Design and Optimization of Helical Gear for Bending Strength

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Abstract— Gearing is one of the most effective methods for transmitting power and rotary motion from the source to its application with or without change of speed or direction. Gears are mostly used to transmit torque and angular velocity from one shaft to another shaft. The method of optimum design is effective in the field of gear research to determine the optimum gear parameters for satisfactory design. In gear design, number of parameters is involved. The gear design also requires an iterative approach to optimize the gear parameters. From the calculation of existing design data given by the company, it can be seen that the bending stresses in the last pair of gearbox is not within permissible limit. So in this dissertation the bending strength of Helical gear is to be maximized for fixed center distance and fixed weight of gearbox, and also fulfilling the certain necessary requirements. Due to the complexity of the problem, it is complicated and time consuming to optimize the design manually. Hence in this dissertation, MATLAB optimization program (Genetic Algorithm) is used for determining the best possible combination of gear parameters and comparison of this result is made with ANSYS results.

Key words: Bending stress, Helical gear, Genetic Algorithm

I. INTRODUCTION

Gears are used in all applications where power transmission is required such as automobiles, industrial equipments, airplanes and marine vessels. The overall efficiency of any kind of power transmission machine depends on the amount of power loss in the process. The best way of transmitting power between the shafts is gears. The design of gear is a complex process generally it needs large number of iterations and data sets. If the bending stress is too high, larger module has to be used to decrease the bending stress, but this will increase the tooth size. The objective of the gear drive is to transmit high power with higher load carrying capacity and lower weight. But the gear load capacity may be limited by tooth contact conditions or by the bending strength.

II. LITERATURE REVIEW

The research in this field is carried out by some authors is given here. Upendra kumar Joshi [1] developed the three dimensional Cad Model of Helical gear. This paper investigates finite element model for monitoring the stresses induced of tooth fillet and tooth flank during meshing of gears. Bending stresses produced at critical section because of initial point contact. The involute profile of helical gear has been modelled and simulation is carried out for the bending and contact stresses. The gear bending stresses has been calculated by Lewis equation. The contact stresses is determined by Hertz contact stress equation. To find the effect of face width on bending stress, the study is conducted by varying the face width of helical gear. And it is observed that with increase in face width there is decrease in maximum bending stress.

Dr. M. S. Murthy [2] shows that the complex design problem of spur gear which requires fine software skill. It can be solved by using MATLAB Simulink which provides equivalent results to the AGMA and ANSYS. In this paper the stresses in the spur gear pair is calculated in ANSYS. After that create a Simulink model using curve fitting equation. Then result is compared with both ANSYS and AGMA. For calculation of bending stresses in MATLAB-Simulink an equation of curve fitting is formed by the data of ANSYS. By making program coding in MATLAB then solution is done. Result clearly shows that changing the number of teeth from 18 to 25, the stress is continuously increasing. For constant load and speed the minimum number of teeth gears are suitable.

Dr. V. B. Sondur [3] presented a method for investigating the bending stress at the critical section of "Asymmetric involute spur gear". The determination of tooth form factor, stress concentration factor, critical section parameters and contact ratio has been accomplished for each set of gear. The gears with different pressure angle have been modelled by using CATIA. The results obtained by theoretical method have been verified by using ANSYS. The comparative analysis of bending stress at critical section has been carried out.

Since most of the gears may be expected to reverse operation occasionally, the tooth coast sides are still required to be conjugate, but with only rudimentary performance expectations. This has the effect of producing the teeth which are thicker at the base and more pointed at the tip than standard gear tooth and are subsequently more resistant to bending. The main advantage of asymmetric gears is contact stress reduction, resulting higher torque density. The four sets of gears with different pressure angles have been considered for analysis.

Xiangfei ZHAO[4] introduces a novel curve (quadratic rational Bezier curve) to describe the cutter tip. The geometrical shape of gear tooth fillet profile usually cut out by the cutter tip, plays a significant role in the evaluation of the gear bending stresses With the maximum bending stress as the objective function, sub-problem and first order optimization methods in ANSYS were used to optimize the cutter tip. The study reveals that the relation between the design variable and tooth root bending stress is nonlinear, and the gear cut by the optimized cutter exhibits higher bending strength rather than the gear cut by standard cutting tool. The gear tooth profile usually generated by the cutter tip trajectory is the place of maximum stress concentration. If the bending stress is too high, larger module has to be used to decrease the bending stress, but this will increase the tooth size. In order to accelerate the optimization speed, two-dimensional finite element model of the tooth has been established by APDL.

Y. Sandeep kumar [5] studied the effect of tip radius and tooth width and show how the contact stress varies with these parameters. The gear design is optimized based on FE analysis. The stress was calculated using the Lewis equation and then compared with the FE model. The Bending stress in the tooth root and at mating region were examined using 3D FE model. The gear specification is as under. Number of teeth = 20, Module (m) = 4, Pitch circle diameter = 80 mm, Base circle diameter = 70 mm, Pressure angle = 20°, Addendum circle diameter = 88 mm, Circular pitch = 12.56 mm, Thickness of tooth = 6.25 mm, Material of gear is steel having Modulus of elasticity E= 210000 MPa, Poisson's ratio v = 0.3. The tangential load acting at the tooth W_T = 2500N. The optimum results to minimize the stress value while the fillet radius of 3mm and face width of 25mm. The FEA results are found to be in close agreement with the calculated stresses based on AGMA standards and Lewis equation.

Ivana ATANASOVSKA[6] describes the comparative diagrams study of tooth profile selection with aspect of spur gear tooth bending strength. The described procedure implies research of tooth profile parameters influence on spur gear tooth bending strength. The obtained diagrams are used for making the groups of comparative diagrams which enable simultaneously selection of optimal values of addendum modification coefficients and radius of root curvature for a particular gear pair with aspect of tooth bending strength. When teeth flanks have increased hardness by suitable material heat treatment, the requirement for tooth bending strength is the priority aspect for optimization. The described analysis uses Finite Element Method procedure for stress state calculation. The tooth profile parameters that have significant influence at tooth root stresses and tooth bending strength are: addendum modification and radius of root curvature. Selection of discrete value for defined parameters in accordance with valid standard is necessary. The addendum modification coefficient (x_1+x_2 = 0.5), center distance, module and the number of teeth, face width, helix angle, pressure angle and the material are constrained here.

Alexander L. Kapelevich and Yuriy V. Shekhtman [7] introduced direct gear design uses FEA for bending stress evaluation because the Lewis equation and its related coefficient do not provide a reliable solution to the wide variety of non-standard gear tooth profiles that could be considered. The fillet portion of the tooth profile (where maximum bending stress is expected) has equally spaced nodes with higher density than the rest of the tooth profile. The nodes on the involute profiles and the top land are located to have higher density close to the fillets and lower density in the top part of the tooth. Selection of larger number of tooth profile nodes and high node density coefficients provides a more accurate result, but increases calculation time. In most of the cases the node density coefficient of 1.75-2.5 were used. Bending stress balance allows equalizing the tooth strength and durability for the pinion and the gear. The generating rack profiles with 25° and 28° pressure angle provides a much lower level of maximum bending stress compared to the standard 20° generating rack.

M.S. Hebbal [8] describes the possibility of using the stress redistribution techniques by introducing stress relieving features in the stressed zone to the advantage of reduction of root fillet stress in spur gear. In this work, combination of circular and elliptical stress relieving features were used and better results are obtained than using circular stress relieving features. A finite element model with a segment of three teeth is considered for analysis and stress relieving features of various sizes are introduced on gear tooth at various locations. A maximum of 11% reduction in maximum principle stress is obtained in this analysis.

Metin ZEYVELI [10] et al presented the minimization of volume for gear trains. The constraints in the optimization problem were face width, number of teeth, contact stress and bending strength.

Nikhil kotasthane[11] et al presented the optimization of two stage compound gear train. In this work there are two main objectives considered i.e. to minimize the overall weight and maximize the power transmitted. The optimization was carried out in NSGA-II. There are three design variables considered, which are material of gear, module and number of teeth. It is subjected to constraints such as wear strength, minimum module, dynamic loading and tooth strength.

Y.K. Mogal[12] has used genetic algorithm as an optimization tool for optimization of two stage gear train. The objective of this work is to minimize the power loss of worm gear mechanism while satisfying the constraints. The design variables for optimization are number of teeth on gear, helix angle of worm and friction coefficient. The design constraints are linear pressure, deformation and bending strength of teeth. From the result of GA and analytical method, it was concluded that the minimization of power loss gain was 1.361 kW and 0.879 kW respectively. Hence GA is more effective tool for optimization compared to conventional methods.

III. METHODOLOGY OF WORK

The existing gear design has two gears having over design in terms of strength and in two gears the bending stresses higher than the allowable bending stresses of material for expected service life. So for optimize the gear parameters and to make the design safe against the bending and pitting failure mathematical optimization problem is formulated. Then decided the objective function in terms of design variables, range of variables and the constraints for the design. Then optimization is performed in Genetic Algorithm followed by MATLAB programming. Then the models of gear are created in creo according to the optimized design dimension and analysis is carried out in ANSYS workbench.

IV. OPTIMIZATION PROBLEM FORMULATION

The optimization model of three stage helical gear reduction unit is formulated in this section with maximum bending strength as design objective. The bending strength of gear mainly depends on the module, face width, number of teeth of gear, helix angle and pressure angle.

A. Design Vector:

So the design vector X is

$X = \{m_1, b_1, Z_1, Z_2, m_2, b_2, Z_3, Z_4, m_3, b_3, Z_5, Z_6\}$

Where, m = module (mm),
b = face width (mm),
Z = Number of teeth

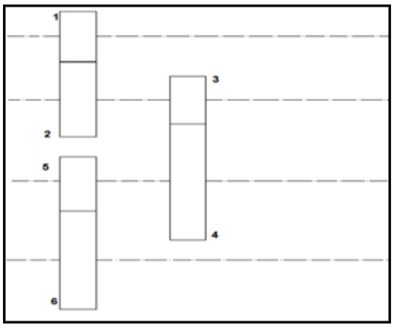

Fig.1: Schematic Diagram of Three Stage Gearbox

B. Objective Function:

The objective function is to minimize bending stress

$$\text{Min } \sigma_b = \frac{W_t K_o K_v K_m K_s K_b}{b\, m\, J}$$

Where,
σ_b = Bending stress (MPa)
W_t = Tangential load (N)
K_O = Overload factor
K_v = Dynamic factor
K_m = Load distribution factor
K_B = Rim thickness factor
K_s = Size factor
J = Geometry factor for bending strength

C. Design Constraints:

1) Contact Stress:

$$\sigma_c = C_p \sqrt{\frac{W_t K_o K_v K_s K_m C_f}{dbI}} \leq \sigma_{all}$$

Where, C_p = Elastic coefficient factor = 191 $\sqrt{MPa}$ for steel material
I = Geometry factor for contact stress

Allowable contact stress $\sigma_{all} = \frac{S_c Z_N C_H}{K_T K_R}$

S_c = Allowable contact stress number (MPa)
Z_N = stress cycle factor for contact stress
C_H = Hardness ratio factor
K_T = Temperature factor
K_R = Reliability factor

2) Minimum number of teeth (Z_{min}):

To avoid interference there must be certain number of teeth on the gear depending upon the pressure angle and helix angle. It is also possible to use lesser number of teeth for the same pressure angle and helix angle, but for that we have to modify the addendum of gear, and this is known as addendum modification or profile shift (x).

$$x = k \times m_n$$

k = addendum modification coefficient
m_n = Normal module of gear

$$Z \geq \frac{2(1-k) - \cos\psi}{\sin^2 \phi_t}$$

Where, ψ = helix angle
ϕ_t = Transverse pressure angle

3) Center Distance (C):

The center distance between the first shaft and the last shaft can be given as follow.

$$C = \frac{d_1+d_2}{2} + \frac{d_3+d_4}{2} + \frac{d_5+d_6}{2} \text{ (mm)} \leq 457.577 \text{ mm}$$

Existing center distance = 457.577 mm

4) Weight of Gear Material (W):

The volume of gear material can be given as follow.

$$w = \left[\frac{\pi}{4}\{(d_1^2 + d_2^2)b_1 + (d_3^2 + d_4^2)b_2 + (d_5^2 + d_6^2)b_3\}\right] \times \rho \text{ (kg)}$$

The volume of gear material should be less than or equal to the existing gear material volume.

V. Material Specification

20MnCr5 (Gear)	20MnCr5 (Gear)
– Case-carburized steel, case-hardened – ISO 6336-5 Figure 9/10 (MQ), core strength >=25HRC – Specific weight (kg/m³)= 7830.000	– Case-carburized steel, case-hardened – ISO 6336-5 Figure 9/10 (MQ), core strength >=30HRC – Specific weight (kg/m³)= 7830.000

VI. Result

A. Existing Design Data:

Input Power (kw) =	3.7
Input Speed	960.000
Output Speed	9.000
Helix Angle Pair 1	13^O
Helix Angle Pair 2	12^O
Helix Angle Pair 3	11^O
Pressure Angle	20^O

Table 1:

B. Comparison of Existing Design and Optimized Design:

Variables and Objectives	Manufacturer Catalogue	By using Genetic Algorithm	After rounding of GA	Percentage Change
m_1	3	2.23	2.25	
b_1	30	23.864	25	
Z_1	13	12	12	
Z_2	59	58.15	59	
m_2	3	3.49	3.5	
b_2	35	39.94	40	
Z_3	15	12	12	
Z_4	85	60.23	60	
m_3	4	5.45	5.5	
b_3	40	55.10	55	
Z_5	18	12	12	
Z_6	75	52.12	52	
Bending Stress (MPa)	1594.47	1229.26	1239.60	Reduce by 22.28%
Center distance (mm)	457.577	402.994	398.541	Reduce by 12.72 %
Gear material Volume (cubic mm)	5890000	5890000	5940000	Increase by 0.97%

Table 2:

C. Comparison of Bending Stress:

	Existing Design	Optimized Design	

Gear Number	Bending stress (MPa)	Bending stress (σ_b) (MPa)	Allowable stress $(\sigma_{b,all})$(MPa)
1	77.64	162.87	402.13
2	78.28	164.48	407.85
3	226.82	183.09	413.11
4	234.64	190.27	420.64
5	472.99	261.26	426.06
6	504.09	277.25	379.09

Table 3:

VII. CONCLUSION

From table – 3 we can conclude that the gear 5 and gear 6 have factor of safety less than 1 in existing design. When in optimized design the bending stresses of all six gear is within permissible limit. Hence the design is safe against the bending fatigue failure and surface fatigue failure.

From table –2, it can be concluded that the overall bending stress in the optimized gear design is reduced by 22.28 % compared to existing design. The center distance between input and output shaft is reduced by 12.72 %. And the material volume increase by 0.97%, hence the weight of gearbox is increase by 0.97%, but it is negligible.

Genetic algorithm is important tool for optimization of complex problem.

VIII. FUTURE WORK

This optimization process can be useful to optimize the gear trains of more multi-stages (other than three).

Particularly for this type of problem, one can use different materials and heat treatment for each of the gear depending upon the torque to be transmitted by each of the gear. By this method one can allot different materials to each gear to satisfy their bending strength requirement. And can analyse the cost and weight of gearbox.

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