schlumberger, Stonehouse, United Kingdom
e-mail: OPolyntsev@slb.com

Abstract. *Reliability analysis of epicyclical gearboxes has been developed using an analytical fatigue damage summation model and statistical field test data. Some specific results and conclusions have been presented. The approach shown in this work could be applied to reliability estimations during early design stages of drilling tools.*

Keywords: *reliability, fatigue, gearbox, drilling tools*

1. Introduction

Reliability analysis is an important step of a design process of a directional drilling tool. However, it is a common situation that engineers do not have all the required information to do trustworthy calculations beforehand. In such a situation, one must use either qualitative experience-based data or statistical data accumulated during drilling jobs that used tools similar to the one being designed. Oil-field service companies have employed a few different approaches to estimating reliability of the mechanical components during the concept stages of the project development. One of the most widely used techniques is failure mode and effect analysis (FMEA). FMEA can indicate potential areas of concern, but cannot give accurate comparisons of different design solutions.

This paper considers a methodology of the reliability analysis that has been used for gearboxes of a drilling tool designed by Schlumberger. The approach is based on consideration of fatigue of the most critical components of the epicyclical gearboxes.

2. Methodology

In this work, the reliability analysis methodology is illustrated by example of epicyclical gearboxes in a drilling tool. The methodology is based on fatigue calculations of the gear teeth along with evaluations of the probability of failure.

Fatigue calculations use torque spectrum estimations from the field test data of a tool similar to the one being designed. Fig. 1 presents an example of a load spectrum, which is assumed to be scalable for any arbitrary number of hours. The spectrum is separated on the build and lateral sections. The difference between the two is that when the tool is in the build regime, there is no rotation of the gearbox.

The *Palmgren-Miner's* rule for linear summation of damage is used, applying correction for the special features of the load spectrums, see [2]. According to the rule damage takes place when

$$\sum_{i=1}^{M} \frac{n_i}{N_i} = A,$$

where n_i – is the number of cycles with load *i*; N_i – the number of cycles before failure with load *i* (taken from the S-N-curve); and *M*, the number of load blocks. Note that n_i/N_i is also referred to as fatigue damage.

Stress calculations on the gear teeth could be done using both standardized techniques and finite-element modeling, allowing for the relevant correction factors for materials, loads and working conditions [1 - 4].

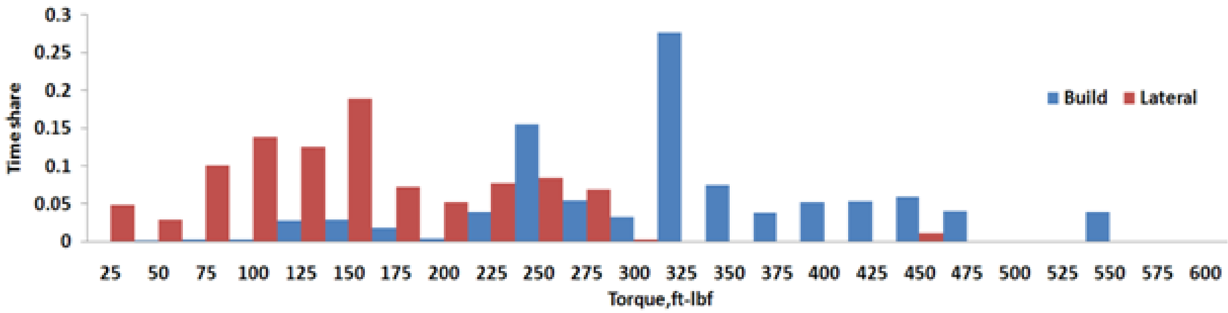

Fig. 1. Example of torque spectrums

The value of *a* is defined via the following relationship

$$A = \frac{\frac{\sigma_{amax}}{\sigma_f} \varsigma - K}{\frac{\sigma_{amax}}{\sigma_f} - K}.$$

Here σ_{amax} is the maximum amplitude of cycling load (or equivalent amplitude for the cycles with non-zero mean stresses); σ_f – the allowable stresses (fatigue strength); and K – the coefficient allowing for relative level of load amplitudes causing damage (according to experiments $K = 0.5\ldots0.7$). Factor ς is calculated the following way

$$\varsigma = \sum_{i=1}^{M} \frac{\sigma_{ai}}{\sigma_{amax}} t_i,$$

where σ_{ai} is the amplitude of cycling load (or equivalent amplitude for the cycles with non-zero mean stresses) under load *i*, and

$$t_i = \frac{\nu_i}{\sum_{i=1}^{M} \nu_i},$$

where ν_i is the number of load *i* repetitions within the load spectrum.

If $A < 0.2$ then it is recommended to accept $A = 0.2$. According to [2] with such a correction of the cumulative damage rule the possible error in the life estimations does not exceed 2 to 2.5 with probability 95 %, whereas without the correction ($A = 1$) the life could be five to seven times over- or underestimated.

The number of cycles to fail with a particular load spectrum is given by

$$N_f = \frac{A}{\sum_{i=1}^{M} \frac{n_i}{N_i}}.$$

The density of the normal (Gaussian) probability distribution, which is commonly used for probabilistic problems of fatigue, is described as [4]:

$$\varphi = \frac{1}{\sqrt{2\pi} \cdot S(\log[N])} \cdot e^{-\left[\frac{(\log[N_\Sigma] - \log[N_f])^2}{2S(\log[N])^2}\right]},$$

where $S(\log[N])$ is the standard deviation of the logarithm of the number of cycles before failure (for example, for the helicopter transmission parts $S(\log[N]) = 0.2$, [5]), and $N_\Sigma = \sum_{i=1}^{M} n_i$.

The probability of failure is given by

$$q = \int_{-\infty}^{\log[N_\Sigma]} \varphi \, d(\log[N]).$$

A. Lateral Case

The gearbox is rotating when drilling laterals, and the planetary gear teeth experience reverse cycling load (stress ratio R = –1). Straight planetary gear teeth see one-plane bending only.

The number of cycles for the planetary gear teeth is calculated using the following relationship

$$n_{pi} = \frac{RPM}{60} \cdot U \cdot \frac{2}{1 - \frac{D_S}{D_R}} \cdot \tau_i \cdot T \cdot K_r,$$

where RPM is the number of revolutions of the drilling tool per minute; U – the gear ratio of the gearbox; D_S – the diameter of the sun gear; D_R – the diameter of the ring gear; K_r – the number of resonant spikes during teeth interaction; τ_i – the share of the load i, taken from the load spectrum; and T – the input time in seconds.

The number of cycles for the ring gear is

$$n_{ri} = \frac{RPM}{60} \cdot U \cdot z_p \cdot K_r \cdot \tau_i \cdot T,$$

where z_p – is the number of planets.

An overloading factor of 1.75 is taken as recommended in [3] and also defined in ISO 6336:1996. In this case the source of power is uniform, and the driven machinery subject to heavy shocks. There is no reliable data to prove that this factor is too conservative. Torque oscillations from the field test data seem to be in range of ±30 % (i.e. overload factor 1.3), but this cannot allow for torsional and bending resonances in the gear train, which seem to be unavoidable in such a broad range of RPM in the entire transmission system.

The probability of failure of the planetary gear is given by

$$P_p = 1 - q_p^{z_p},$$

where q_p – is the probability of failure of one tooth; and Z_p – the number of teeth.

Similarly, for the ring gear

$$P_r = 1 - q_r^{z_r},$$

Fig. 2 exemplifies the damage to the planetary and the ring gear teeth by particular torque regimes. As seen in this example, the highest damage is being done by the 450 ft-lbf regime (0.0466, or 15.5 % of $A = 0,2999$), although it has relatively rare occurrence (1.1 % of overall time).

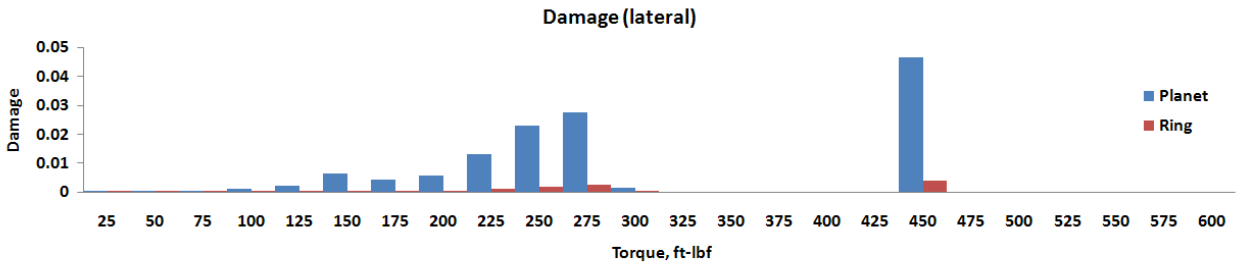

Fig. 2. Fatigue damage distribution, lateral case

<u>B. Build Case</u>

In this case the gearbox is not rotating, only rarely changing its angular position. The teeth are loaded by cycling force, which acts in one direction (i.e. has mean value and relatively small alternating value).

Amplitude of the torque is supposed to have value of 30 % of the torque, which more or less represents the field test observations (stress ratio R = 0.7/1.3 = 0.54).

The number of cycles is calculated as follows

$$n_{ri} = \frac{RPM}{60} \cdot U \cdot z_p \cdot K_r \cdot \tau_i \cdot T,$$

where RPM_m is the RPM of the mud motor; K_h – the number of the major load harmonics (normally it equals the number of rotor lobes of the positive displacement motor,); and N_{TF} – the number of tool face changes.

The probabilities of failure for a planetary gear and ring gear are

$$P_p = 1 - q_p^{z_{pb}},$$

$$P_r = 1 - q_r^{z_{rb}},$$

where $Z_{pb} = 2 N_{TF}$ is the number of teeth of the planetary gear, holding the load throughout the drill; and $Z_{rb} = z_p N_{TF}$ – the number of teeth of the ring gear, holding the load throughout the drill.

Fig. 3 exemplifies damage to the planetary and ring gear teeth by particular torque regimes for the case of build.

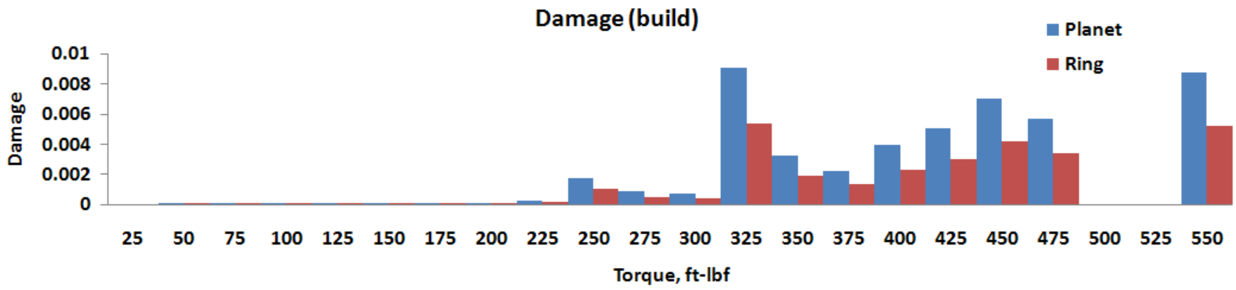

Fig. 3. Fatigue damage distribution, build case

3. Results

Using the methodology described above a reliability analysis of the gearbox has been performed considering three typical load spectrums shown in Fig. 4 - 6.

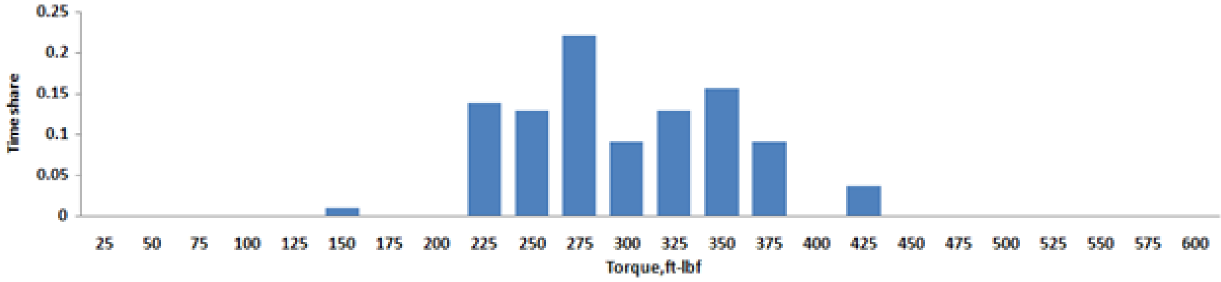

Fig. 4. Load spectrum 1 (build section)

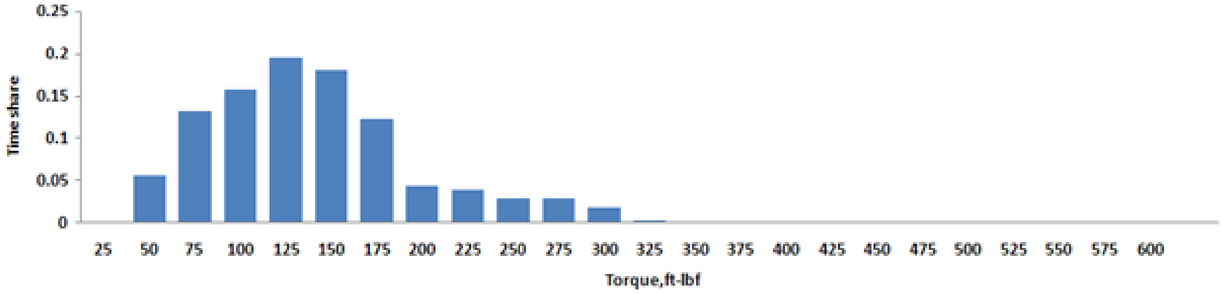

Fig. 5. Load spectrum 2 (lateral section)

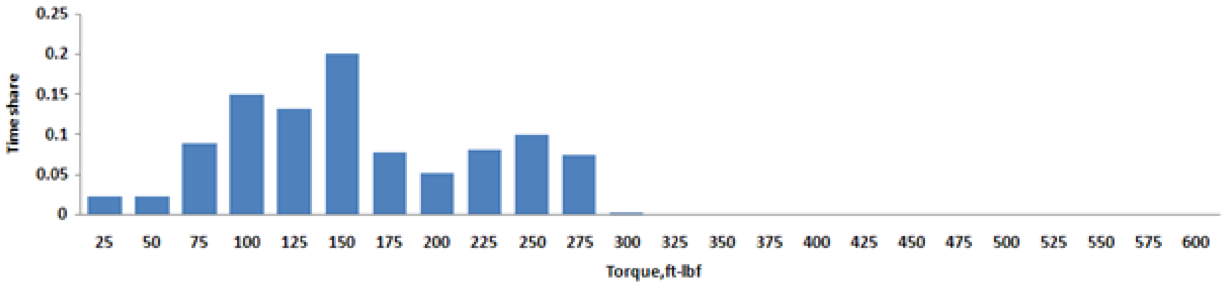

Fig. 6. Load spectrum 3 (lateral section)

Fig. 7 - 9 present the fatigue damage distributions for the three considered spectrums. Fig. 10 shows probabilities of success for the three cases based on the critical part – planetary gear. For 90 % success probability the following number of hours have been defined: Spectrum 1 – 518 hours; spectrum 2 – 214 hours; spectrum 3 – 125 hours.

These estimations have helped to establish the safe drilling envelope and also to identify possible ways of increasing the reliability of the gearbox during the jobs, such as applying fish-tailing techniques rather than continuously drilling the lateral sections.

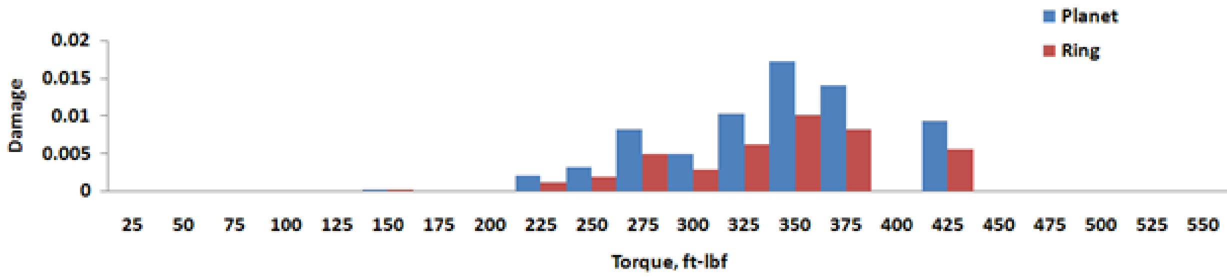

Fig. 7. Fatigue damage, load spectrum 1

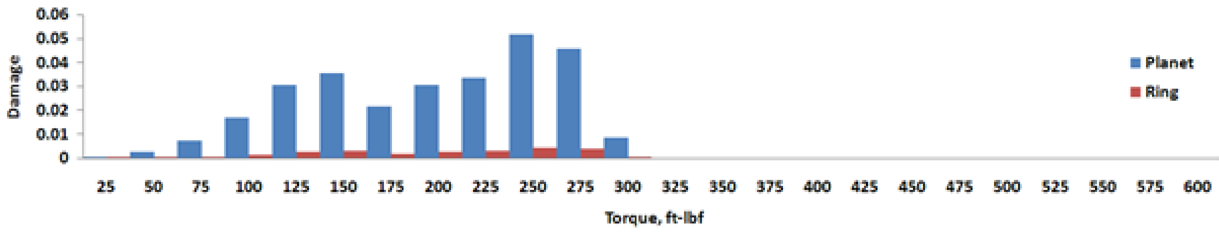

Fig. 8. Fatigue damage, load spectrum 2

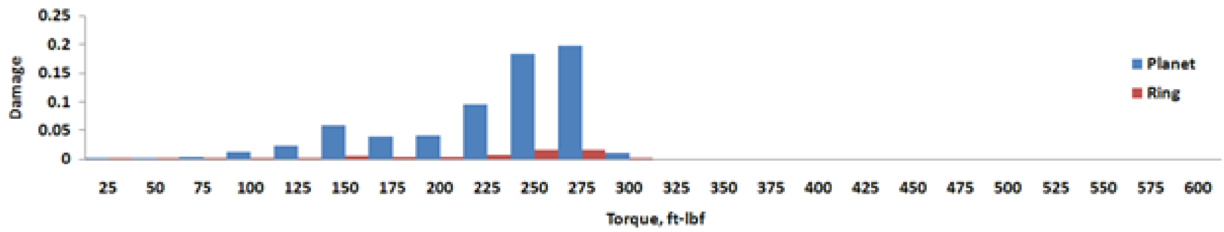

Fig. 9. Fatigue damage, load spectrum 3

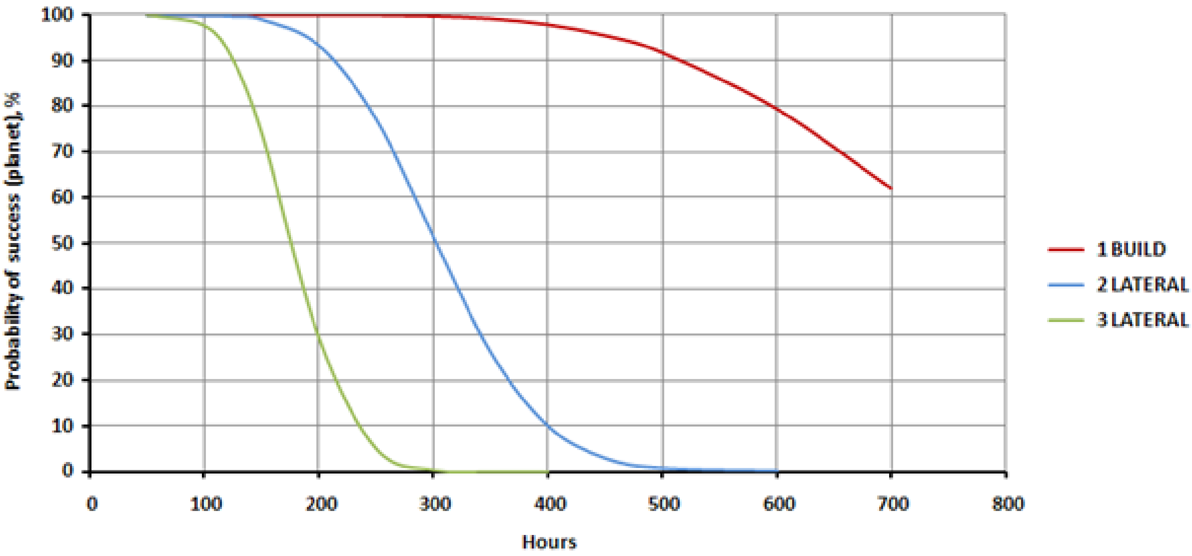

Fig. 10. Probabilities of success vs. drilling hours

4. Conclusions

This article presents reliability analysis methodology applied to the design of epicyclical gearboxes along with some particular results. The approach described in this paper is used for analytical predictions of reliability of the tools during the design stages. It is based on the fatigue damage summation techniques and statistical data from the field tests. Such a methodology gives more solid foundation for the design solutions taken at the conceptual stages of the design process.

References

1. ANSI/AGMA 2001-D04. American National Standard. Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

2. Serensen S.V., Kogaev V.P., Shneiderovich R.M. Nesushchaya sposobnost i raschet detalei mashin na prochnost (Load-bearing capacity and stress analysis of machine parts). Moscow, Mashinostroenie, 1975. 488 p.

3. Dudley D.W. (ed). Gear Handbook. McGraw-Hill, New-York, 1962.

4. Birger I.A., Shorr B.F., Iosilevich G.B. Raschet na prochnost detalei mashin (Strength calculations of the machine elements). Moscow: Mashinostroenie, 1979. 640 p.

5. Mil M.L. et al. Vertolety. Raschet i proektirovanie. Tom 2. Kolebaniya i dinamicheskaya prochnost' (Helicopters calculation and design. Vol. I. Vibrations and dynamic stability). Moscow: Mashinostroenie, 1967. 424 p.