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Heavy Duty Helical Gear Pair Volume Optimization Using Genetic Algorithm and ANSYS

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Abstract

Helical gears work quietly and smoothly under heavy loads. Their design optimization includes multiple objectives and large number of design variables. Thus, a reliable and robust optimization technique will be useful to achieve the optimal design solutions. The optimization of helical gear drive can be carried out using different non-conventional optimization techniques like genetic algorithm, particle swarm optimization and ant colony optimization, etc.

In the present work, volume optimization of helical gear drive is carried out using genetic algorithm. Several factors for the strength and size based on DIN standards are considered from the literature. The formulation of multivariable, constrained non-linear optimization problem with derived objective function and constraints is presented. The volume minimization using genetic algorithm is performed in MATLAB and results obtained are found to be satisfactory.This optimum set of parameters is used for modeling of helical gear pair by CREO 4.0 Software Now import the helical gear pair for stress analysis by using ANSYS 16.2 software and find that the stress value is less than the permissible stress of case hardened steel.

KEYWORDS: *Helical gear, genetic algorithm and FEA.*

NOMENCLATURE

σ_H Calculated contact stress, N/mm^2
σ_{Hlim} Allowable contact stress number, N/mm^2
σ_{HG} Modified allowable contact stress number, N/mm^2
σ_{HP} Permissible contact stress, N/mm^2
σ_{Ho} Nominal contact stress, N/mm^2
σ_F Tooth – root stress, N/mm^2
σ_{FE} Allowable bending stress number, N/mm^2
σ_{FP} Permissible tooth root stress, N/mm^2
σ_{FG} Tooth – root stress limit, N/mm^2
ϵ_β Overlap ratio
ϵ_α Transverse contact ratio
K_V Dynamic factor
K_A Application factor
Z_W Work hardening factor
Z_X Size factor (pitting)
Y_X Size factor (tooth – root)
Y_β or Z_β Helix angle factor
m_n Normal module, mm
b Facewidth, mm
Z_1 Number of teeth on pinion
Z_2 Number of teeth on gear
β Helix angle, degrees
ϵ_1 Transverse contact ratio for pinion
ϵ_2 Transverse contact ratio for wheel

1. INTRODUCTION

Helical gears are used in power plant, sugar mills, automobiles, cranes, marine, rolling mills and coal plant as a power transmitting device owing to their relatively silent and smooth operation at higher speed and heavy loads [9]. To design heavily loaded helical gears for transmitting power, that are good in strength and low level in noise necessitates the suitable analytical methods which can be easily used in practice and also provide useful information on bending and contact stresses. Many researchers have attempted the design of these gears using finite element methods [4, 6, and 8]. Linhong et al. [8] analysed the surface of gear tooth for minimum noise under the multiple loads. Padmanabhan et al. [10] used the Genetic Algorithm (GA) and Finite Element Analysis (FEA) for the design of spur gear pair for optimum space requirements for given values of bending and crushing stresses.

In the present work, an attempt has been made for minimizing the volume of heavy duty single stage helical gears. Several factors that affect the gear assembly and working conditions are considered for the constrained volume minimization using GA in MATLAB optimization toolbox.

2. . DESIGN OF HELICAL GEAR

This section presents the design constraints, design objectives and the objective function that minimizes the volume of single stage helical gear pair. The volume minimization is carried out for the following input data [14]:

Transmitted Power = 120 kW; Gear ratio = 5.18; Helix angle = 12^o, Pressure angle = 20^o, Module = 14 (calculated from Lewis equation); Material – Case hardened steel (20MnCr5)

Table 2 shows other design parameters for single stage helical gear pair based on DIN 3960 and 3990 standards [2, 3]. Table 3 & Table 4 given in Appendix, respectively, summarize other geometrical parameters and strength based factors for pinion and gear wheels.

3. Problem Formulation

The following five design variables for volume minimization of helical gear pair are considered:
Module$(m_n \ or \ x_1)$, Face width $(b \ or \ x_2)$, Pinion teeth $(Z_1 \ or \ x_3)$, Gear teeth $(Z_2 \ or \ x_4)$, Helix angle $(\beta \ or \ x_5)$.

3.1 Objective function

The expression for volume of cylindrical gear pair may be expressed as [14]

$$Volume = \frac{\pi}{4}\left(\frac{{m_n}^2 b}{cos^2 \beta}\right)({Z_1}^2 + {Z_2}^2) \qquad (1)$$

And, objective function for the volume minimization of helical gear pair in terms of the design variables as [14]

$$f\ (x) = V = f\ (m_n\ , b, Z_1, Z_2\ , \beta) = f(x_1, x_2, x_3, x_4\ , x_5) = \frac{\pi}{4}\left(\frac{{x_1}^2 x_2}{cos^2 x_5}\right)({x_3}^2 + {x_4}^2) \qquad (2)$$

3.2 Constraints

The different strength considerations such as safety factor for tooth-root stresses,safety factor for pitting, transverse contact ratio, etc. and geometric constraints such as face width, number of teeth, helix angle, centre distance, etc. are calculated [14] and summarized in Tables 2, 3 & 4.

Safety factor for tooth-root stress

Bending strength of gear teeth is related to the resistance to cracking at the critical section in internal gears and at the tooth-root fillet in external gears [7]. The tooth-root stress or tooth breakage considers the factors like life, size, relative toughness, reliability, dynamic load, type of application (load), helix angle and distribution of load for bending stress, etc.

The transverse contact ratio is expressed as [5]

$$\epsilon_\alpha = \frac{1}{2\pi}[z_1(\tan\theta_1 - \tan\ \alpha'_t) - z_2\ (\tan\theta_2 - \tan\ \alpha'_t)] \qquad (3)$$

For the present problem, the expressions for transverse contact ratio for pinion and gear wheels, respectively, may be expressed as [5]

$$\epsilon_{\alpha 1} = \frac{x_3\left|\tan\left\{cos^{-1}\left(\frac{d_{b1}}{d_{a1}}\right)\right\} - \tan\left[cos^{-1}\left[\frac{\frac{x_1}{cos\ x_5}(x_3+x_4)}{2a'/cos\alpha_t}\right]\right]\right|}{2\pi}$$

$$= \frac{x_3\left[0.681 - \tan\left[\cos^{-1}\left[\frac{\frac{x_1}{\cos x_5}(x_3+x_4)}{595.94}\right]\right]\right]}{2\pi} \quad (4)$$

$$\epsilon_{\alpha 2} = \frac{x_4\left[\tan\left\{\cos^{-1}\left(\frac{d_{b2}}{d_{a2}}\right)\right\} - \tan\left[\cos^{-1}\left[\frac{\frac{x_1}{\cos x_5}(x_3+x_4)}{2a'/\cos\alpha_t}\right]\right]\right]}{2\pi} = \frac{x_4\left[0.439 - \tan\left[\cos^{-1}\left[\frac{\frac{x_1}{\cos x_5}(x_3+x_4)}{595.94}\right]\right]\right]}{2\pi} \quad (5)$$

The tooth-root stress constraints for pinion and gear wheels, respectively, may be expressed as [14]

$$g_1(x) = 1.4 - \left[\frac{435.43}{\frac{280850}{x_1 x_2}\times 1.818\times\left(0.25+\frac{0.75}{\epsilon_{\alpha 1}+\epsilon_{\alpha 2}}\right)\times\left(1-\frac{x_2 \sin x_5}{\pi x_1}\times\frac{x_5}{120}\right)\times 1.3013}\right] \leq 0 \quad (6)$$

$$g_2(x) = 1.4 - \left[\frac{435.43}{\frac{280850}{x_1 x_2}\times 1.818\times\left(0.25+\frac{0.75}{\epsilon_{\alpha 1}+\epsilon_{\alpha 2}}\right)\times\left(1-\frac{x_2 \sin x_5}{\pi x_1}\times\frac{x_5}{120}\right)\times 1.255}\right] \leq 0 \quad (7)$$

Safety factor for surface durability

Surface durability indicates value of maximum tangential force that the tooth can transmit power without pitting failure [7]. The safety factor constraints for pinion and gear wheel derived from the elasticity, Poisson's ratio, transverse contact ratio and contact stresses [14], respectively, may be expressed as

$$g_3(x) = 1.2 - \left[\frac{1261.82}{189.81\times\sqrt{\frac{1}{\epsilon_{\alpha 1}+\epsilon_{\alpha 2}}}\times\sqrt{\cos x_5}\times 38.251\times\sqrt{\frac{1}{x_2}}\times 1.1407}\right] \leq 0 \quad (8)$$

$$g_4(x) = 1.2 - \left[\frac{1262.44}{189.81\times\sqrt{\frac{1}{\epsilon_{\alpha 1}+\epsilon_{\alpha 2}}}\times\sqrt{\cos x_5}\times 38.251\times\sqrt{\frac{1}{x_2}}\times 1.1202}\right] \leq 0 \quad (9)$$

Centre distance

The centre distance constraints between the pair of wheels may be expressed as [5],

$$g_5(x) = 2934.37 - \left(\frac{x_1}{2\cos x_5}\right)(x_3 + x_4) \leq 0 \quad (10)$$

Number of teeth

In order to avoid the interference between helical gear pinion and wheel, the tooth constraints may be expressed as [14]

$$g_6(x) = 25 - x_3 \leq 0 \quad (11)$$

$$g_7(x) = x_3 - 56 \leq 0 \quad (12)$$

$$g_8(x) = 130 - x_4 \leq 0 \quad (13)$$

$$g_9(x) = x_4 - 290 \leq 0 \quad (14)$$

Face width

The constraints for minimum and maximum face width may be expressed as [14]

$$g_{10}(x) = 50 - x_2 \leq 0, \quad (15)$$

$$g_{11}(x) = x_2 - 250 \leq 0 \quad (16)$$

Helix angle

The minimum (4° to increase the load capacity) and maximum (for controlled vibrations and axial load) helix angle constraints may be expressed as [14]

$$g_{12}(x) = 4 - x_5 \leq 0, \quad (17)$$

$$g_{13}(x) = x_5 - 19.5 \leq 0 \quad (18)$$

4. VOLUME OPTIMIZATION USING GENETIC ALGORITHM

GA, an adaptive heuristic search algorithm, depends on the evolutionary concepts of natural selection and genetics [9]. This technique is inspired by Darwin's theory about evolution- *Survival of the fittest* [3]. GA repeatedly modifies the population of individual solutions. At every step, the GA chooses individuals at random from the current population (parents) and uses them to produce children for the next generation [13]. Over the successive generations, population "evolves" towards an optimal solution [10]. This evolutionary method has been used to solve the present volume minimization problem. Fig.1 shows the steps for optimization using Genetic Algorithm.

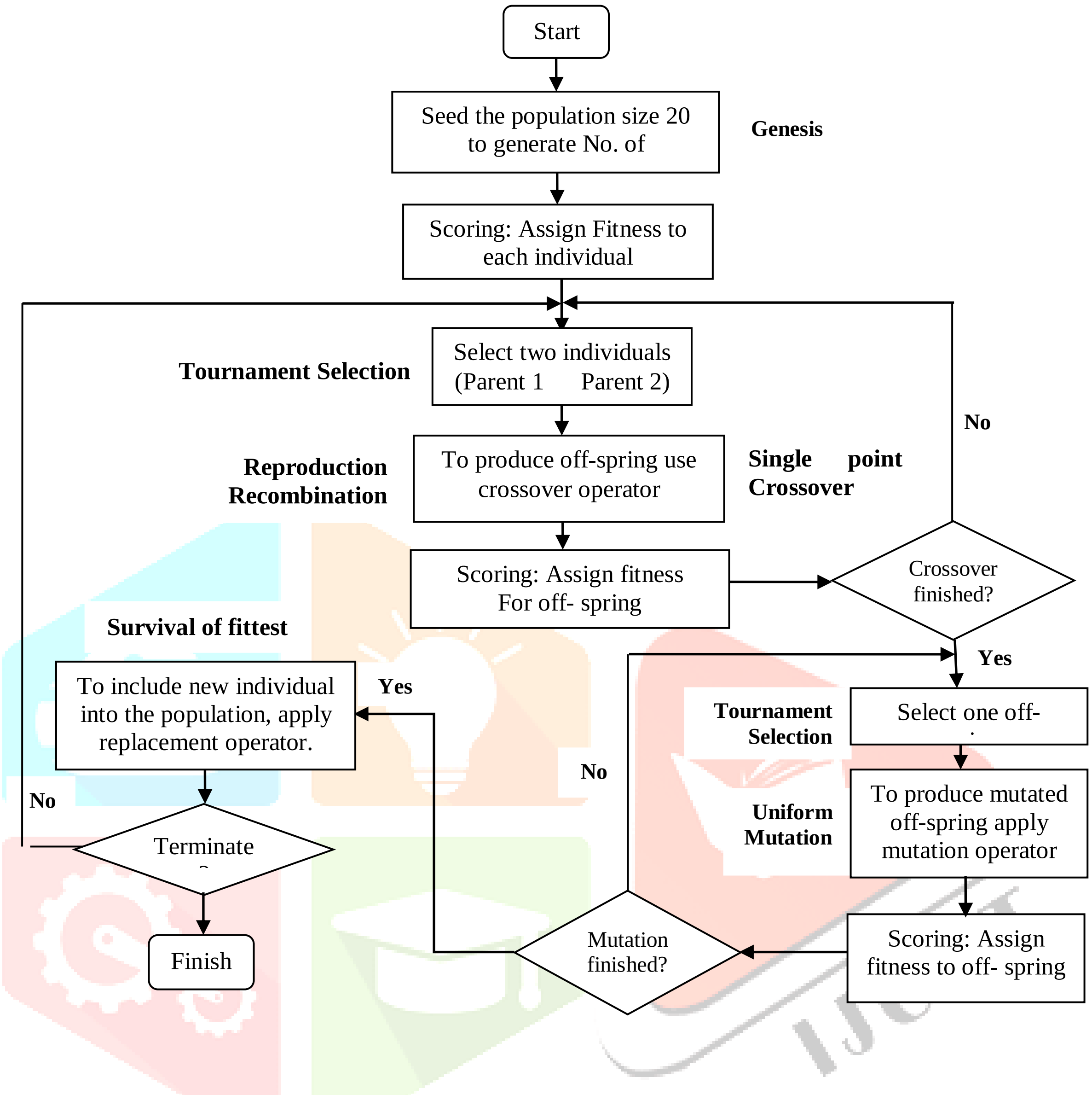

Fig.1 Steps for optimization using Genetic Algorithm [10]

4.1 Design variables

At the beginning, the specified population is developed randomly. The control string of design variables is characterized as [10]

$X = [m_n , b, Z_1, Z_2, \beta]$ (15)

The design parameters within the specified limits are characterized as discrete parameters.

4.2 Tournament selection

The major objective of selection operator is to indicate the good solution and remove the worst solution in a population while maintaining its size [3]. The tournament selection is carried out by creating, tournament competition among the individual strings. The best individual (winner) from this group is selected as the parent. This process is repeated until the mating pool for new off spring is filled [13].

4.3 Crossover operator

The crossover operator is used to create new solution from the existing solutions available in the mating pool after applying the selection operator [10]. This operator exchanges the gene information between the solutions in mating pool. A Single-point crossover point is selected, binary string from beginning ofchromosome to the crossover point is copied from one parent, and the rest is copied from the second parent [13].

4.4Mutation

Mutation is the process in which new features into the strings solution of population pool is introduced to maintain dissimilarity in the population [13]. This operator is used to replace the value of selected gene with uniform random between the upper and lower bounds specified by the user for that gene [10]. This operator can only be used for integer and float genes.
Let a_i and b_i be the lower and upper bound, respectively for variable i.Uniform mutation selects only one variable, j, and sets equal to an uniform random number $U(a_i, b_i)$:

$$x'_i = \begin{cases} U(a_i, b_i), & if\ i = j \\ x_{i,} & otherwise \end{cases}$$

5. STRESS ANALYSIS

Bending stresses and contact stresses are considered as vital role to achieve an efficient, compact and high power transmitting helical gear pair.

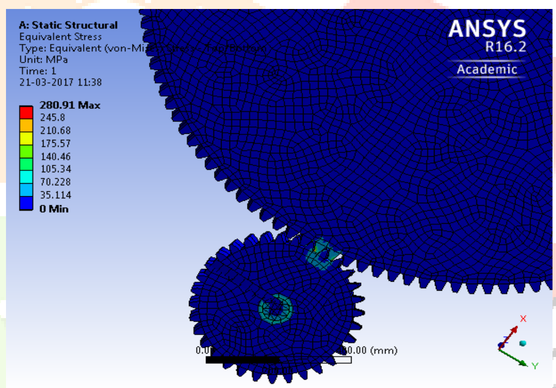

Fig. 2 Stress analysis

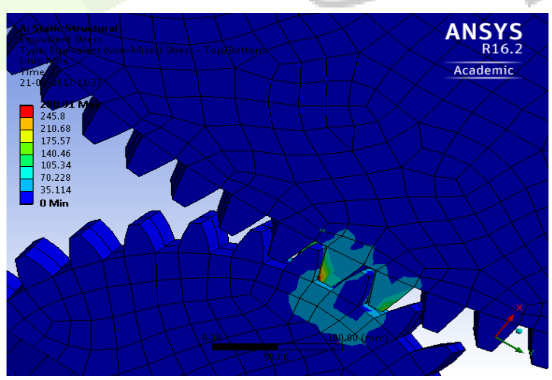

Fig.3 Maximum Stress Region

The Von-mises stress is calculated on the basis of distortion energy theory of failure. However the designs are carried out for bending stress only. Hence the results from software are differing than the allowable bending stress i.e.430 MPa.

Fig.2 shows that the equivalent stress from analysis is 280.91MPa which is less than the value of maximum permissible bending stress of material, 430 MPa. So, from safety point of view the optimization result is acceptable.

6. RESULTS AND DISCUSSIONS

The mathematical models for test problem have been formulated in terms of the five design parameters m_n, b, Z_1, Z_2 and β. Initially, input values are generated randomly with their parameter bounds. If the generated values satisfy the design constraints, the objective function is calculated. The following input is used for GA:

INPUT DATA-

Population size: 20, Tournament size: 4, Elite: 2, Crossover probability: 0.6, Mutation probability: 0.01, Generation: 50, Tolerance: 10^{-5}

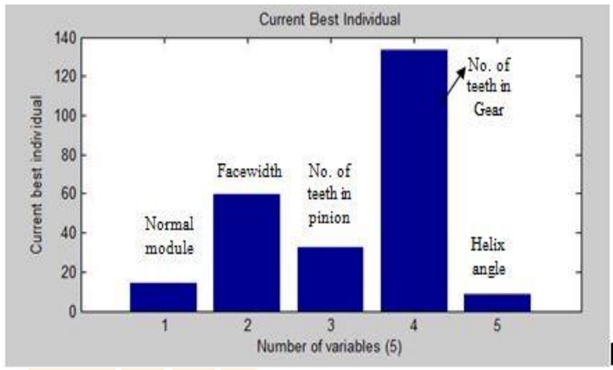

Fig.4 Optimum set of design parameters

Table 1 Result comparison- Optimum design parameters using Genetic Algorithm

Tool / Design Parameters	Genetic Algorithm	Particle swarm optimization [14]	Improvement in Volume (%)
Face width, mm	**59**	53	**2.6%**
No. of teeth on pinion	**32**	30	
No. of teeth on gear	**133**	142	
Helix angle, degree	**8.5**	11.9	
Volume, mm^3	**$1.75748 * 10^8$**	$1.8046 * 10^8$	

Table 1 shows the set of optimum design parameters using GA and results are compared with the same obtained from particle swarm optimization technique [14].

5. CONCLUSIONS

An attempt has been made to achieve minimum volume design solution for a helical gear pair using GA. Bending stresses and contact stresses are considered as vital constraints to achieve an efficient, compact and high power transmitting helical gear pair. The results show an improvement in 2.6% volume (alternatively, weight) of helical gear pair as compared to the same obtained using particle swarm optimization technique [14].

However the designs are carried out for bending stress only.The equivalent stress from analysis is 280.91MPa which is less than the value of maximum permissible bending stress of material, 430 MPa. So, from safety point of view the optimization result is acceptable.

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Appendix

The expressions for safety factor, permissible bending and contact stresses are given below [5]:

Safety factor for tooth root stress, $S_F = \left(\frac{\sigma_{FG}}{\sigma_F}\right) \geq S_{Fmin}$

Where, $N_L = 3 \times 10^6$ load cycles

Permissible bending stress,

$$\sigma_{FP} = 0.92\, \sigma_{FPref} \left(\frac{10^{10}}{N_L}\right)^{0.01} = \left(\frac{\sigma_{FG}}{S_{Fmin}}\right)$$

Permissible tooth root stress for long life, $\sigma_{FPref} = \left(\frac{\sigma_{FE}}{S_{Fmin}}\right) Y_{\delta refT}\, Y_{RrefT} Y_X$

Tooth - root stress, $\sigma_F = \left(\frac{F_t}{b.m_n}\right) Y_F Y_S Y_\beta K_A K_V K_{F\alpha} K_{F\beta} \leq \sigma_{FP}$

Safety factor for surface durability, $S_H = \left(\frac{\sigma_{HG}}{\sigma_H}\right) \geq S_{Hmin}$

Permissible contact stress, $\sigma_{HP} = 0.92\, \sigma_{HPref} \left(\frac{10^{10}}{N_L}\right)^{0.0157} = \left(\frac{\sigma_{HG}}{S_{Hmin}}\right)$

Where, $N_L = 5 \times 10^7$ load cycles

Permissible contact stress for long life, $\sigma_{HPref} = \left(\frac{\sigma_{Hlim}}{S_{Hmin}}\right) Z_I\, Z_R\, Z_V\, Z_X$

Contact stress, $\sigma_H = \sigma_{Ho} Z_{BorD} \sqrt{K_A K_V K_{H\alpha} K_{H\beta}} \leq \sigma_{HP}$

Table 2 Other design parameters for pinion and gear [14, 7]

m_n	Normal module, mm	14	m_t	Transverse module, mm	14
P	Power, kW	120	α	Pressure angle, degrees	20^0
β	Helix angle, degrees	12^0	r	ratio	5.18
F_t	Tangential Force, N	280850	F_b	Permissible bending stress, N/mm^2	430
Y_V	Lewis form factor	0.308	K_A	Application factor	1.0
α_{wt}	Working pressure angle, degrees	20.32^0	P_t	Transverse circular pitch, mm	44.96
P_n	Normal circular pitch, mm	43.97	β_b	Base helix angle, degrees	11.269
X	Sum of addendum shift coefficient	0.1093	C_α	Working centre distance, mm	710
K	Addendum modification factor	1.0	S_t	Transverse tooth thickness, mm	32.89
h	Whole depth, mm	33.587	ϵ_α	Transverse contact ratio	1.578
ϵ_β	Overlap ratio	1.1587	K_V	Dynamic load factor	1.001
Z_E	Elasticity factor for contact stress	189.81	Z_H	Zone factor for contact stress	2.429
Z_ϵ	Contact ratio for contact stress	0.796	Z_β	Helix angle factor for contact stress	0.989
Z_L	Life factor for contact stress	1.0	Z_I	Lubrication factor for contact stress	0.92
Z_W	Work hardening factor for contact	1.0	V_m	Mean velocity, m/s	0.44
Z_R	Surface roughness factor for contact	0.9	$K_{H\alpha}$	Transverse load distribution factor	1.0
Y_F	Tooth profile factor for bending	1.9	Y_β	Helix angle factor for bending	0.9

Table 3 Geometrical parameters calculated for pinion and gear [14, 5]

Symbol	Geometrical parameters	Pinion	Gear
Z_V	Virtual number of teeth	17.007	88.222
d	Nominal pitch circle diameter, mm	229	1187
D_W	Operating pitch circle diameter, mm	229.495	1190.505
Z	Number of teeth	16	83
d_b	Base diameter, mm	214.627	1113.397
d_t	Tip circle diameter, mm	259.78	1261.19
h_a	Addendum, mm	15.388	14.118
h_d	Dedendum, mm	18.20	19.470
S_n	Normal tooth thickness, mm	23.01	22.085

Table 4 Strength based factors for pinion and gear [14]

Symbol	Strength based factors	Pinion	Gear
$K_{H\alpha}, K_{\alpha\beta}$	Load distribution factors	1.3	1.254
K_R	Reliability factor	1.015	1.014
Y_ϵ	Contact ratio factor for bending	2.74	2.79
Y_R	Relative roughness factor	0.957	0.957