A COMPARATIVE STUDY OF DESIGN OF SIMPLE SPUR GEAR TRAIN AND HELICAL GEAR TRAIN WITH A IDLER GEAR BY AGMA METHOD

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ABSTRACT

In recent times, the gear design has become a highly complicated and comprehensive subject. A designer of modern gear drive system must have to remember that the main objective of gear drive is to transmit higher power with comparatively smaller overall dimensions of the driving system which can be constructed with minimum possible manufacturing cost, runs reasonably free of noise and vibration and which requires little maintenance. In this paper single stage spur gear train and helical gear train with a idler gear are designed by American Gear Manufacturing Association (AGMA) standard. A idler gear is placed between two gearwheel to obtained the same direction of rotation. AGMA stress equation is used to determined the tooth bending strength and surface contact strength. As a result, dimensions of gears are find out and comparative study is carried out to select the optimum design of gear train for a given input parameter

KEYWORDS- Spur gear, Helical gear, AGMA standard, bending stress, contact stress.

INTRODUCTION

In Engineering and Technology the term gear is defined as a machine element used to transmit motion and power between shafts by means of progressive engagement of projections called teeth. In case of Spur gears the teeth are cut parallel to the axes of shaft. The profile of the gear tooth is in the shape of involute curve and it remains identical along with the entire width of the gear wheel, the teeth are parallel to the axes of shaft. Spur gears are used only when the shaft are parallel. Spur gears impose radial loads on the shaft. In case of helical gear, the teeth are cut at an angle with the axes of shaft. Helical gear have involute profile similar to that of spur gears, However this involute profile is in a plane which is perpendicular to tooth element. The magnitude of the helix angle of pinion and gear is same however the hand of helix is opposite. A right hand pinion meshes with left hand gear and vice versa. Helical gear impose radial and thrust loads on the shaft. Spur gear generates noise in high speed application due to sudden contact over entire face width between two meshing teeth. In helical gears the contact between two meshing teeth begins with a point and gradually extends along the tooth, resulting in quite operation. Therefore helical gears are preferred for high speed power transmission. From the cost consideration spur gears are cheapest, they are not only easy to manufacture but there exist a number of methods to manufacture them. Whereas the manufacturing of helical gear is specialized and costly operation. There are two basic modes of gear tooth failures-breakage of tooth due to static and dynamic loads and the surface destruction. The complete breakage of tooth can be avoided by adjusting the parameters in the gear design such as module and face width, so that beam strength and wear strength of gear tooth is more than sum of static and dynamic loads. Selection of right kind of gear for right kind of application is an open issue and there is no ready method which can be specified for the purpose. In case of automobiles, which uses spur, helical as well as bevel gear for transmission gear boxes and differentials, gears are generally cut from low alloy steel forging which after teeth cutting are heat treated to the desired hardness, the gear tooth should be very accurate in the initial stage itself as no post hardening, tooth correcting processes are employed. Case hardened automobiles gears usually have a surface hardness of around 60 HRC and core hardness of 30 HRC. This imparts the gear properties of wear resistance, strength and shock absorbing capability.

LITERATURE SURVEY

This chapter presents the work did by the researcher in the field of design of spur and helical gear. Some of them are summarised below.

Ishan Patel Dr. M.S. Murthy compared the bending stresses for different number of teeth of spur gear obtained using MATLAB Simulink with AGMA and ANSYS, results obtained from both ANSYS and Simulink are close to the results of AGMA also bending stress increases with increase in number of teeth. Authors concluded that simulink is also an equivalent method if modeled properly by using curve fitting equation.[1]

Parveen Kumar and Harsh Raghuvanshi had done the work on Design & Analysis of a Spur Gear in different Geometric Conditions like the effect of gear ratio, face width and normal module on dynamic tooth load by changing the concentration of SIC in SIC based aluminium gear. In this Lewis method is used for design of spur gear. Addition of SIC increases the strength of Spur Gear. [2]

M. S. Hebbal, V. B. Math, B. G. Sheeparamatti had worked on Reducing the Root Fillet Stress in Spur Gear Using Internal Stress Relieving Feature of Different Shapes. In this work, combination of circular and elliptical stress relieving features are used and better results are obtained than using circular stress relieving feature alone. The root fillet stress calculated using AGMA standards and compared with FEA. Stress relieving features of various sizes were introduced on gear teeth at various locations and analysis revealed that, combination of elliptical and circular stress relieving features at specific, locations are beneficial than single circular, single elliptical, two circular or two elliptical stress reliving features.[3]

Y. Sandeep kumar, R.K. Suresh, B.Jayachandraiah had done the investigation on Optimization of design based on Fillet radius and tooth width to minimize the stresses on the Spur Gear. The stresses were calculated by using Lewis Equation and AGMA Standards and then compared with finite element analysis. The Stress at the contact and fillet region decreases with the increase of face width. The results obtained from finite element analysis are found to be in close agreement with the calculated stresses based on AGMA standards and Lewis Equation. [4]

B. Venkatesh, V. Kamalaesign had worked on the design, modelling and manufacturing of helical Gear. In this work, dimensions of gears are found out by theoretical method (lewies method) and structural analysis on a high speed helical gear used in marine engines have been carried out. The stresses generated and the deflections of the tooth have been analyzed for different materials and found the best suited material for the marine engines based on the results.[5]

Venkatesh and Mr. P.B.G.S.N. Murthy had done the investigation on design and structural analysis of high speed helical gear using ansys. In this paper bending and contact stresses are calculated by using AGMA stress equation. Results obtained are compared and it is found that Induced bending stress is a major function of number of teeth and helix angle influence is less on contact stresses, Error percentage is around 6 % in bending stresses and around 1 % in contact stress.[6]

A.Sathyanarayana Achari, R.P.Chaitanya and Srinivas Prabhu had done the investigation on comparison of bending stress and contact stress of helical gear as calculated by AGMA standards and FEA. Parametric study is conducted by varying the face width and helix angle and concluded that maximum bending stress decreases with increasing face width and it will be higher on gear of lower face width with higher helix angle. [7]

Above literature gives information of work did by the various researchers in the field of design of gear. Previously, lewies equation and Buckingham's equation are used for design of gears to avoid bending and pitting failure. Modifications are made in conventional design process and various factors are used to consider the effect of dynamic loading in the AGMA method which gives the accurate design of gears. It involves less no. of iterations. Finally AGMA results are compared with the Finite element analysis to check the accuracy of design. Hence in the proposed study, AGMA method is used to design the simple gear train with a idler gear by using spur and helical type of gear and comparative study is carried out to select the optimum design of gear train for the given input parameter.

DESIGN METHODOLOGY

In the gear design, the surface contact strength and tooth bending strength of the gears are assumed to be one of the major contributors for the gear failures in a gear pair. Failure by bending will occur when the significant tooth stress equals or exceeds either the yield strength or the bending endurance strength. A surface failure occurs when the significant contact stress equals or exceeds the surface endurance strength. Therefore, the determination of stresses in a gear has very important to reduce or to minimize the failures and for optimal design of gear pairs. Here the American Gear Manufacturers Association (AGMA 2101-DO4) standard is used, which gives the fundamental rating factors and calculation methods for involute spur and helical gear teeth. Gears operates in pairs, the smaller of the pair being called the "pinion" and larger the "gear", usually the pinion drives the gear and system acts as speed reducer and torque converter. The simple gear train system of spur and helical gear consist of idler gear which act as a pinion and will be driving two gears simultaneously one on each side. Since the pinion have lesser no. of teeth than their respective gearwheels, it is safe to assume that gears wheels are stronger. Hence the pinion is design to check bending and wear failure.

INPUT PARAMETER

Table No.1 Input Parameter

Name of parameter	Value of parameter
Input torque (From motor)	960 N-m
Input Speed (Speed of pinion)	263 rpm
Centre distance	100 mm
Gear ratio	1.74
Number of teeth on pinion	19
Number of teeth on gear	33
Speed of gear	151.15 rpm
Load cycle (Considering 300 hrs of running of pinion)	9.468×10^6 cycle
Reliability	99 %

PRELIMINARY DRAWING AUTOCAD

Fig.3.2 shows the Single stage gear train with idler gear. In this gear train, pinion is the input which rotates in clockwise direction and act as an idler gear which will be driving two gears simultaneously one on each side in anticlockwise direction. As the reduction ratio is less i.e. 1.74, reduction will be carried out in single stage and it is called as single stage gear train or simple gear train.

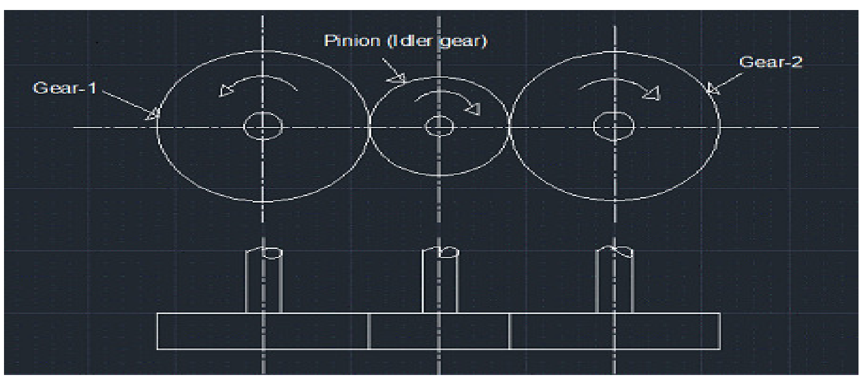

Figure 1: Simple Spur or Helical Gear Train with Idler Gear

SELECTION OF MATERIAL

For both the pinion and gear wheels similar material is selected. i.e. 17CrNiMo6 case hardened alloy steel. Chrome-Nickel-Molybdenum case hardened steel has HRC 60-63 and a tough strong core with a typical tensile strength range of 900-1300 MPa, in small to fairly large sections. As the similar material is used for pinion and gears and pinion drives two gear simultaneously, pinion is subjected to reversible bending i.e. tooth of pinion engages two times in one revolution and hence pinion is the weakest member and design of pinion is carried out to avoid the bending and pitting failures of gear tooth.

THEORETICAL CALCULATIONS

DESIGN OF SIMPLE SPUR GEAR TRAIN

- Determination of Nominal Metric module (m) in plane of rotation
  Centre distance for spur gear,

$a = \frac{m(N_P + N_G)}{2}$ .....(1), $100 = \frac{m(19+33)}{2}$, Module (m) = 3.85 mm

The size of gear tooth is specified by the module. Preferring the standard value of module given under Choice-1 series, Module (m) = 4 mm

Modified Centre distance , a = 104 mm

- Pitch circle diameter of pinion and gear for spur gear, $d = m \times N$.....(2)

Let, d_P = pitch circle diameter of pinion, $d_P = m \times N_P = 4 \times 19$ = 76 mm From (2)

d_G = pitch circle diameter of gear, $d_G = m \times N_G = 4 \times 33$ = 132 mm From (2)

- Calculation of Pitch line velocity of pinion

Pitch line velocity of pinion is given by the following formula

$$V = \ldots \frac{\pi dn}{60 \times 1000} \text{......(3)}$$

$$V = 1.1016 m/\min$$

- Calculation of Tangential load on tooth

$$W_t = \frac{2 \times T_M}{d_P} \quad \text{...... (4)}$$

Since the pinion is in reversible bending transmitted tangential load will be divide equally on the tooth.

$$W_t = 12631.58 \text{ N}$$

- Face width of tooth: Taking, $F =$ 10 to 12 times module, Taking Face width of 45 mm.
- Calculation of bending stress by AGMA method
  - AGMA Bending Stress

$$\sigma = \frac{W_t K_a}{K_v} \frac{P_d}{F} \frac{K_S K_m}{J} \text{......(5)}$$

$$\sigma = \frac{12631.58 \times 1}{0.91844} \times \frac{1}{4 \times 45} \times \frac{1 \times 1.3}{0.32}$$

AGMA Bending stress: 310.40 N/mm^2

- Calculation of Allowable Bending strength by AGMA method

Since the pinion is subjected to reverse bending (loading in both directions), 70% of the allowable bending strength should be used.

$$\sigma_{all} = \frac{S_t K_L}{K_T K_R} \text{.......(6)}$$

$$\sigma_{all} = \frac{480 \times 0.70 \times 1.01863}{1 \times 1}$$

Allowable Bending strength= 342.26 N/mm^2

- Safety Factor S_F against bending fatigue failure

$$S_F = \frac{\sigma_{all}}{\sigma} \text{........(7)}$$

$$S_F = 1.1$$

- Calculation of contact stress by AGMA method

AGMA contact stress equation

$$\sigma_c = C_p \left(\frac{W_t C_a}{C_v} \frac{C_s}{Fd} \frac{C_m C_f}{I} \right)^{1/2} \text{.......(8)}$$

$$\sigma_c = 191 \left(\frac{12631 \quad .58 \times 1}{0.91844} \frac{1}{45 \times 76} \frac{1.3 \times 1}{0.102048} \right)^{1/2}$$

AGMA Contact stress = 1367.1 N/mm^2

- Calculation of Allowable contact strength by AGMA method

Allowable contact strength

$$\sigma_{c,all} = \frac{S_c C_L C_H}{C_T C_R} \text{...........(9)}$$

$$\sigma_{c,all} = \frac{1550 \times 1.00127 \times 1}{1 \times 1}$$

Allowable contact strength = 1551.96 N/mm^2

- Safety actor s_H against pitting failure

$$s_H = \frac{\sigma_{c,all}}{\sigma_c} \text{...........(10)}, \quad s_H = \frac{1551.96}{1367.1}, \quad s_H = 1.135$$

Design of simple helical gear train

- Helix angle (φ)=15^0

- Determination of Normal module (m_n) in plane perpendicular to the tooth element

Centre distance for helical gear

$$a = \frac{m_n (N_P + N_G)}{2 \times COS\varphi}, \text{ .... (11)}$$

Module (m_n) = 3.72 mm, Preferring the standard value

Normal Module (m_n) = 3.75 mm

- Modified Centre Distance

$$a = \frac{3.75 \times (19 + 33)}{2 \times \cos 15}$$ ....From (11) Modified centre distance (a) = 100.94 mm

- Pitch circle diameter calculation

$$d_P = \frac{m_n \times N_P}{\cos \varphi} = \frac{3.75 \times 19}{\cos 15} = 73.76 \text{ mm ....Using (2)}$$

$$d_G = \frac{m_n \times N_G}{\cos \varphi} = \frac{3.75 \times 33}{\cos 15} = 128.12 \text{ mm........Using (2)}$$

- Calculation of Pitch line velocity of pinion

Pitch line velocity of pinion is given by the following formula

$$V = \frac{\pi \times 73.76 \times 263}{60 \times 1000} = 1.01576 \text{ m/min.....Using (3)}$$

- Calculation of Tangential load on tooth

$$W_t = \frac{2 \times 960}{0.07376} = 26030.36 \text{ N....... Using (4)}$$

Since the pinion is in reversible bending transmitted tangential load will be divide equally on the tooth.

$$W_t = 13015.18 \text{ N}$$

- Face width of tooth (F): Minimum face width required for helical gear is given by the following formula, $F \geq \dfrac{\pi m_n}{\sin\varphi}$ …….(12), Face width= 45.5 mm , Taking 46 mm .

- Calculation of bending stress by AGMA method

AGMA Bending stress

$$\sigma = \frac{13015.18 \times 1}{0.9188} \times \frac{1}{3.75 \times 46} \times \frac{1 \times 1.2}{0.56} \text{..........} \textit{ from } (5)$$

AGMA Bending stress = 175.97 N/mm^2

- Allowable Bending strength by AGMA method

Allowable Bending strength= 342.26 N/mm^2 (As Spur pinion and helical pinion material is same)

- Safety Factor S_F against bending fatigue failure

$$S_F = \frac{342.26}{175.96} \text{.........} \textit{from}(7)$$

$$S_F = 1.95$$

As the factor of safety is 1.95, Value of face width is reduced and taken as 40 mm

Bending stress, $\sigma = 202.36$ N/mm^2

$$S_F = 1.69$$

- Calculation of contact stress by AGMA method

$$\sigma_c = 191\left(\frac{13015.18}{0.9188} \times \frac{1}{40 \times 73.76} \times \frac{1.2 \times 1}{2.288}\right)^{1/2} \text{....} \textit{ from } (8)$$

AGMA Contact stress: 303.088 N/mm^2

- Allowable Contact Strength: 1551.96 N/mm^2 (As the material is same for the both the gear trains)

- Safety actor s_H against pitting failure

$$s_H = \frac{1551.96}{303.088} \text{.......} \textit{from}(10)$$

$$s_H = 5.12$$

RESULTS OF AGMA METHOD

RESULTS OF BENDING STRESS AND CONTACT STRESS

Table No.2 Results of Bending stress and Contact Stress for Spur and Helical Pinion

Type of gear Spur/ Helical Pinion	Bending stress [N/mm^2]	Allowable bending strength [N/mm^2]	Safety factor against bending	Contact stress [N/mm^2]	Allowable contact strength [N/mm^2]	Safety factor against pitting
Spur pinion	310.40	342.26	1.10	1367.07	1551.96	1.14
Helical Pinion	202.36	342.26	1.69	303.08	1551.96	5.12

DIMENSIONAL PARAMETERS FOR THE GEAR TRAINS

Table No.2 Dimensional parameters of Simple spur and helical gear train

Name of Parameter	Value for spur gear train		Value for helical gear train	
	Pinion	Gear	Pinion	Gear
Number of teeth	19	33	19	33
Pitch circle diameter	76 mm	132 mm	73.76 mm	128.12 mm
Module	4 mm	4 mm	3.75 mm	3.75 mm
Pressure angle	20^0	20^0	20^0	20^0
Face Width	45 mm	45 mm	40 mm	40 mm
Addendum	4 mm	4 mm	3.75 mm	3.75 mm
Dedendum	5 mm	5 mm	4.69 mm	4.69 mm
Clearance	1 mm	1 mm	0.94 mm	0.94 mm
Tooth thickness	6.2832 mm	6.2832 mm	5.9 mm	5.9 mm
Fillet radius	1.6 mm	1.6 mm	1.5 mm	1.5 mm
Tip circle diameter	84 mm	140 mm	81.26 mm	135.12 mm
Root circle diameter	66 mm	122 mm	64.38 mm	118.74 mm
Base circle diameter	71.42 mm	124.04 mm	69.3 mm	120.38 mm
Centre Distance	104 mm		100.94 mm	

COMPARISON AND DISCUSSION

Simple speed reducer gear train is designed by using spur and helical type of gears for a gear ratio of 1.74 and centre distance of 100 mm. 20^0 full depth involute tooth system is used for both the gear trains. As the similar material is used for pinion and gears the value of Allowable bending strength and Allowable contact strength are same for spur and helical pinion, Result of AGMA standard shows that induced bending stress and contact stress are more in case of spur pinion than the helical pinion. The value of Module is 4 mm in case simple spur gear train and 3.75 mm in case of simple helical gear train. As the module specifies the size of gear, size of spur gear train is more than simple helical gear train and hence more material is required to manufacture a simple spur gear train. Face width is more in case of simple spur gear train i.e. 45 mm, whereas simple helical gear train with comparatively less face width of 40 mm gives the better results for gear tooth stresses. Both the gear train are designed to obtained fixed centre distance of 100 mm, the result of simple helical gear train is nearest to the design requirement i.e. 100.94 mm, whereas in case of simple spur gear train centre distance is 104 mm which is more than the requirement.

CONCLUSION

Theoretical design of simple spur gear train and simple helical gear train carried out using standard design formulae as per AGMA procedure.AGMA result of the simple helical gear train satisfied the design requirement for the given input parameter than the simple spur gear train. For the simple helical gear train value of module and face width are small with lower value of helix angle, which gives the compact arrangement. Helical gear with higher value of helix angle increases the contact stresses, hence lower value of helix angle is selected. Helical gear can bear the more load and run quietly, the disadvantage is axial force caused by the helix form, hence proper type of bearing is selected to take the effect of axial force. Whereas the value of module and face width are more for simple spur gear train, which increases the size of gear train.

ACKNOWLEDGEMENT

I express my sincere gratitude to my guide, Prof. Ashtekar Jaydeep (Assistant Professor, Mechanical Engineering Department, and VACOE Ahmednagar) for their valuable guidance, proper advice during the work. I would like to thank Mr. Kulkarni Mangesh and Mr. Ghogare Vikas, (Scientist, VRDE, Ahmednagar) for their kind guidance, support and sharing of their knowledge.

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