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Design, Modeling & Analysis of Helical Gear According to Bending Strength Using AGMA and FEMAP

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Abstract: ***Gears are used to transmit the power. They can change torque, speed and direction of power source. While transmitting the power they are subjected to two main stresses i.e. bending stress and tooth contact stresses. These two stresses results in failure of the gear teeth, contact stress results in pitting failure at the contact surface and root bending stress results in fatigue fracture. In this paper bending stress is calculated by using analytical method for which AGMA standard (American Gear Manufacturing Association) is used and the model is designed in AUTODESK INVENTOR 2016 and saved in IGES format and then imported in the FEMAP with NASTRAN software where it can be analyzed. The main objective of this study to calculate bending stress and then compare results of both analytical and FEA approach.***

Keywords: Helical gears, bending stress, AGMA standard, Autodesk Inventor, FEMAP with Nastran

1. Introduction

In today's industries power transmission with a minimum power loss plays a vital role. Power can be transmitted through various methods like belt and rope, chain drive, gears, clutches etc. Gears are used for transmitting the power and motion when the distance between the driving and driven shaft is relatively small and when a constant velocity ratio is desired. The crucial requirement of effective power transmission in various machines, automobiles, elevators, generators, etc. has created an increasing demand for more accurate analysis of the characteristics of gear systems, for instance in automobile industry highly reliable and lightweight gears are essential. Furthermore the best way to diminution of noise in engine requires the fabrication of silence gear system.

Helical gears are now being used more and more for power transmission owing to their larger load carrying capacity, higher operating speeds, relatively smooth and silent operation. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods that can easily be put into practice and also give useful information on contact and bending stresses.

2. Literature Review

[1] There are mainly two kinds of stresses in gear teeth, tooth contact stresses and bending stresses. While designing the gears these both stresses are to be considered. Based on the calculation of bending stress one of the principal failure modes is studied in this paper. Helical gears are used in industry where noiseless operation and the power transmission is required at heavy loads with smoother operation. In this paper a 3D model helical gear was is prepared in Pro engineer and stress analysis part is carried out in ANSYS 11.0.

[2] In this paper finite element model for monitoring the stresses induced in tooth flank, tooth fillet during meshing of gears is prepared. The Involute profile of a helical gear has been modeled and the simulation is carried out for the bending and contact stresses. To estimate contact and bending stresses, 3D models for different face width, helix angle are generated by modeling software and simulation is done by finite element software. AGMA bending equation and AGMA contact stress equations are used for analytical method. The helix angle and face width are important geometrical parameters during the design of gear. Maximum bending stress decreases with increasing face width. It will be higher on gear of lower face width with higher helix angle.

3. Objective

The salient objectives of the present study have been identified as follows:
a) Design of gear to transmit desired power at required rpm using AGMA procedure.
b) Gear Modeling using Autodesk Inventor.
c) Gear analysis using FEA.
d) AGMA and FEA result comparison.

4. Gear Design

4.1 AGMA stress equation

Two fundamental stress equations are used in the AGMA methodology, one for bending stress and another for pitting resistance (contact stress). In AGMA terminology, these are called stress numbers.

i. Bending strength

σ_F =Ft *K_O *K_V * K_S * K_H *K_B / b / m_t / Y_J

where

Ft is transmitted tangential load (N) = P/V

V is pitch line velocity (m/s) = π.dp.Np /60

dp is pitch diameter of pinion (m) = Np/Pt

Pt is transverse diametrical pitch (teeth/mm) = Pn. cos ψ

Pn is normal diametrical pitch (teeth/mm) = 1/m

The various factors which are used in AGMA bending strength equation are as below.

ii. Overload factor, Ko

The overload factor Ko is intended to make allowance for all externally applied loads in excess of the nominal tangential load Ft for a particular application. Overload factors can only be established only after considerable field experience is gained in a particular application.

iii. Dynamic factor, Kv

The dynamic factor Kv makes allowance for the effect of gear-tooth quality related to speed and load, and the increase in stress that follows. AGMA uses a transmission accuracy number Qv to describe the precision with which tooth profiles are spaced along the pitch circle.

iv. Load distribution factor, K_H

The load-distribution factor modified the stress equations to reflect no uniform distribution of load along the line of contact. The idea is to locate the gear "midspan" between two bearings at the zero slope place when the load is applied.

v. Rim thickness factor, K_B

When the rim thickness is not sufficient to provide full support for the tooth root, the location of bending fatigue failure may be through the gear rim rather than at the tooth fillet. In such cases, the use of a stress modifying factor K_B is recommended. This factor, the rim-thickness factor K_B, adjusts the estimated bending stress for the thin-rimmed gear.

4.2 Helical gear parameters

Table 1: Technical data for helical gear

Sr. No.	*Description*	*Value*	*Units*
1	Operating center distance, a	330	mm
2	Number of teeth-pinion, Np	64	No's
3	Number of teeth-gear, Ng	64	No's
4	Normal metric module, mn	5	
5	Nominal pressure angle, φn	20	degree
6	Helix angle, β	13	degree
7	Tooth width, b	85	mm
8	Ultimate tensile strength	800	MPa
9	Yield strength	550	MPa
10	Modulus of elasticity E1, E2	210000	MPa
11	Poisson's ratio, v1, v2	0.3	
12	Hardness in tooth side	600	BHN
13	Quality number, Qv	7	
14	Installed power, P	18.5	kW
15	Speed, N	750	rpm

4.3 Helical gear calculations

Using above parameters and AGMA equation, tooth bending stresses are calculated

Pn is normal diametrical pitch (teeth/mm)= 1/5
= 0.2

Pt is transverse pitch (teeth/mm), = Pn. cos β
= 0.193

Pitch dia of pinion, dp (mm) = Np / Pt
= 331.29

Pitch line velocity, V (m/s) = π.dp.Np /60
= π*331*29.64/60
= 13.01

Transmitted load, Ft (N) = P / V
= 185000 / 13.01
= 1422

K_O is overload factor = 1.1

K_V is dynamic factor = 1.65

K_S is size factor = 1

K_H is load distribution factor = 1.25

$_B$ is rim thickness factor = 1

Y_J is geometry factor for bending strength = 0.66

m_t is transverse metric module (mm) = 5.32

σ_F is bending stress number (N/mm^2) = 10.85

5. FEA modeling

Modeling is the art of representing the object, system or phenomenon. A general-purpose commercial finite element software, MSC Nastran is applied to conduct the static simulations and analysis. The FEA model of gear in this study is constructed based on the geometry. A 3-D solid model is constructed for the static test simulation in Autodesk Inventor.

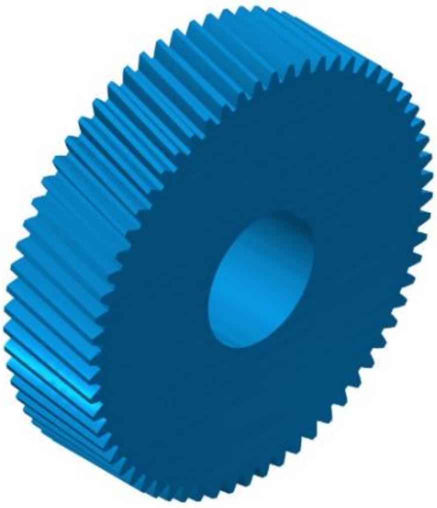

Figure 1: 3D model of helical gear

6. Meshing and FEA analysis

The finite element method is a well-known tool for the solution of complicated structural engineering problems, as it is capable of accommodating many complexities in the solution. In this method, the actual continuum is replaced by an equivalent idealized structure composed of discrete elements, referred to as finite elements, connected together at a number of nodes. The finite element method was first applied to problems of plane stress, using triangular and rectangular element. The method has since been extended and we can now use triangular and rectangular elements in plate bending, tetrahedron and hexahedron in three-dimensional stress analysis, and curved elements in singly or doubly curved shell problems. Thus the finite element method may be seen to be very general in application and it is sometimes the only valid form of analysis for difficult problems.

The 3D cad model in IGES file format is imported in FEMAP for the preparation of FE model. Cleanup and defeature to modify the geometry data and prepare it for meshing operation. This process involves deletion of curvature of very small radius. Mixed type of elements which contains quadrilateral as well as triangular elements, have been used in analysis. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z-axes. The sensitive regions have been re-meshed by manually considering the shape and size of the parts. Quality check of all the elements has been performed and mesh is accordingly optimized.

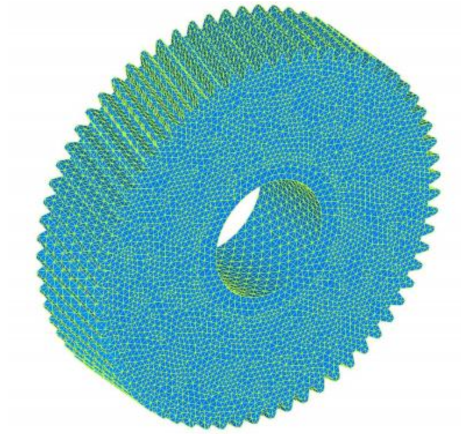

Figure 2: Meshed model of helical gear

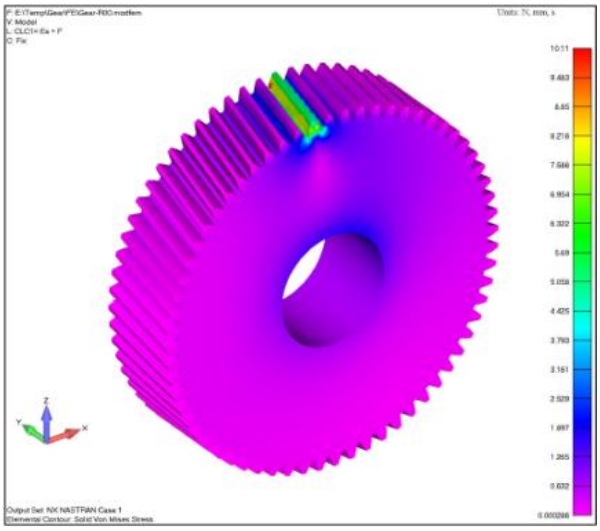

Figure 3: Stress plot for helical gear

7. Results and Conclusion

The obtained bending stress as per the AGMA procedure is 10.85 MPa.

The obtained bending stress as per FEA analysis is 10.11 MPa.

The results obtained from FEMAP when compared with the AGMA procedure, it shows that there is a little variation with a difference in percentage of 7.30%. From the results we can conclude that FEMAP can also be used for predicting the values of bending stress at any required face width which is much easier to use to solve complex design problems like gears.

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