

Methods of Statistical Dynamics for the Calculation of Gear Stress Distribution and its Effect on Gear Failure Probability

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American Gear Manufacturers Association



TECHNICAL PAPER

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ABSTRACT:

The process of most machine loading has a random character which is determined by external variation and dynamic qualities of the system. Such an approach was used to obtain the load spectrum for gears and the probability of its failure. Analysis and experiments showed that gain factor for gear is distinguished by the similar parameter of the entire drive system. This is explained by the uniform distribution of gear stress even for cases of static loading. Method of gear strength calculation with the statistical parameter is discussed.

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American Gear Manufacturers Association
1500 King Street, Suite 201
Alexandria, Virginia, 22314

October, 1991

ISBN: 1-55589-613-8

METHODS OF STATISTICAL DYNAMICS FOR THE CALCULATION OF GEAR STRESS DISTRIBUTION AND ITS EFFECT ON GEAR FAILURE PROBABILITY

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I. INTRODUCTION

In [1] Ing. Theodor Hosel compared the accepted standards of AGMA, ISO, DIN, and Comecon. One of the conclusions of this research is that "Different interpretations of geometrical data and different calculating methods within the standards can lead to considerable differences of surface stress, root stress, and transmissible torque, calculation for one and the same gear pair" [1; page 13]. This shows that such differences are determined because of different values of some factors that were applied for stress analysis.

For example, the author calculated the Y factor for the same gear (AGMA-Y=0.917, DIN-Y=0.799) and explained that a 15% difference in this factor can be understood if we take into consideration the different AGMA's and DIN's definitions of the critical points and critical sections at the tooth root. Another example of his research is when the use of the same helical gear ($\psi = 10^\circ$) yielded a difference in increasing the load capacity benefit. The author showed that, the torque ratio between the helical gear and the spur gear is :

$$T_{10}/T_0 = 1.87 \text{ (AGMA)}; T_{10}/T_0 = 1.45 \text{ (DIN)};$$

$$T_{10}/T_0 = 1.3 \text{ (Comecon)};$$

But, it can be noted that, as a rule, despite differences in the value of various factors of strength calculations, the results can be smoothed out due to the application of a "safety factor".

Despite the fact that AGMA's fundamental formulae do not contain a denominated Safety Factor, the sense of AGMA's formulae is the same and the standard explains that, "The application factors make allowance for any externally applied loads in excess of the nominal tangential load W" [2; page 14]. It is clear that the practice of applying the safety factor reflects our knowledge based on the experience of studying similar

machines and using the same materials. On the other hand, its application reduces the importance of determination of the real picture of the stress distribution for the gear tooth as it allows one to use only nominal load.

C.C.Wang, offering the economic approach for the use of an application factor, noted that the "Empirical data is a good reasonable approximation, but, it is an over-simplified shortcut for dynamic systems" [3; page 1].

In fact, there is a real scheme of stress distribution and the possibility of simulating this scheme is dependent on our ability to reflect the real physical processes by analytical methods. Among the various factors which are taken into consideration for determination of maximum stress, the value of the load distribution factor is one of the most important.

There are many conditions which determine nonuniform load intensity, and each of them has an influence on the length of contact line between pair of gears.

The real contact length of a pair of teeth compared with the whole of the face width depends on two main factors which are reflected in every one of the standards.

- position of gears relative to the bearing supports.

- ratio of the whole of the face width of the gear to the diameter of the pinion.

Besides, there are researches for evaluation of the influence of another special characteristic on the length of contact line and consequently, on stress distribution. In [4], T. Tobe and K. Inoue analyzed the influence of element of manufacturing errors to estimate the maximum load intensity. In [5], W. Mark described the transmission error by three scalar components and showed that even a factor such as bearing misalignments may be included in the calculation.

Because of this, some of the authors gave integral coefficients (K) which are dependent on



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