DESIGN OF SPUR GEAR AND ITS TOOTH PROFILE ON MATLAB

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ABSTRACT

Spur Gears are the most widely recognized means of transmitting power in the current engineering applications. They change 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 utilized to solve scientific and research problems due to its accuracy and numerous built in functions which makes it flexible. In present study spur gears has been designed. A MATLAB code has been written when run, ask for the inputs and executes the essential design calculations and provides required 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 a gear 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.

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.
Gear Design and Calculations

Basic terms of spur gear
Module: Ratio of diameter to number of teeth. \( m = \frac{d}{n} \)
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.

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.
   \([\sigma_b]\) of wheel and pinion for minimum module and \([\sigma_c]\).
   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 wheel surface 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 14.5°, 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 \( k_{d_k} \) 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 \( k_{d_k} \) 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 \( \psi = b/a \), where ‘b’ is face width of the gear and ‘a’ is centre distance.

a. Determine \( E_{eq} = \frac{2E_1 E_2}{E_1 + E_2} \), based on the selected materials.

b. Select \( \psi \).

c. Calculate the minimum centre distance ‘a’. Use \([\sigma_c] \) of weaker material.

\[
a \geq (i + 1)^3 \sqrt[3]{\frac{0.74}{\sigma_c}} \frac{E_{eq} [M_t]}{i b}
\]

5. Determination of minimum module

After calculation of centre distance we will calculate the minimum module based on the bending strength \([\sigma_b] \) 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 \( \psi_m = b/m \), 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 \( \psi_m \)

c. Select form factor Y corresponding to \( Z_1 \)

d. Calculate minimum module. Use \([\sigma_b] \) of weaker material

e. Round off to standard value.

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

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 \( \psi \) 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 \( \psi \) and \( \psi_m \) and take the bigger value.
Gear input parameters

1. Power to be transmitted = 15KW
2. Speed of pinion in rpm = 1440 rpm
3. Speed of gear in rpm = 500 rpm
4. Pressure angle = 20°
5. Number of teeth in pinion = 25
6. Life of gear

![Fig. 2 (a and b) Input parameters](image-url)
### Gear output parameters (RESULT)

1. Pitch circle diameter = 10.000000
2. Face width = 9.700000
3. Dedendum circle diameter = 9.120000
4. Addendum circle diameter = 10.640000
5. Centre distance = 19.400000
6. Module = 4.000000
7. Addendum = 0.320000
8. Dedendum = 0.440000
9. Working depth = 0.640000
10. Clearance = 0.120000
11. Circular pitch = 20.315632

![Fig. 3 Gear tooth profile](image-url)
References


