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ANALYSIS OF THE BENDING STRESS OF GEAR TOOTH

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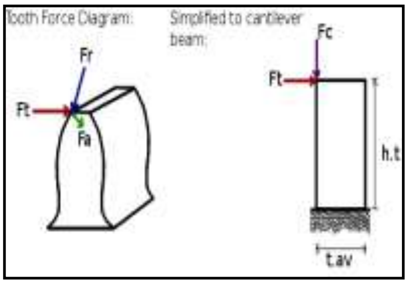
ABSTRACT
Gears are one of the most essential components in mechanical power transmission systems. Bending stress plays a significant role in gear design and one of the main contributors for the failure of the gear sets wherein its magnitude is controlled by the nominal bending stress and the stress concentration due to the geometrical shape. The bending stress is indirectly related to shape changes made to the cutting tool. This is work analyse the bending stress during operation condition. Bending stress evaluation in modern gear design is generally based on the more than one hundred year old Lewis equation.

Key words: Bending stress, Lewis equation, Gear sets.

INTRODUCTION
Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined of gear technology. A gear as usually used in the transmission system is also called a speed reducer, gear head, gear reducer etc., which consists of a set of gears, shafts and bearings that are factory mounted in an enclosed lubricated housing. Different kinds of metallic gears are currently being manufactured for various industrial purposes. Seventy-four percent of them are spur gears, fifteen percent helical, five percent worm, four percent bevel, and the others are either epicyclical or internal gears.

Researchers in the gear field have proposed many solutions to tackle the problem of failure. Lewis suggested the idea of considering the tooth as a cantilever beam and some researchers still used this approach to analyze the bending stress used FEM to analyze the stress of gear.



LEWIS BENDING STRESS

$$\sigma = \frac{MC}{I},$$

we get the maximum bending stress

$$\sigma_t = \frac{W_t P_d}{FY}$$

Where:

W_t is the tangential laod

P_d is the diametral pitch

F is the face width and

Y is the Lewis form factor

The form factor Y is a function of teeth, pressure angle, and involute depth of the gear. It accounts for the geometry of the tooth, but does not include stress concentration.

ALLOWABLE BENDING STRESS

Arriving at a safe allowable stress level for various gear materials is not straight-forward with the Lewis method – but then it is only a simplified approximation. Unless you are given a specific material allowable value or a table of values, it is reasonable to estimate an allowable strength as Sut / 3 , one third of the material's ultimate tensile strength. Be aware that the teeth of gears functioning as idlers experience reversed bending because they are loaded in one direction by the driver and in the opposite direction by the driven gear.

AGMA BENDING STRESS

The AGMA* spur gear bending method can be viewed as a
detailed refinement of the Lewis method.

$$\sigma = \frac{W_t P_d}{F Y_j} \frac{K_a K_s K_m K_B}{K_V}$$

Yj is the Lewis form factor corrected for several geometry factors,
including stress concentration effects.

Ka is the Application factor (1 to 2.75) that accounts for pulsation and
shock in the driver and load.

Ks is the Size factor (1 to 1.4) which penalizes very large or wide teeth.

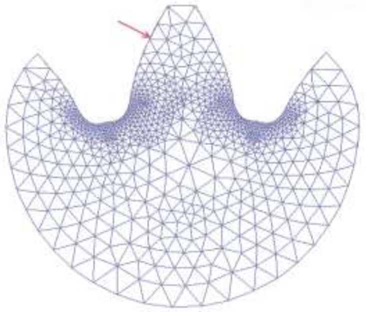
Km is the Load Distribution factor (1 to 2) that is a function of face width.

KB is the Rim Thickness factor which penalizes for the rim flexibility of non-solid gears.

KV is the Dynamic factor (0.5 to 0.98), essentially a tailored Barth velocity factor that considers gear quality.

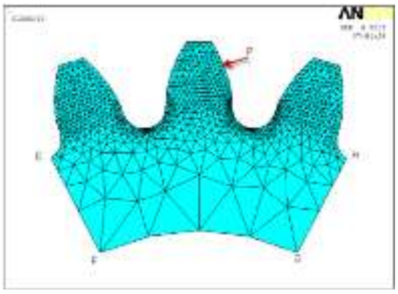
GEAR TOOTH MODELING

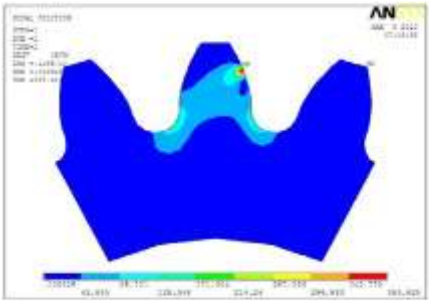
The direct gear design method defines parameters of the gear mesh to provide complete geometry of the involute profile of the teeth, including the base diameter, form diameter, out-side diameter, tooth thickness, tip radii etc. the fillet profile initially is defined as a trace of the tip of the mating gear tooth. The 2-d FEA model in figure shows a gear tooth profile that is limited from the sides and bottom by a constrained border with stationary nodes. All other nodes on the tooth profile and inside the tooth contour are movable. The fillet portion of the tooth profile has equally spaced nodes with higher density (number of nodes per unit of profile length) than the rest of the tooth profile. The nodes on the involute profiles and the top land are located to have higher density close to the fillets and lower density in the top part of the tooth. The number of tooth profile nodes and the node density coefficient (ratio of the fillet profile node density to an average node density of the involute and top land profiles of the tooth) are selected. Fewer tooth profile nodes and lower node density coefficients yield less accurate stress calculations. Selection of larger numbers of tooth profile nodes and high node density coefficients provides a more accurate result, but increases calculation time. In most cases, 80–100 tooth profile nodes and node density coefficients of 1.75–2.5 were used.



The tooth load distribution problem is considered to define a value, a set of application point coordinates, and the direction of the force resulting in maximum bending stress. The friction effect at the contact point has been ignored. The load application point typically does not exactly match with a tooth profile node. It is replaced by a pair of forces that are applied to the two closest nodes above and below the load application point. The combined load value of those forces equals an initial load and distributed reversal proportional to the distances between the nodes and the load application point.

APDL (ANSYS Parameter Design Language) is a kind of parametric design language, used for parameterized finite element analysis, analysis of the batch, the secondary development of a dedicated analysis system, as well as the optimal design. APDL language can create complex models to avoid the undesirable factors of transmission between different software models. In this paper, the APDL language is used to establish a finite element model of the tooth and optimize the design. When the cutter tip fillet radius R =0.38*m* (After calculation, the quadratic Bezier curve represents Arc with R =0.38*m* when w1=0.8192). A series of key points were established to generate standard gear tooth profile by B-Spline curve fitted. In order to accelerate the optimization speed, PLANE 82 is used to establish two-dimensional finite element model. Normal lord P = 254N/mm is applied to the highest point of single tooth contact. The line segments EF, FG, GH are full-constraints and free meshing is adopted .the results are shown in Figure.





CONCLUSION

This paper is review of the fillet radius optimization and analyze the bending stress strength. Direct Gear Design uses FEA for bending stress evaluation because the Lewis equation and its related coefficients do not provide a reliable solution to the wide variety of non-standard gear tooth profiles

that could be considered. Bending stress balance allows equalizing the tooth strength and durability for the pinion and the gear. Optimization of the fillet profile allows reducing the maximum bending stress in the gear tooth root area by 10–30%. It works equally well for both symmetric and asymmetric gear tooth profiles. The bending stress reduction leads to:

• Size and weight reduction

• Longer life

• Higher load application

• Cost reduction (less expensive materials, heat treatment, etc.)

• Noise and vibration reduction, increased efficiency

(finer pitch, more teeth will result in higher contact ratio for the given center distance). The paper also describes an approach to the tooth parameters' tolerancing and tool profile definition.

REFERENCES

1.Lewis W. Investigation of the Strength of Gear Teeth. Proceedings of the Engineers Club, Philadelphia, 1893.

2.Wilcox L, Coleman W. Application of Finite Element to the Analysis of Gear Tooth Stresses,Journal of Engineering for Industry,1973,95(4): 1139 -1148.

3. AL Kapelevich, YV Shekhtman. Tooth fillet profile optimization for gears with symmetric and asymmetric teeth [J] . Gear technology ,2009 (9):73-79.

4. V. Senthil Kumar, D.V. Muni, G. Muthuveerappan. Optimization of asymmetric spur gear drives to improve the bending load capacity [J] . Mechanism and Machine Theory 2008, 43(07): 829–858.

5.ANSI/AGMA 1006-A97—Tooth Proportions for Plastic Gears, Appendix F: "Generating Gear Geometry without Racks," AGMA, Alexandria, VA, 1997.

6.Pulley, F.T., M.W. Kipling, G.A. Matson, D.L. Thurman and B.W. Avery. "Method for producing and controlling a fillet on a gear," US Patent #6164880, 2000.

7.Kapelevich, A.L. "Geometry and design of involute spur gears with asymmetric teeth," Mechanism and Machine
Theory, 2000, Issue 1 35, pp. 117–130.