DESIGN OF SPUR GEAR AND ITS TOOTH PROFILE FOR GIVEN POWER

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Abstract

Spur Gears are the most widely recognized means of transmitting power in the current engineering applications. They changes from tiny size utilizes as a part of watches to the bulky gears utilized in marine speed reducers; bridge lifting components and railroad turn table drivers. They are important elements of main and auxiliary mechanism in numerous machines, for example, metal cutting machine tools, marine engines, transmitting machinery, automobiles, tractors, rolling mills, hoisting etc. MATLAB has been used to design gear in the present work. MATLAB has been widely utilizedto solve scientific and research problems due to its accuracyand numerous built in functions which makes it flexible. In present studyspur gears has been designed. A MATLAB code has been written when run, ask for the inputs and executes the essential design calculations and providesrequired output values. MATLAB code also gives the tooth profile of involute gear with correct dimensions.

Keywords: Spur gear design, Bending and Compressive stress, Tooth profile, MATLAB

Introduction

Gears are one of the oldest of humankind's creations. About all the devices we consider as a machines use gear of some sort. Gear innovation has been produced and extended as the centuries progressed. Much of the time, gear design is considered as a speciality. Nevertheless, the configuration or determination of agear is just a part of the general system design. From industry's angle, gear transmission systems are viewed as one of the critical aspects of vibration examination. The knowledge of the behaviour when gears are in contact is essential if one needs to perform system checking and control of the gear transmission system. Albeit there are vast measures of research studies about different themes of gear transmission, the fundamental knowledge of gear in mesh still needs to be affirmed.

The four noteworthy failure modes in gears are contact fatigue, tooth bending fatigue, surface wear and scoring. Under repeated loading due to fatigue gears can be damaged by two types; pitting of gear teeth flanks in particular setting of apparatus and tooth damage in tooth root. Tooth breakage is unmistakably the most noticeably worst harmful case. Gears could have

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genuinely hampered working condition or even be annihilated. Due to this reason stress in the tooth ought to dependably be precisely studied in all functional gears application. The fatigue procedure prompting tooth breakage is separated into crake propagation period and crack initiation.

The primary crack can be framed because of different reasons. The most widely recognized reasons are short-term over-load, material imperfections, deserts because of mechanical or material fatigue and thermal treatment. The primary crack then spreads under imprudent loading tillit reached to critical length, when thorough tooth breakage happens. The working lifespan of gear with crack in tooth root can be calculated numerically (e.g. with finite element analysis FEA) or experimentally. The fatigue life of parts subjected to sinusoidal loading can be evaluated by utilizing cumulative damage theories. Their extension to arbitrary load, through straight forward, may not be exceptionally precise inferable to inherent scatter presentation by the fatigue phenomena. Because of loading on the structureand complex geometry FEA finite element analysis is preferably adopted.

Literature Review

Gears are a discriminating part in the rotating apparatus industry. Different examination techniques, for example, theoretical, numerical, and experimental, have been done during the time in regards to gears. Because experimental testing can be costly theoretical and numerical techniques are favoured. Therefore, various numerical models of gears have been created for distinctive purposes. This part displays a brief audit of the papers as of current distributed in the regions of gear configuration, transmission mistakes, vibration investigation, and so forth, also including brief data about the models, estimations, and presumptions made.

Wyluda and Wolf [1] conducted an elastic-plastic Finite Element Analysis (FEA) of the quasi-static loading of two acetal copolymer gears in mesh. The compared applied load versus gear rotation set with genuine experimental results. By varying thickness between rim and web gear geometryis modelled. Plane strain components were utilized as a part of the finite element model. Gear tooth failure has been considered and modelled utilizing strategies for deactivating and isolating components when the elasticity is surpassed. Accordingly, the mechanical conduct and forecast of copolymer acetalgear is truly complex. PC simulations and part testing has blended a superior comprehension of copolymer acetalgearsdesign. They also concluded that the linear elastic approach is only suitable when the machines are under

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low loads and distortions. Along these lines, performing non-linear examination is keyto advance a gear set.

In 2003, Baroneet al. [2] targeted at finding the behaviour of a face gear transmission with respect to contact path under loads and stresses, for an unchanged gear set with shaft arrangement and amendment on pinion profile. 3D CAD system with a FEA code by simulating the meshing of gear with three teeth and pinion, using meshed elements and an algorithm has been studied. The concluded that the load influenced the theoretically calculated contact paths, contact areas, contact length and load sharing. They also noted that usefulness of numerical approach on meshing problem with complex geometry. Other considerations are surplusesloads due to pinion disarrangement and change of contact areas.

In 2001, Howard et al. [3] discover the friction effects on the resultant gear case vibration utilizing a simple gear dynamic model. The model comprisesdevelopment of a FEA (Finite Element Analysis) to see the consequence of deviations in gear tooth torsional mesh stiffness as gears mesh. A dynamic equation has been developed for frictional force between teeth. Frequency spectrum displays single tooth crack effects. They indicate theinfluence of the tooth crack for all dynamic variables in the time waveforms neglecting friction. They also show that when with friction consideration into the model diagnostic techniques worked clearly, while results are more or less same in most cases.

In 2005, Wang and Howard [4] conducted FEA for high contact ratio gears in mesh. The numerical modelhas been developed under quasi-static conditions when gears in mesh. Other consideration are tooth root stress against various input loads over a complete mesh cycle, load-sharing ratio, contact stress, transmission error, combined torsional mesh stiffness. Numerousmodifications in tooth profile have been presented and comparisons between the results suggest optimal profile modification to gain the maximum benefit of high contact ratio gears.

One year later (2006), Wang and Howard [5] examinednumerous 2D and 3D gear models utilizing(FEA) finite element analysis. In their models they included contact analysis between teeth in mesh, a gear body, and teeth with and without a crack at the tooth root. They compared the results for different parameters liketooth stresses and stress intensity factors, torsional mesh stiffnessthat has been obtained under assumptions of plane stress, plane strain, and 3D analysis. In their models face width variation has also been taken into account.They got accurate results by FEA. Theyconcluded that error in 2D modelling can be significant

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when tooth root fatigue crack is considered. Therefore, 2D solutions can only be applied in very narrow range. Ignoring these errors (fatigue analysis) leads tosignificant errors in results.

In 2007, Carmignaniet al. [6] numerically simulate the dynamic behaviour of a faulted geartransmission. They estimated meshing stiffness as a function of the gearangular position statistically using finite element analysis (FEA) with gear meshing models. The teeth deformation under load condition and the faulted gears for example tooth cracks of different lengths atdifferent locations on the tooth flank were taking into account in the simulations.Simulink tool has been used to numerically simulate withdifferent applied torques and gear angular velocities conditions. They concluded fracture causesdeviation in the meshing stiffness with faulty tooth in meshing. If the cracked zone is loaded between tooth root andcontact point, crack affects stiffness. Nevertheless, if more teeth are in contact, load will be shared by uncracked teethalso, which unloads the cracked tooth and therefore reduces the effect by stiffnessdisturbance.

Gear Design and Calculations

Overview

The foremosttenacity of gearing is to transfer motion from one shaft to another. Motion cannot be transmitted if there is any mistake or error on the gears. Errors on the gears may abolish or severely damage the parts in the gearbox. Therefore, it is very essential to deeply understand the questions of gearing. In order to improve knowledge of gearing, awareness about the design of gear and theory of gear tooth action should be taken.

Types of gears

There are wide ranges of gears utilized by industry, yet every one of these gearshas the samereason, which is to transmit motion starting from one shaft toother. Generally, gearing comprises of a couple of gears with axes are either parallel or at 90°. Among all the gears in the world, the utmostfrequentlytalk over gears are spur gear, helical gear, bevel gear, and worm gearing. Easiest form of gears is spur gears; they comprises of teeth parallel to the axis of revolution. $14\,{}^{1}/_{2}{}^{\circ}$, 20° and 25° pressure angle are the common pressure angles utilized for spur gears. Advantage of a low pressure angle gear is smoother and noise less tooth action.Interestingly, bigger pressure angle gears have the benefits of better load conveying capacity.

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Materials selection

Different materials used for gear manufacturing are:

1. Steel
2. Cast iron
3. Aluminium
4. Bakelite
5. Nylon
6. Plastics
7. Bronze
8. Phenolic

Basic terms of spur gear

*Module:*Ratio of diameter to number of teeth. m= d/n

*Face width:*Width along the contact surface between the gears.

*Tooth thickness:*Thickness of the tooth along the pitch circle.

Addendum: Radial distance between the pitch circle and the top land of the gear.

Dedendum: Radial distance between the pitch circle and the bottomland of the gear.

*Pressure angle:*Angle between the line joining the centres of the two gears and the common tangent to the base circles.

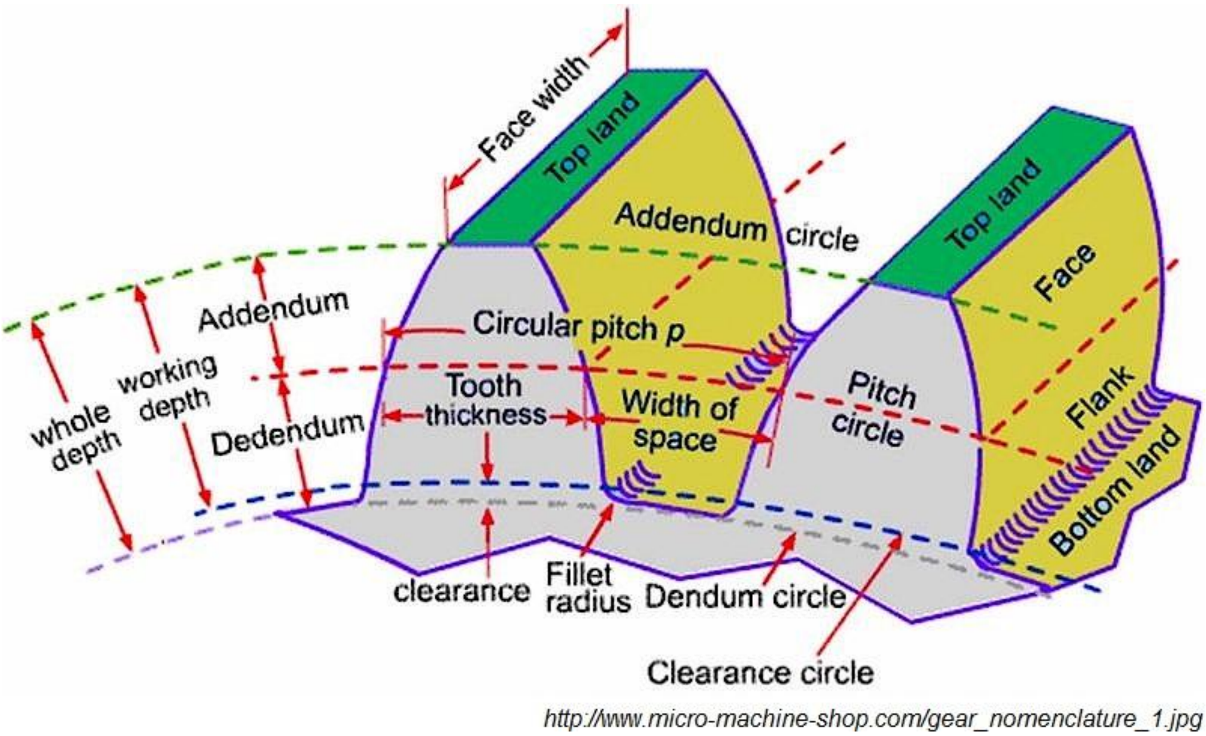

Fig. 1 Gear nomenclature

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Design procedure of spur gear

Input data: Horse power, Speed of the driver, speed ratio, Working Life, Working conditions.

1. *Select suitable materials for pinion and wheel*

   From the design data book first we will select the type of material for pinion and wheel based on the input parameters.

   [σ_b] ofwheel and pinion for minimum module and [σ_c].

   Surface hardness (We will check the difference between the surface hardness values of pinion and wheel, and it should be $\geq$ 30 HB, if both pinion and wheelsurface hardness values is< 350 HB. Pinion must be harder than wheel)

2. *Assume the pressure angle, if not given*

   There are three pressure angle values for spur gear$14\,{}^{1}/_{2}{}^{\circ}$, 20° and 25°.Usually it is 20°values considered.

3. *Find the design torque transmitted by pinion*

   Then we will calculate the design torque based on the equation, $P = \frac{2\pi n\, M_t}{60}$, where P is given power in *kW,* n is the speed of pinion in rpm and M_t is the torque. Then we will assume the value of k_dk from the design data book and finally design torque will be calculated by the equation shown in (c)

   a. Calculate M_t. Use the eqn.$M_t = 97420\,\frac{kW}{n}$
   b. Initially assume k_dk value
   c. Calculate the design torque $[M_t] = M_t k_d k$

4. *Determination of minimum Centre Distance (C.D)*

   After calculation of design torque we will calculate the centre distance between the wheel and pinion by the eq. represented in step (c). But before that E_{eq} which is equivalent young's Modulus for wheel and pinion will be calculated which appears in centre distance eq. We will also assume the value of ψ=b/a, where 'b' is face width of the gear and 'a' is centre distance.

   a. Determine$E_{eq} = \frac{2E_1E_2}{E_1+E_2}$, based on the selected materials.
   b. Selectψ.
   c. Calculate the minimum centre distance 'a'. Use [σ_c] of weaker material.

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$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{\sigma_c}\right)^2 \frac{E_{eq}[M_t]}{i\psi}}$$

5. *Determination of minimum module*

After calculation of centre distance we will calculate the minimum module based on the bending strength [σ_b] by the eq. shown in step (e). For this we will assume the minimum number of teeth for pinion then we will assume the value of ψ_m=b/m, where ‘b’ is width of the gear and ‘a’ is module of the gear. We will calculate the values of from factor from design data book based on the number of teeth on the pinion. We will calculate the value of module and will round it to the standard value from design data book.

a. Assume Z_1. number of teeth on pinion (minimum 18)
b. Select ψ_m
c. Select form factor Y corresponding to Z_1
d. Calculate minimum module. Use [σ_b] of weaker material
e. Round off to standard value.

$$m \geq 1.26\sqrt[3]{\frac{[M_t]}{Y[\sigma_b]\psi_m Z_1}}$$

6. *Determination of centre distance, pitch circle diameter and width of the gears*

After calculation of the updated module the necessary terms will be calculated like gear ration, updated centre distance, pitch circle diameter of the wheel and pinion, face width of the gear ‘b’ based on both the modules calculated ψ and $\psi_m$and will choose the higher value.

a. Determine Z_2 so that $i = \frac{Z_2}{Z_1}$
b. Determine the Centre Distance a’
c. If a’ >a (minimum C.D. already calculated), take a’ as the final centre distance (Don’t change to Standard value)

   If a’<a, increase Z_1& Z_2 (or) module and again calculate a’ so that a’>a.
d. Calculate d_1& d_2, the pitch diameters of pinion and wheel respectively
e. Calculate b (width of the gear wheel) using the values of ψ and ψ_m and take the bigger value.

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