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Effect of Various Design Factors on the Bending Stress of a Close Conversion Gear with Cycloidal Teeth

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ABSTRACT:

Due to the significant mechanical power transmission, gears have been subjected to many studies to prevent breakdowns, reduce noise and vibration, and introduce improvements based on strength and density. This work focuses on its ability to carry the resulting flexible loads with a straight bevel gear using a cycloidal profile instead of involution. The main objective is to investigate how modulus, taper angle and roll radius affect bending stress. 3D models are characterized mathematically as a collection of points representing a cycloidal profile, which are entered into the FE solution at the beginning of the generation process. The bending strength is determined by the von-Misses stress. The results show that modulus plays a crucial role in minimizing bending stress; the higher the modulus, the lower the flexural stress value. Furthermore, the results show that the bending stress can be effectively controlled by reducing the roll radius, while increasing the taper angle decreases the bending force .

KEYWORDS: Spur bevel gears; Cycloid profile; Fillet tension

INTRODUCTION

The demands of modern man, together with rapid and continuous development in many areas of life, have increased the requirements of traditional technical rules and principles, especially in the field of transportation, which is known for its high speed and tranquility. performance mechanically. perspective and dependence on mass transfer [1]. In practice, the most reliable mechanical component for transmission of mechanical power is the gearbox, which can be of several types depending on factors and transmission needs. Bevel gear, a conical workpiece with teeth whose teeth taper on the surface of a pitch cone in its cross-sectional area, is an important component in the transmission of power between many shafts in different directions [2]. Researchers studied how the number of teeth, modulus and taper angle affect bending and contact stresses, vibration, noise, efficiency and power loss. They then introduced various geometric changes in the tooth profile, such as pressure angle, profile adjustment, rounded tips, etc. to get closer to the ideal transmission design. In this work, cycloidal tooth profiles are used in fits, and the maximum bending stress is investigated using the finite element method with respect to geometric factors. The kinematics of the relative motion between the forming roll and the generated gear blank is used to trace the epicycloidal and hypocycloidal parts of the tooth profile before introducing a narrow angle to reflect the intended gear geometry. In order to obtain reliable results and to obtain a clear research idea and appropriate approach, the tooth geometry was validated and evaluated using FEM to ensure the correctness of the overall research assumption and idealization as well as the mathematical model. OBJECTIVE AND METHODS This work uses a cycloidal tooth profile instead of developments and investigates the effect of tooth shape on bending stress in different scenarios of combinations of geometric parameters to evaluate the effect of each parameter on bevelling strength. Based on the geometry generation process of cycloidal curve and straight bevel gear, the principle of coordinate transformation and transfer was used to simulate the cutting process and achieve an accurate cycloidal bevel tooth profile in a rectangular coordinate system. The 3-d models were entered into the FE solution to estimate the maximum static bending stress under worst-case loading conditions. The effect of each parameter is explained in light of the FEM results.

BEVEL GEAR

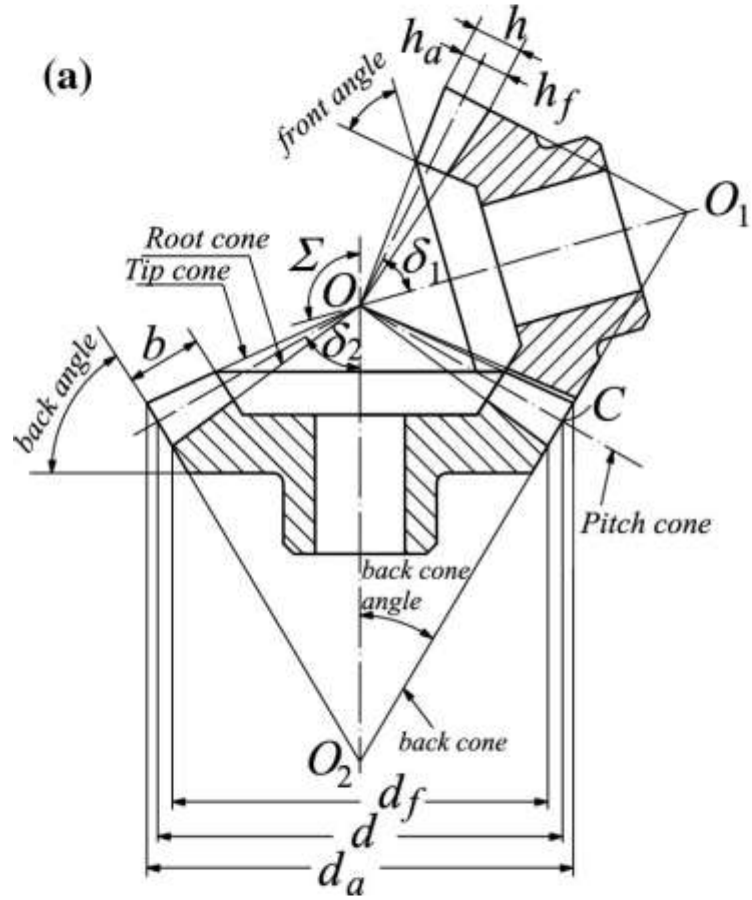


Figure 1. General layout of bevel gear drive geometry [5].

Since its original shape appearance, spur gear still the most used element among other gear types as the prime power mover with huge variety of proposed geometrical design parameters. The only disadvantage of spur gear, like other cylindrical gears, is their inability to transmit power between any shafts other than parallel shafts [3]. Bevel gears have been arisen to solve such problem and change the transmission direction to any desired angle. Straight bevel gears are analogous to the spur gear but with conical shape pitch surface instead of the cylindrical pitch one. The transmission shafts are intersecting and their teeth work under radial, tangential and axial force components. Due to the change in teeth cross sectional area along their face width, module of gear is variable from its minimum value at toe to the larger value at heel section but the larger value of module is assigned as a characterizing parameter [4]. Figure 1 shows the general layout of bevel gear drive geometry and its terminology [5]. The most important geometrical design parameters that control bevel gear shape are: module, teeth numbers, cone angle, face width and fillet radius. Generally the most common used bevel gears are of involute teeth profile which has a lot of advantages such as easiness of manufacturing, center distance error allowance [6].

GENERATION

In cycloid spur gears, tooth profile is generated by the locus of point lay on a circle rolls without slip on the spur pitch circle, in case of bevel gears the generating or rolling circle is replaced by a cone rolls on the bevel gear pitch cone without slip as shown in Figure 2. The most important point is that the ratio of the rolling circle radius to the pitch circle radius at the tooth toe is the same as that at the tooth heel, consequently each tooth section must has different module value.

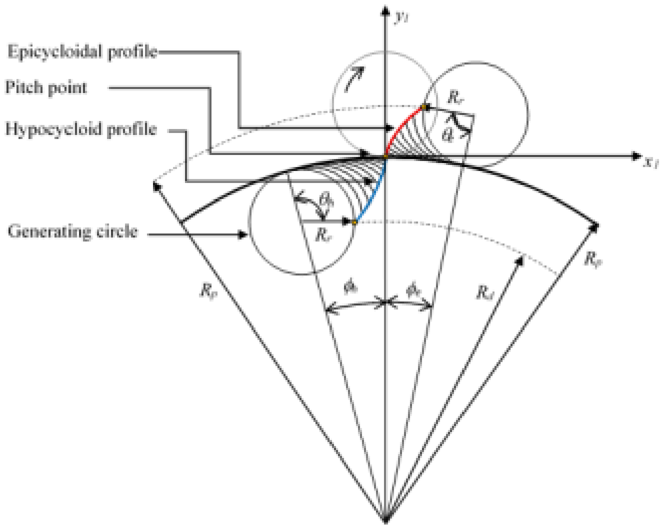


Figure 2. Generation of Cycloid profile

To simplify the generation process toe and heel will be generated separately and gear model will be no more than straight lines connecting toe and heel sections, taking into account that teeth are perpendicular to the pitch cone where;

$$x_{1\mathrm{e}} = R_r \, sin \, \theta_e \tag{1}$$

$$y_{1\mathrm{e}} = -R_r \, cos \, \theta_e \tag{2}$$

$$z_{1\mathrm{e}} = 0 \tag{3}$$

$$x_{2\mathrm{e}} = x_{1\mathrm{e}} \tag{4}$$

$$y_{2\mathrm{e}} = y_{1\mathrm{e}} cos\delta \tag{5}$$

$$z_{2\mathrm{e}} = z_{1\mathrm{e}} + y_{1\mathrm{e}} \tag{6}$$

$$x_{\mathrm{e}} = x_{2e} - (R_p + R_r) \tag{7}$$

$$y_{\mathrm{e}} = y_{2\mathrm{e}} + (R_p + R_r) \tag{8}$$

$$z_{\mathrm{e}} = z_{2\mathrm{e}} - R_p / \tan\delta \tag{9}$$

Equations (7), (8) and (9) represent the loaded side epicycloidal tooth profile while the hypocycloid al part of the same side is described as follows:

$$x_{1h} = -R_r sin\theta_h \tag{10}$$

$$y_{1\mathrm{h}} = R_r \cos\theta_h \tag{11}$$

$$z_{1h} = 0 \tag{12}$$

$$x_{2h} = x_{1h} \tag{13}$$

$$y_{2h} = y_{1h} cos\delta \tag{14}$$

$$z_{2h} = y_{1h} sin\delta \tag{15}$$

$$x_H = x_{2h} + (R_p - R_r)_h \tag{16}$$

$$y_h = y_{2h} + (R_p - r_r)_h \tag{17}$$

$$z_h = z_{2h} - R_p/tan\delta \tag{18}$$

where

$$\boldsymbol{\phi} = \frac{\boldsymbol{rr}}{\boldsymbol{R_p}}\,\boldsymbol{\theta} \tag{19}$$

$$\boldsymbol{R_p} = \mathbf{0.5}\,\boldsymbol{m}\,\boldsymbol{N} \tag{20}$$

The most important condition to specify the maximum value of θ for epicycloidal and hypocycloid profiles i.e. to stop the generation process at the tooth tip and tooth root, is:

$$\sqrt{\boldsymbol{x}_e^2 + \boldsymbol{y}_e^2} = \boldsymbol{R}_a \tag{21}$$

For epicycloidal profile while for hypocycloid will be:

$$\sqrt{x_h^2 + y_h^2} = R_d \tag{22}$$

$$R_a = R_p + h_a \tag{23}$$

$$R_d = R_p - h_d \tag{24}$$

$$h_a = m \tag{25}$$

$$h_d = 1.25\,m \tag{26}$$

Figure 3 illustrates the general layout of bevel gear drive and coordinate transformation. Roller radius must be greater than the module value for epicycloidal part and more than 1.25 times module value for hypocycloid profile to generate full depth teeth.

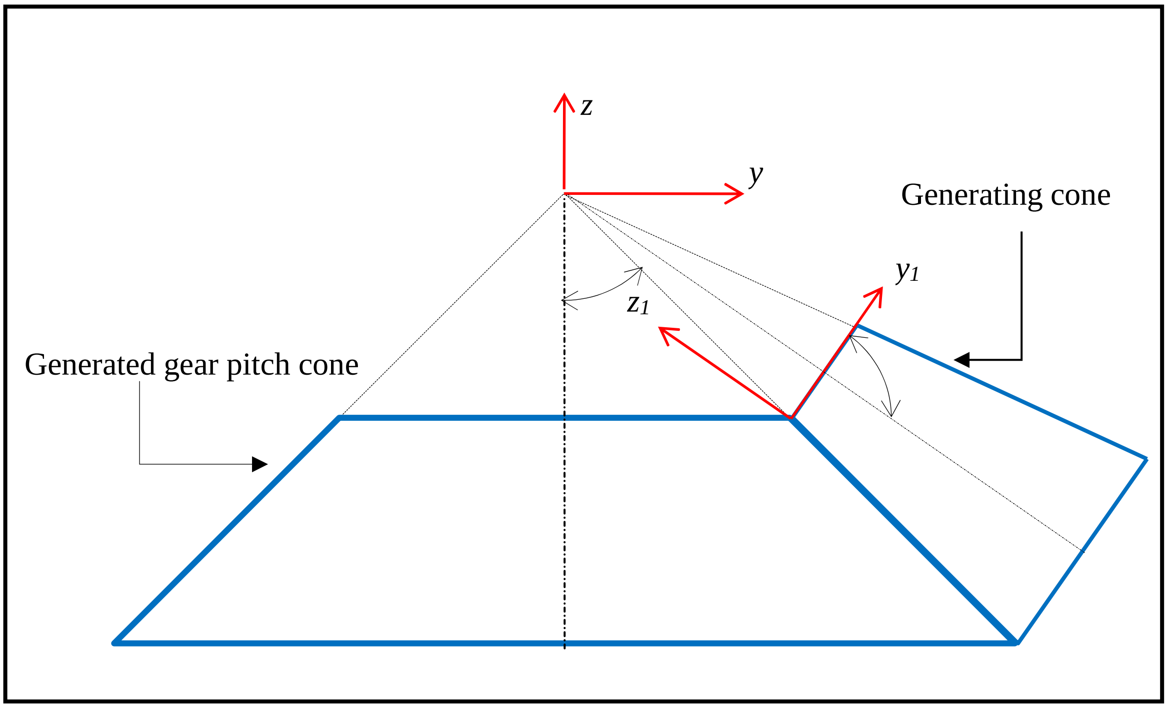


Figure 3. Coordinate transformation and shifting

VERIFICATION OF TEETH GEOMETRY

To ensure the validity of the proposed model, it will be compared by any available works that approach any of its special or general forms. Till the preparation of the present work, cycloid teeth- straight bevel gear hasn't been proposed yet so that the toe and heel sections will be compared with a cycloid spur of the same geometrical specifications. Table (1) shows tooth thickness at the root section for different roller radii compared with those in reference (7) for the same geometrical parameters. The main geometrical characteristics to be compared are tooth tip thickness (tt) and base thickness (tb).

Table 1. Tooth thickness in mm at the addendum circle (tip width) for cycloid spur gear teeth with m=2 mm, Z=14 and different roller radius (Rr).

Rr / Rp	Rr (mm)	Tooth thickness (mm), Ref. [7]		Tooth thickness (mm), Present work	
		Tip	Base	Tip	Base
0.2	2.8	1.152	4.0328	1.152	3.98
0.275	3.85	1.4382	3.5253	1.4382	3.49
0.3	4.2	1.4998	3.3979	1.49	3.35
0.4	5.6	1.666	2.9806	1.66	2.98
0.5	7	1.763	2.6278	1.76	2.687

FINITE ELEMENT METHOD (FEM)

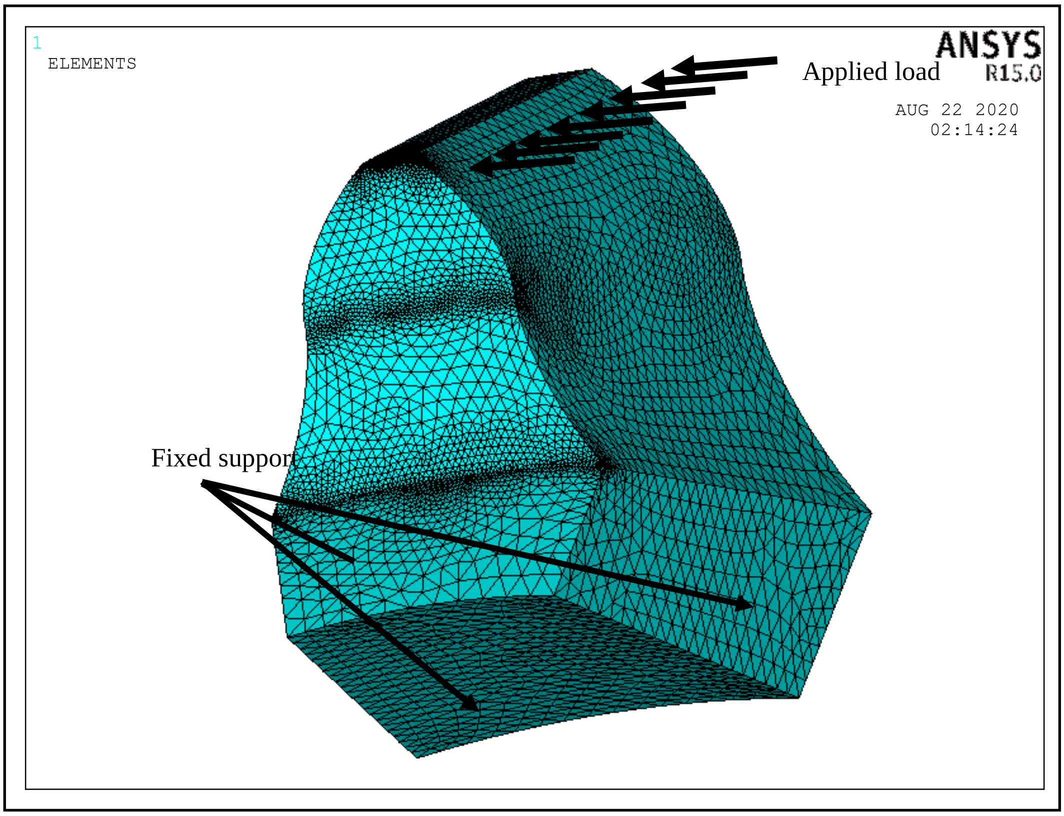


Figure 4. Descretization of single tooth model

Finite element method is an alternative tool uses numerical techniques to simulate and solve different mathematical and physical problems with unique advantages regarding the high accuracy, easiness and low cost use, avoidance of studied models manufacturing errors, and availability [8]. This work adopts Ansys Mechanical APDL as FE solver to model the bevel gear geometry and analyze the induced bending stresses. Starting from the generation equations the toe and heel profiles are feed to Ansys APDL as key points each has its (x, y, z) coordinates. These points are connected using spline passes through each point from tooth root to tip for loaded side only which is in turn reflected on (YZ) plane to generate the unloaded side tooth parts. Toe and heel sides are connected by straight lines, the final tooth embodiment is done by converting the edges lines to a surface area which is the base of volume creation. The mechanical properties are 200 GPa for the modulus of elasticity and Poisson's ratio is 0.3 and it has been chosen as a homogeneous linear structural material. A single tooth is investigated under worst loading condition i.e. one tooth transmit the whole applied load alone at tip engagement position while it is fixed at it three rim surfaces. Models were meshed using 20 nodes solid brick element as shown in Figure 4, which has shown a great constancy regarding the convergence test [9]. Results have been extracted using Ansys workbench ver. 15.

Table 2. Bevel gear physical properties and geometrical variable values

Modulus of elasticity (Gpa)	Poisson's ratio	Teeth number	Face width (mm)	Teeth height (mm)
200	0.3	14	14	15.75

RESULTS AND DISCUSSIONS

From the numerical results it is clear that there were some noticeable findings that must be highlighted and interpreted these are:

Increasing cone angle has a negative effect on the bending stress in case of constant tooth face width. Results show that increasing the cone angle from 25 to 55 could maximize the bending stress by about 80%, such weakness could be attributed to the reduction in the projected tooth root area as shown in Figure 6 and 9. The most important result is that increasing module value leads to great reduction in bending stress. It has been found that changing module value from 3 mm to 8 mm could reduce the bending stress by about 75% as shown in Figure 7 and 10, such effect could be attributed the ability of module to magnify the size of gear teeth which in turn increase the bending strength. It has been found that the value of the maximum bending stress have a direct relationship with roller radius, decreasing the roller radius reduces bending stress by about 25% in case of changing roller radius from 14 mm to 7 mm as shown in Figure 8 and 11, the cause behind this is, teeth roots cross sectional area widening under the use of smaller roller circle.

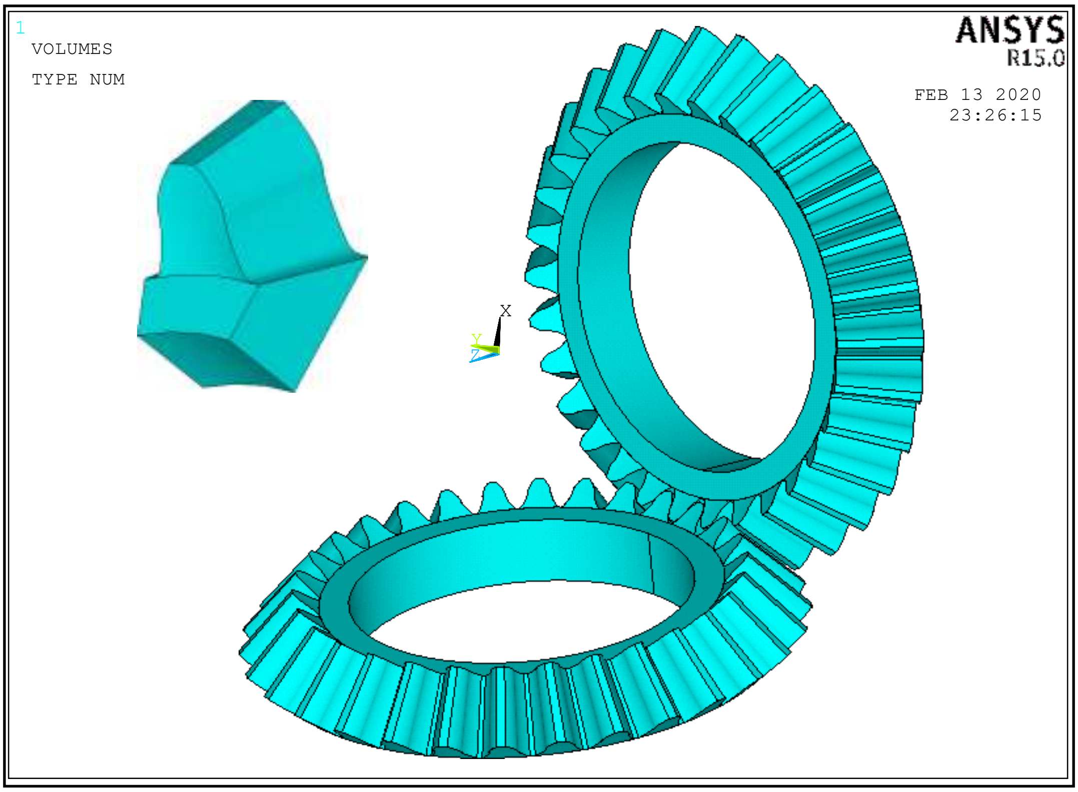
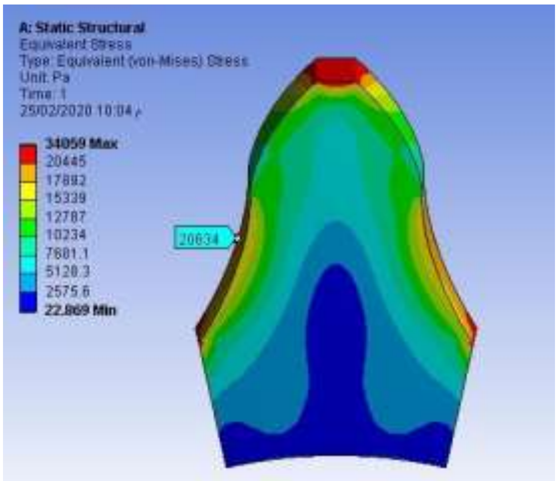
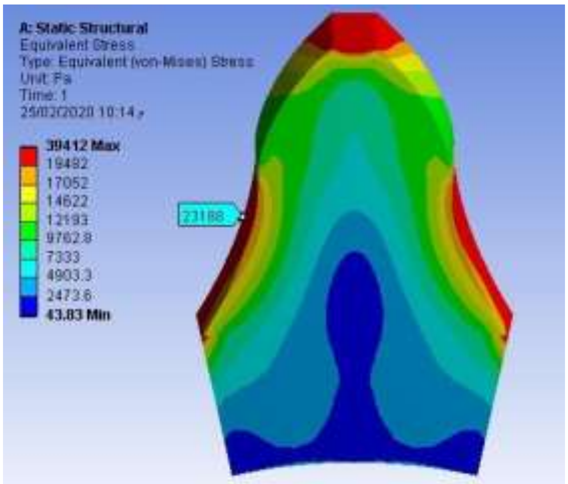
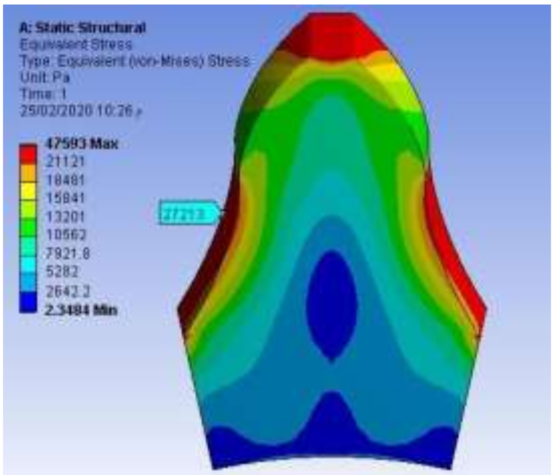


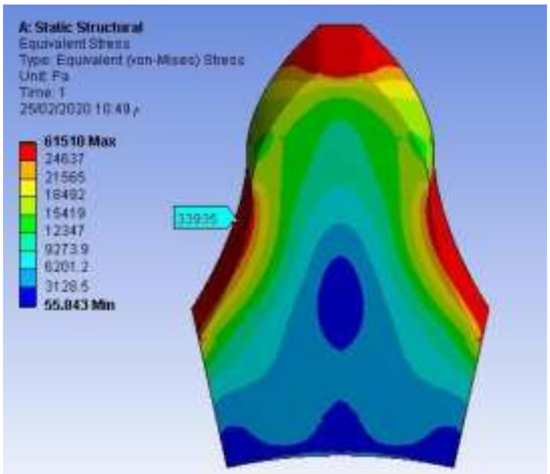
Figure 5. Generated 3- D bevel gear model.





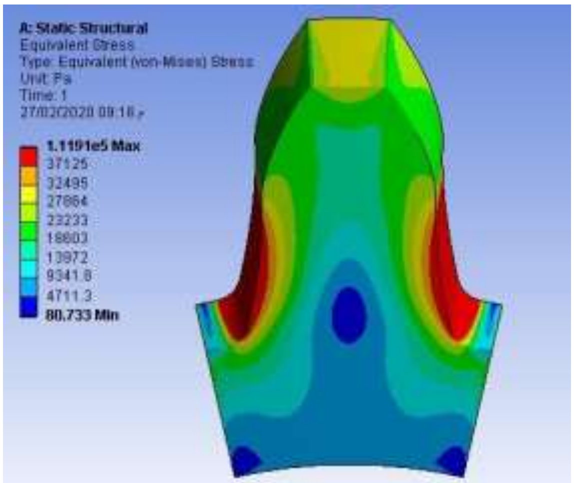
(a) (b)

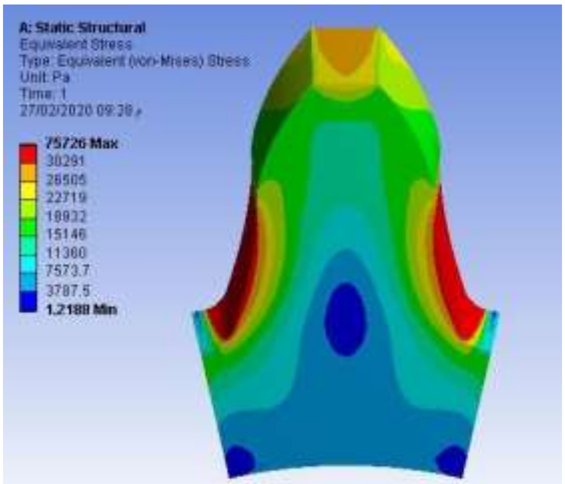




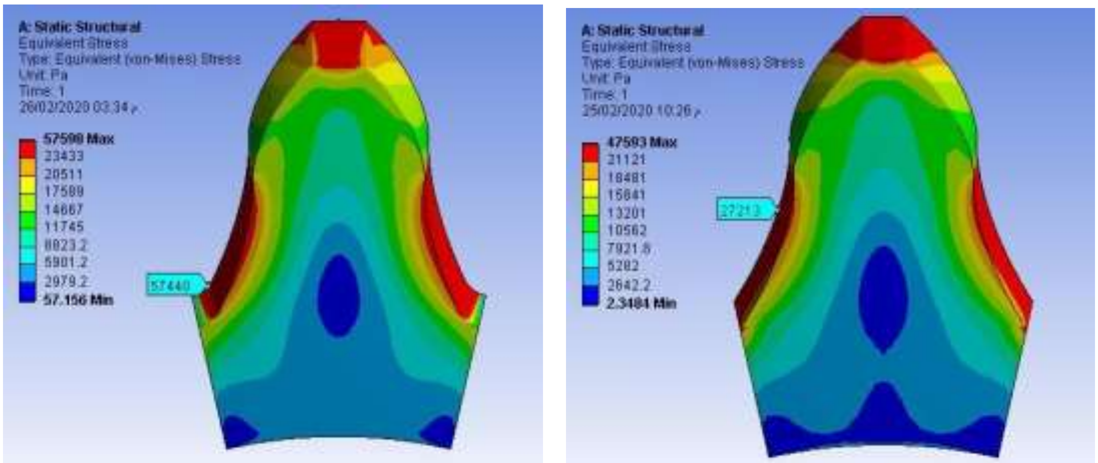
(c) (d)

Figure 6. FEM bending stress for cycloid bevel gear teeth of N=14 teeth, m=7 mm, Rr=7 mm, b= 3mm, a) δ=25o , b) δ=35o, c) δ=45o, d) δ=55o



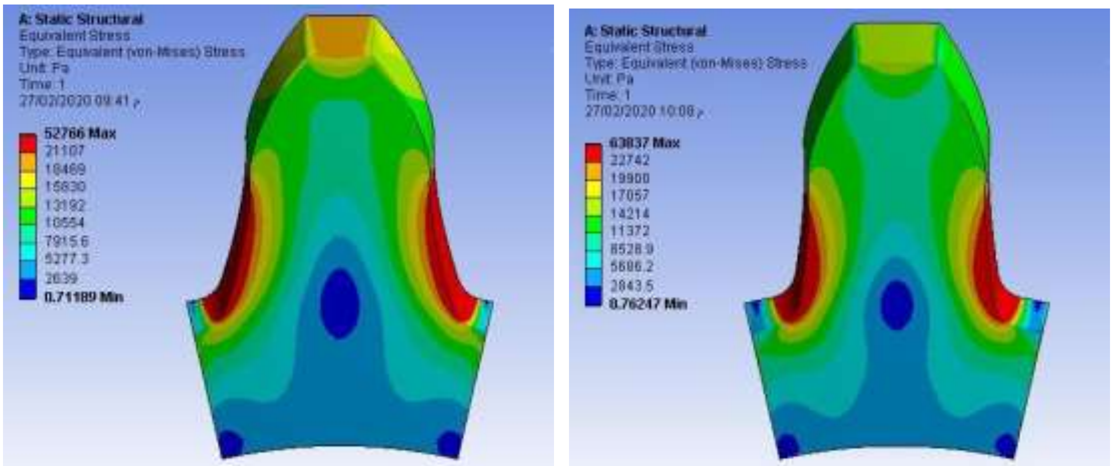


(a) (b)



(c) (d)

Figure 7. FEM bending stress for cycloid bevel gear teeth of N=14 teeth, Rr=7 mm, b= 3mm, δ=45o a) m= 4 mm, b) m=5 mm, c) m=6 mm, d) m=7mm



(a) (b)

Figure 8. FEM bending stress for cycloid bevel gear teeth of N=14 teeth, m=7 mm, b= 3mm, δ=45o , a) Rr= 10 mm, b) Rr=14 mm

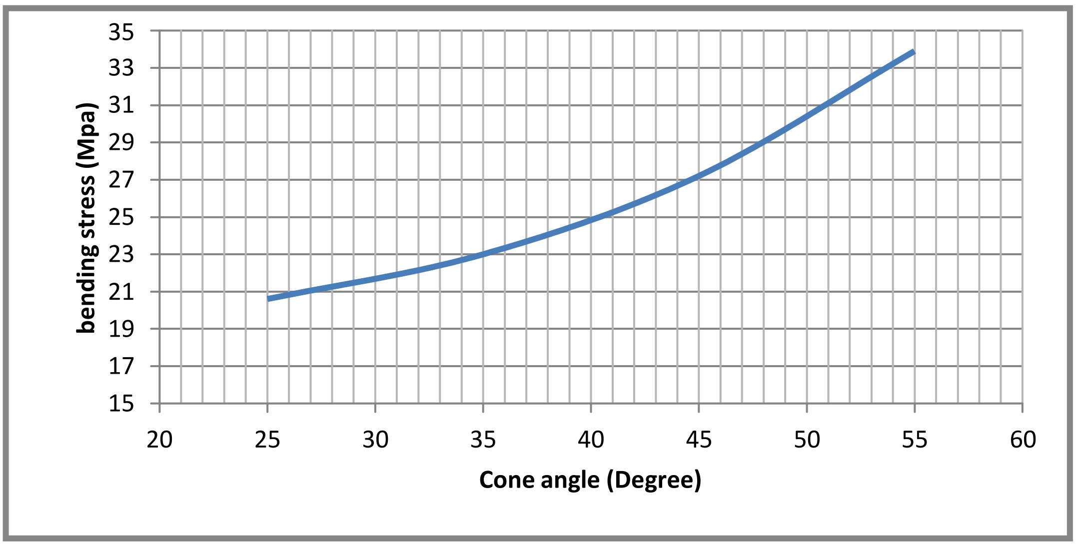


Figure 9. Variation of maximum bending stress with the variation of con angle

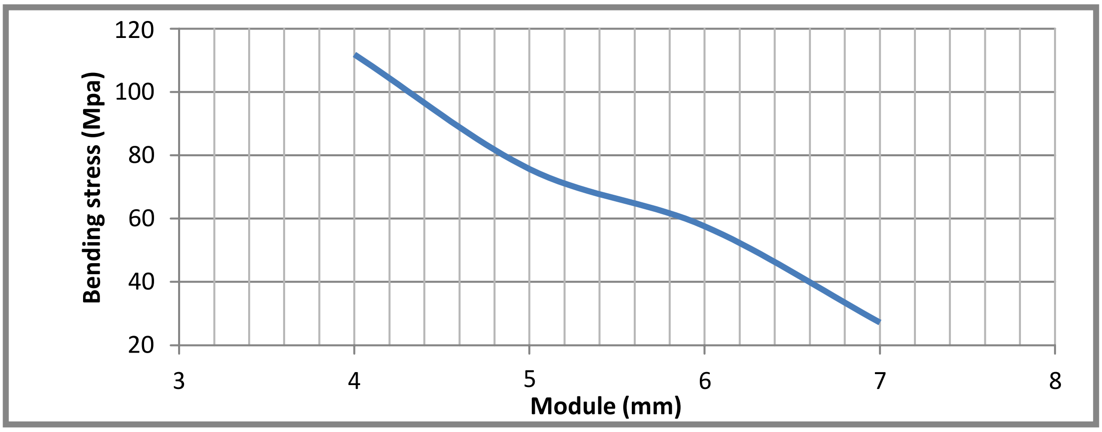


Figure 10. Variation of maximum bending stress with the variation of module.

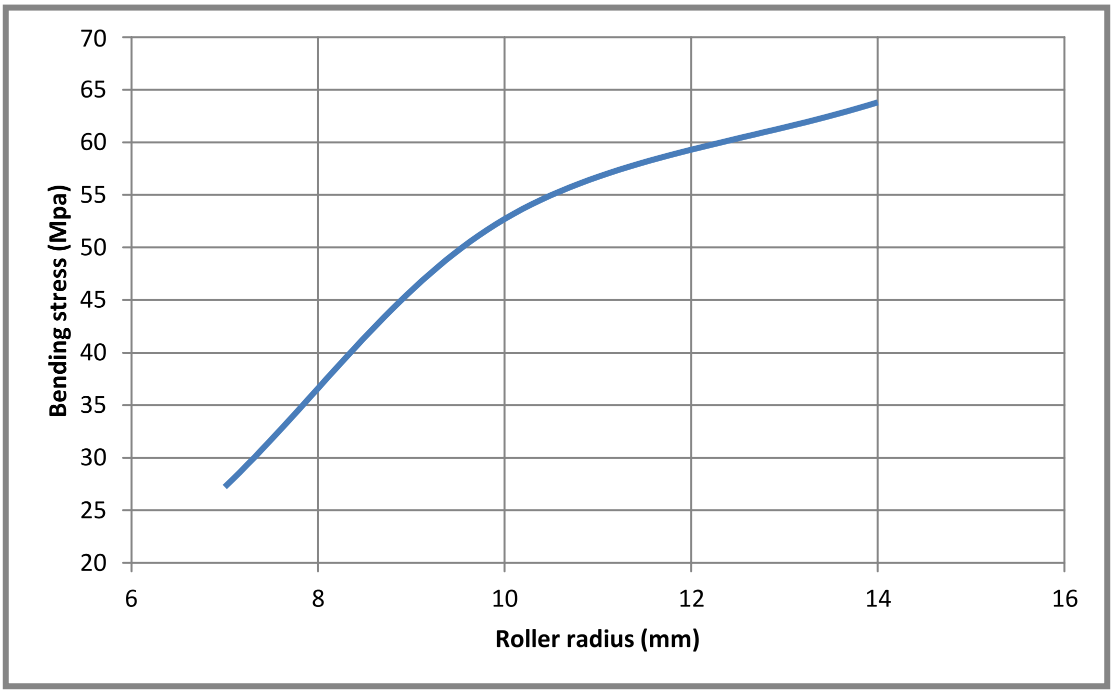


Figure 11. Variation of maximum bending stress with the variation of roller radius.

CONCLUSION

The geometrical model of the cycloid teeth-straight bevel gear and the FEM results led to the following conclusion: The roller radius should be as narrow as feasible to maximise the cross-sectional area of the cycloid teeth and minimise the bending stress.

The bending stresses decrease with increasing gear module size.

To keep bending stress at its lowest level, the bevel gear cone angle must be as minimal as feasible.

Although cycloid teeth's root thickness is improved by a lowering roller radius, the full-depth teeth may tip and become stub teeth as a result.

NOMENCLATURE

b: Face width (mm)

E: Modulus of elasticity (Gpa)
ha: Tooth addendum height (mm)
hd: Tooth dedendum height (mm)
N: Teeth number
m: Module (mm)
Rr: Roller radius (mm)
Rp: Pitch radius (mm)
Ra: Addendum radius (mm)
Rd: Dedendum radius (mm)
xe, ye, ze: Epicycloidal profile coordinates (mm)
xh, yh, zh: Epicycloidal profile coordinates (mm)
δ: Cone angle (degree)
θ: Rolling angle (degree)
ϕ: Description angle (degree)
$\sum$: Transmission angle (degree)

REFFERENCES

[1] T.J. Kim, “Transportation Engineering and Planning”, Book, Volume I, 2009.

[2] S.P. Radzevich, D.W. Dudley, “Handbook of Practical Gear Design, CRC press”, 1994.

[3] W.J. Patton, “Mechanical Power Transmission”, Book, Prentice Hall, 1980.

[4] S.P. Radzevich, “Advances in Gear Design and Manufacture”, Book, CRC press, 2019.

[5] V. Vullo, “Gears; Volume 1: Geometric and Kinematic Design”, Book, Vol. 10, Springer, 2020.

[6] J.R. Colbourne, “The Geometry of Involute Gears”, Book, University of Elberta, 1987.

[7] R.M. Kumar, “Studies in Cycloid Gears Made of Isotropic and Orthotropic Materials”, Ph.D. Thesis, Bharathiar University 2003.

[8] Z.Q. Qu, “Model Order Reduction Techniques with Applications in Finite Element”, Book, British Library, 2004.

[9] D.J. Wang, “Numerical and Experimental Analysis of Spur Gear in Mesh”, Ph. D. Thesis, Mechanical Engineering, University of Technology, Australia 2003.