Probabilistic Design of Spur Gear considering Bending Failure

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ABSTRACT

Among the various mechanical power transmission means, gears are the most rugged and durable, but proper designing of the gears is a challenging task. The present trend in the industry is to follow AGMA (American Gear Manufacturing Association) to design a reliable gear. This approach doesn't consider the practical variation occurring in gearing. In the present work, a design methodology of considering statistical variations in various parameters of spur gear has been presented. The importance of the proposed approach has been illustrated considering three cases. The appropriate values of correction factors are accounted considering the statistical variations in operating and geometric conditions in the form of their mean and standard deviation values.

Key words: Reliability, gear, statistical variation, AGMA.

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INTRODUCTION

Among all power transmission devices, gears are the most robust device. In the present work a study on external spur gears has been studied. These gears experience combination of both high speed and heavy load conditions, and improper design results in increases the noise level and reduces the gear service-life. Designing a gear for long life is a challenging task and it becomes more severe in the case of fluctuating operating conditions, variable material properties and statistical geometric dimensions.It is interesting to note that statistical variation in operating conditions is very common in most of mechanical components such as bearings [1-33], brakes [33-49], gears [49-54]. Therefore there is need of a robust design methodology to incorporating the statistical variations in the components.

In the current industry practice, AGMA (American Gear Manufacturing Association) standards has been followed to design a gear. This standards includes a large number of empirical "design factors". The selection of the values of design factors is very subjective. There is a need of design methodology to provide appropriate information related to the design factors so that subjective decision can be minimized.

In the present work a probabilistic design approach has been proposed incorporating all the statistical variations. The proposed probabilistic approach can be applied to any machine component that necessarily involves consideration of statistical material strength and the complicated stress state to which the component is subjected.

To exemplify the present approach, three cases have been considered and results of bending stresses have been compared. An attempt has been made to explain the importance of considering statistical variations which are ignored in AGMA approach.

Gear Design using proposed approach

The fundamental stress equation used in classical (AGMA) approach for estimating bending stress is given by equation (1).

$$\sigma_b = \frac{K_v W_t}{F m Y_j} K_o K_B K_H K_s \qquad (1)$$

The detailed description of classical AGMA approach has been described in Budynas, Nisbett [55].The AGMA approach uses a single value for predefined conditions which may result in wrong design if conditions are subject to statistical variations. To overcome such problem, probabilistic approach incorporating the statistical variation in various design factors needs to be used. In this approach, it is assumed that each variable follows the probability density function, expressed in terms of its mean μ_Q and its standard deviation $\hat{\sigma}_Q$ as given in equation (2):

$$f(Q) = \frac{1}{\hat{\sigma}_Q \sqrt{2\pi}} exp\left[-\frac{1}{2}\left(\frac{Q-\mu_Q}{\hat{\sigma}_Q}\right)^2\right] \qquad (2)$$

To normalize the Eq. (1), variables can be expressed in terms of 'normal variable' having a mean of zero and a standard deviation of unity, such as:

$$Z = \frac{Q-\mu_Q}{\hat{\sigma}_Q} \qquad (3)$$

Reliability is the probability that any machine system will perform its intended function satisfactorily without failure. As stress and strength are statistical in nature, theequations 2 and 3 can be used to find the reliability of the machine component. To understand it let us consider the probability density functions for stress and strength, σ and S_y. The mean values of stress and strength are μ_σ and μ_S respectively. The margin of safety for any value of stress σ and strength S_y is defined as "$Q=S_y-\sigma$". In classical approach, the part safety is decided based on the positive value of $\overline{Q} = \mu_s - \mu_\sigma$. However, there are chances of overlap in stress and strength distributions, which can be determined by considering the cumulative probability of parameter Q having value greater or equal to zero. To find the chance that Q≥0, the value of 'Z' variable (equation (4)) is obtained for Q=0.

$$Z = \frac{Q-\mu_Q}{\sigma_Q} = -\frac{\mu_Q}{\sigma_Q} \qquad (4)$$

Where $\mu_Q = \mu_S - \mu_\sigma$ and $\sigma_Q = \sqrt{\sigma_s^2 + \sigma_\sigma^2}$.

From the value of Z, the reliability of the component is estimated using the table given in A-10 of book authored by Budynas, Nisbett [55[.

In the proposed approach, the design factors given in equation (1) are considered with statistical variations to obtain the values of mean and standard deviation of the bending stress. The mean value (μ_σ) of stress is obtained by substituting the mean value of each variable given in equation (1). To determine the standard deviation of stress (σ_s), the stress equation (1) is differentiated w.r.t. individual independent variable. If the design variable is dependent then it is substituted as function of independent variables. To understand the procedure, consider the equation (1). The stress equation given in equation (1) is can be rewritten as a function of $K_{v,}$ $W_{t,}$ b, m, Y_j, K_o, K_s, K_H and K_B as given in equation (5).

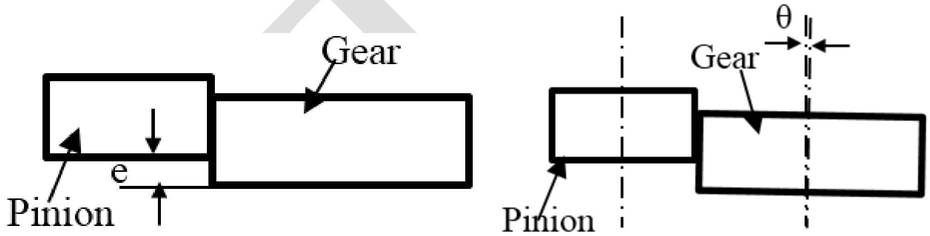
$$\sigma_b = f\left(K_v, W_t, b, m, Y_j, K_o, K_s, K_H, K_B\right) \tag{5}$$

The standard deviation of equation (5) is estimated by equation (6)

$$\sigma_b = \sqrt{\sum_{i=1}^{n}\left(\frac{\partial\sigma_b}{\partial x_i}\right)^2 \partial x_i^2} \tag{6}$$

$$\sigma_b = \sqrt{\left(\frac{\partial\sigma_b}{\partial K_v}\right)^2 \sigma_{K_v}^2 + \left(\frac{\partial\sigma_b}{\partial W_t}\right)^2 \sigma_{W_t}^2 + \left(\frac{\partial\sigma_b}{\partial b}\right)^2 \sigma_b^2 + \left(\frac{\partial\sigma_b}{\partial m}\right)^2 \sigma_m^2 + \left(\frac{\partial\sigma_b}{\partial Y_J}\right)^2 \sigma_{Y_j}{}^2 + \left(\frac{\partial\sigma_b}{\partial K_o}\right)^2 \sigma_{K_o}^2 + \left(\frac{\partial\sigma_b}{\partial K_s}\right)^2 \sigma_{K_s}^2 + \left(\frac{\partial\sigma_b}{\partial K_H}\right)^2 \sigma_{K_H}^2 + \left(\frac{\partial\sigma_b}{\partial K_B}\right)^2 \sigma_{K_B}^2}$$

As explained earlier the variables must be independent, the variable load (W_t) is dependent on the applied torque (T) and pitch diameter (D) (W_t=2T/D). Therefore *f*(W_t) is replaced by *f(T,D)*. Similarly the $K_v$is function pitch line velocity (V) and V is function of angular speed (N) and pitch diameter (D) (V=πDN/60).
In the present approach variation in load is accounted by considering variations in torque and speed; therefore there is no need to consider the over load factor (K_o).
AGMA bending geometry factor (Y_j) is function of number of teeth and nominal pressure angle, which will has zero standard deviation. Therefore, there is no need to consider standard deviation of Y_j. Similarly, deviation in value of module (m) is almost negligible, so no need to consider its standard deviation.



(a) Axial offset (b) Angular Offset

Fig. 1 axial and angular offset during gear and pinion engagement

The load distribution factor (K_H) is the due to the improper engagement of gear and pinion as shown in figure 1. Due to the assembly error in engaging the gear and pinion, as well in the mounting of bearings, the faces of gear and pinion shift and as a result effective face width in contact decreases.
It is interesting to note that gear effective width depends on the alignment and rigidity of the support bearings. Figure 1(a) depicts an axial offset between the gears, which results in reduction in contacting face width. Similarly an angular offset is shown in figure 1(b). These offsets arise due to the clearance in the bearings and coupling (i.e. Jaw coupling) between driving and driven shafts. Therefore to incorporate the design factor for load distribution, the full assembly has to be considered. However to get the accurate results, intensive study has to be performed which is out of scope of the present work.
In statistical approach, this can be accounted by consider the statistical variation in the value of face width. By considering this, there is no need to account K_H and standard deviation in face width.
In statistical approach variations in material properties and face width are accounted; therefore, there is no need to account size factor K_s. There is very rare chance of designing thin rim of gear. Generally t_r is far greater than the tooth thickness (m_B>1.2). The basic reason for stating

m_B>1.2 is that the diameter of shaft is generally much smaller than the pitch diameter of the gear. Therefore, there is no need to account K_B.
As per discussions provides in previous paragraphs, Eq (5) and Eq (6) can be expressed as:

$$\sigma_b = f\left(K_v, T, D, b_{\min}, m, Y_j\right) \qquad (7)$$

$$\sigma_b = \sqrt{\left(\frac{\partial\sigma_b}{\partial T}\right)^2 \sigma_T^2 + +\left(\frac{\partial\sigma_b}{\partial D}\right)^2 \sigma_D^2 + \left(\frac{\partial\sigma_b}{\partial N}\right)^2 \sigma_N^2 + \left(\frac{\partial\sigma_b}{\partial b}\right)^2 \sigma_b^2} \qquad (8)$$

To understand the importance of the proposed design approach against the classical AGMA approach, let us consider the following example:
Example: Determine safety of a medium quality (Cut or milled profile) pinion of spur gear system to transmit a torque of 550Nm @ 4500 rpm with speed reduction of 3.5. Assume the pressure angle = 20°, number of teeth (Z_p)=29 and module(m)=6mm. The corrected endurance strength of gear material varies in the range of 300 MPa to 400 MPa. Standard deviations in torque is in speed 100 rpm and in pitch diameter 0.25mm. Due to crowning and manufacturing errors, the face width varies from 40mm to 42mm. Due to possibilities of angular and parallel offset, effective face width may varies in the range of 34mm to 42mm. Considered three cases of standard variation of (i) σ_T= 5 Nm, and (ii) σ_T= 30 Nm, and (iii) σ_T= 100 Nm.
Solution: To determine the safety of spur gear, one of the two approaches (i) AGMA classical approach and (ii) proposed probabilistic approach can be used. In the present example, both the approaches have been used and comparison among the obtained results has been presented.

- Classical Approach

The stress calculated using the AGMA equation given in equation (1) is estimated as follows:
(i) As per provided information, the dynamic factor for milling operation of pinion is given as:

$$K_v = \frac{6.1+V}{6.1} \qquad (10)$$

The pitch line velocity (V) is a function of pitch circle diameter of pinion (D) and speed (N). On incorporating these values, equation (16) is rewritten as:

$$K_v = \frac{6.1+V}{6.1} \Rightarrow 1+0.0086DN \qquad (11)$$

(ii) From the graph provided in (Richard, Budynas, [55]), the value of form factor (Y_j) for 29 number of teeth is 0.356.
(iii) b_{avg}=(34+42)/2=38mm
(iv) b_{std}=(42-34)/6=1.333mm
(v) The value of K_o for uniform moderate shock is 1.25
(vi) The value of K_s is considered to 1
(vii) The value of K_H for crowned pinion with face width 0.05 which is well assembled at the center of the shaft is 1.25.
(viii) Due to the solid gear the rim factor (K_B) is considered to be 1

Substituting the value of various factor in equation (1) we get:

$$\sigma_{\sigma_b} = \frac{K_v W_t}{bmY_j} K_o K_B K_H K_s$$

$$= \frac{2T\left(1+0.163DN\right)}{DbmY_j} 1.25\times1\times1.6\times1 = 415MPa$$

$$FOS = \frac{\left(400+300\right)}{2\sigma_Q} = 0.843$$

FOS of safety is 0.843, which is indicating under design of gear. Generally after accounting all correction factors, FOS should not be more than 1.

(ii) Probabilistic Approach

Substituting the above estimated values in equation (1) we get:

$$\sigma_s = \frac{K_v W_t}{b_{avg} m Y_j} \Rightarrow 2T \frac{(1+0.0086DN)}{b_{avg} m Y_j} \qquad (12)$$

Partial differentiation equation (12) w.r.t. D, N, T and b, we get

$$\frac{\partial \sigma_s}{\partial N} = 2T \frac{(0.0086D)}{D b_{min} m Y_j} \Rightarrow 1.7 \times 10^8$$

$$\frac{\partial \sigma_s}{\partial T} = 2 \frac{(1+0.0086DN)}{D b_{min} m Y_j} \Rightarrow 7.74 \times 10^5$$

$$\frac{\partial \sigma_s}{\partial D} = -2T \frac{(1+0.0086DN)}{D^2 b_{min} m Y_j} + 2T \frac{(0.0086N)}{D b m Y_j} \Rightarrow -1.27 \times 10^8$$

$$\frac{\partial \sigma_s}{\partial b} = 2T \frac{(1+0.0086DN)}{D b_{min} m Y_j} \Rightarrow -3.4 \times 10^9$$

To calculate the value of normalized factor 'Z', μ_{sy}, σ_{sy}, μ_Q, σ_Q is calculated as follows

σ_{sy}=(400-300)/6=16.66MPa,

μ_{sy}=(400+300)/2=350MPa.

$$\mu_\sigma = 2\frac{\sqrt{1+0.0412\sqrt{DN}\,T}}{b_{min} m Y D} \Rightarrow$$

$$1130 \times 2 \times \frac{(1+0.0086 \times 0.174 \times 4500)}{0.036 \times 0.006 \times 0.356 \times 0.174} => 224.3MPa$$

$\mu_Q = \mu_{Sy} - \mu_\sigma$ = 350-224.3=125.7MPa

(i) Case 1 (σ_T= 5 Nm)

Given σ_T=5 Nm, σ_N= 100 rpm, σ_b= 0.002m and σ_D= 0.25mm. Substituting the values obtained and given values in equation (9), the estimated value of σ_s is 6.35MPa,

$\sigma_Q = \sqrt{\sigma_{sy}^2 + \sigma_\sigma^2}$ =17.83MPa. The value of 'Z' is estimated by equation (4)

$$Z = \frac{Q-\mu_Q}{\sigma_Q} = -\frac{\mu_Q}{\sigma_Q} = -\frac{125.77}{17.83} = -7.05$$

The maximum value of reliability given by (Budynas and Nisbett, 2014) is 0.999995 for the value of Z =-4.417, while the value of Z estimated in the present case is -7.67 who's reliability will be close to 100.

From the above discussion it can be concluded that as per AGMA design the gear failed but having low fluctuation in the torque values have high reliability.

(ii) Case 2 (σ_T= 30 Nm)

For the second case σ_T= 30 Nm, the value of FOS estimated using AGMA will be same while the reliability value of the gear will be different due to variation in standard of torque. Substituting the values obtained and given values in equation (9), the estimated value of σ_s is 29.35 MPa,

$\sigma_Q = \sqrt{\sigma_{sy}^2 + \sigma_\sigma^2}$ =33.7MPa. The value of 'Z' is estimated by equation (4)

$$Z = \frac{Q-\mu_Q}{\sigma_Q} = -\frac{\mu_Q}{\sigma_Q} = -\frac{125.77}{33.7} = -3.7$$

The reliability value given by (Budynas and Nisbett, [54]) for the value of Z =-3.7 is (1-0.000108) i.e. 0.9999, which is also high reliable.

(iii) Case 3 (σ_T= 50 Nm)

For the third case σ_T= 50 Nm, the obtained value of σ_s is 29.35 MPa,

$\sigma_Q = \sqrt{\sigma_{sy}^2 + \sigma_\sigma^2}$ =96.9MPa. The value of 'Z' is estimated by equation (4)

$$Z = \frac{Q-\mu_Q}{\sigma_Q} = -\frac{\mu_Q}{\sigma_Q} = -\frac{125.77}{98.3} = -1.27$$

The reliability value given by (Budynas and Nisbett, 2014) for the value of Z =-1.27 is (1-0.102) i.e. 0.898, which is also low reliable.

Hence from the above cases it is clear that variation in the parameter greatly affects the reliability of the gears, which cannot be observed in the case of AGMA.

CONCLUSION

Gears designed using AGMA (American Gear Manufacturing Association) method requires a number of subjective design correction factors. The proposed probabilistic approach reduces the subjectivity by eliminating correction design factors and provides more realistic results. Three case studies to illustrate the procedure to use the proposed statistical approach has been considered. Based on the obtained results it can be said that the proposed approach is a robust approach and shall be used to design various machine elements.

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