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Numerical Investigation of the Effects of Pressure Angle and Face Width on Bending Stress of A Spur Gear

Atanu Barua, Md. Bayazid Ahmed and Wazed Ibne Noor

Department of Mechanical Engineering, University of Creative Technology Chittagong, Chittagong, Bangladesh.

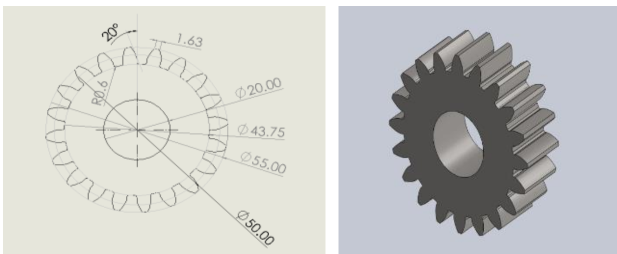
ABSTRACT

The gear mechanism is part of the power transmission system that transmits motion from one shaft to another. Recent studies have focused on tuning the different design parameters. As a result of these studies, the equipment's performance, strength and reliability are being improved. The bending stress of spur gears and the tangential load capacity are analyzed by numerical simulation using microbial multicomponent gas bubbles and the shear force activation concept. Using SolidWorks Simulation and AGMA criteria, various gear shapes were tested under continuously loaded conditions. Improving load-carrying capacity and reducing bending stress in gear teeth can be achieved by increasing the pressure angle and the face width. The tooth face width is increased to reduce the load on the tooth. Increasing the pressure angle will not affect the tooth's stiffness, but it will increase the principal stress. Future gears will be more effective and last longer as a result of this research.

Keywords: Spur gear, Bending stress, SolidWorks, FEA.

1. Introduction

Gears are mechanical devices that vary the speed and direction of the movement of an input shaft to an output shaft. The performance, life, and reliability of gears are affected by pressure angle and face width, which are design parameters. Gear design engineers must be aware of the impact of these parameters on gear design to assist in optimal design. Previous work has been done on the pressure angle of gears. People used different face widths to measure the bending strain. The deformation and pressure of the gear teeth have been measured and monitored in these studies. This indicates the learning about the possibility of observing pressure change through the capacity change of the gear. The tooth's strength and load-bearing capacity are determined by the pressure angle. A rise in pressure angle boosts strength and load capacity, but comes with added friction. A high-pressure angle weakens and reduces load capacity, but lowers friction. The sizes of the faces affect how the stresses are spread. Wider dentitions improve the strength and durability of the dentitions. This, however, increases the use of material. [4][5] This paper uses SolidWorks simulations and AGMA standards to investigate the effects of SolidWorks simulation bending stress and tangential force [6][7]. Past research shows that the geometry, material, and effects of dynamics matter. Sharma et al. [1] and Ben Moussa [2] achieved an optimized design and reduced the stress. Litvin and Düzcükog˘lu studied how pressure angle and tooth width affect durability. Hu and Senthilvelan adjusted the configuration of reinforced and polymer grades of gears at various speeds. Dhomse [8] and Nigus [9] proposed
FEA-based analysis. Real dtta, such as that for the specifications of a Toyota Corolla and further works, helps to deepen the fatigue, stress [10], or optimization understanding. The pressure angle is the inclination between the line of action and the joining line that connects the centers of gears. It is one of the important parameters of gear design [11-13].
This strongly affects the ability, strength, and efficiency of gears. If we choose the pressure angle for the gear tooth to be 30 and 40 degrees, then the strength of the gear tooth increases because the thrust angle becomes more inclined. A low-pressure angle will be smooth and have less friction. However, it will not be able to withstand heavy loads. The gear face width impacts the spread of load on the gear teeth. Stress concentrations could be lowered by bigger face widths. This also enhances the gear's ability to support tangential loads. Thus, it will improve durability. Yet, it also adds more material into the mix and makes the end result heavier. Both the parameters for optimizing the gear design must be considered for the same. This study investigates the Impact of change in pressure angle and face width on tangential force by using SolidWorks Modelling and Simulation.



a) b)

Fig. 1: Geometry of the spur gear- a) Technical drawing, and b) 3D model

*Corresponding Author Email Addres atanubaruactg1@gmail.com

2. Finite element modeling of the spur gear

2.1 Geometry

In this study, the 3D geometry was designed using SOLIDWORKS 2025, where all gear measurements were consistent with the ANSI metric system while maintaining consistency with normal measurements.

Gears were selected from the ANSI metric The tooth width was adjusted in SOLIDWORKS in four steps from 10 mm to 34 mm. On the other hand, the pressure angle was kept from 10 to 23, which was done in 3 steps.

Table 1: Geometrical Specification of the gear

Parameters	Pinion	Gear
Module	2.5	
Number of teeth	20	36
Pressure angle	20 degrees	
Face width	18 mm	
Bore diameter	20 mm	

2.2 Material Property

AISI 304 stainless steel is an austenitic alloy. It has about 18% chromium and 8% nickel, so it has outstanding corrosion resistance and good strength. It shows a tensile strength of about 515 MPa and a yield strength of about 205 MPa with high ductility and moderate hardness [14]. Due to its non-magnetic character and good weldability, it finds application in kitchen utensils, chemical containers, and building construction [15]. Stress corrosion is a problem for it despite its resistance to oxidizing as well as other corrosive environments. The manufactured shape can be evaluated in relation to the share ratio as it is readily machinable [16].

2.3 Meshing and Boundary Conditions

The gear model used for this test, created by "SolidWorks 2025", is the metric data-based gear model created for this test. To create the gear, we first select the gear data that is to be created. Subsequently, we make the gear based on its measurement so that the gear data is precise and suitable for our test. While designing I made a high mesh, where the total nodes are 292217, total elements are 200834. After that, we go to SolidWorks Simulation and run the simulation process.

Here it is said that in this drawing. We have changed the pressure angle and face width. The gear operation was performed using the following fitting parameters in the "SolidWorks 2025" fitting module: "Warn for twisted components" is checked, "Work parameters" is unchecked, and the mesh type is set to tetrahedral. During this SolidWorks simulation, several assumptions were made for the analysis: I) There is no friction loss between the driver and the driven gear. ii) Steady operating conditions are assumed throughout. iii) The meshing teeth do not have any gaps. iv) A uniform torque is applied to the driver gear. These assumptions help to simplify the simulation and ensure consistent boundary conditions for accurate results.

Table 2: Materials property AISI 304.

Elastic Modulus	210000.0005	N/mm^2
Poisson's Ratio	0.28	N/A
Tensile Strength	723.825617	N/mm^2
Yield Strength	620.4219978	N/mm^2
Tangent Modulus		N/mm^2
Thermal Expansion Coefficient	1.3e-05	/K
Mass Density	7700.000118	kg/m^3
Hardening Factor	0.85	N/A

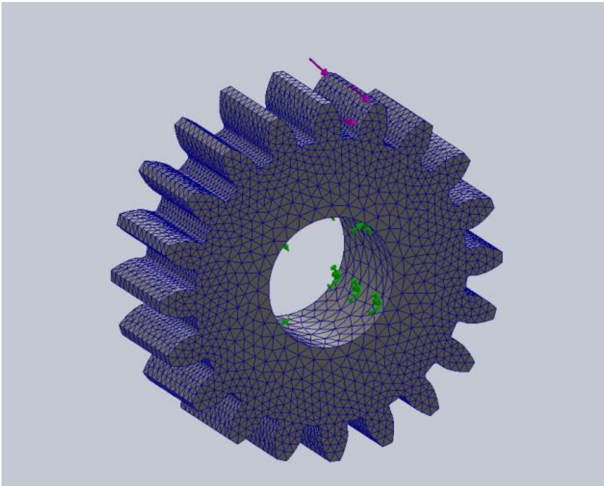


Fig. 2: Mash of the spur gear

2.4 Governing Equations

We will study why the gear deforms while it is in operation. It will give us the reason for this deformation. The yield point of the material is at the von Mises stress, which starts showing plastic behavior of the material. Above that point, it undergoes a permanent deformation. This metric is important to find the failure time of the material, which helps in design decision-making. We examine a theoretical problem in order to obtain various results that will aid any design problem. The analysis governing equations show the material behavior enabling designing for durability and structural integrity. The governing equations used for the analysis are as follows:

$$H_{BP} = H_{BG} = 200 \tag{1}$$

Allowable Bending Stress,

$$\sigma_{allow} = \frac{\sigma_y}{F_s} \tag{2}$$

Allowable contact stress,

$$\sigma_{c,allow} = \frac{\sigma_{c,y}}{F_s} \tag{3}$$

Bending stress,

$$(\sigma_b)_P = \frac{W_t K_a K_v K_s K_m}{Fm} \frac{1}{(Y_J)_P} \tag{4}$$

Factor of safety,

$$(S_F)_P = \frac{S_t(Y_N)_P}{(\sigma_b)_P Y_Z} \quad (5)$$

Gear design must involve the application of these equations to ensure strength and durability. The allowable bending stress $\sigma_{c,allow}$, is calculated by the yield strength of the material divided by the safety factor, and the allowable contact stress σc is calculated by the contact fatigue strength of the material divided by the safety factor. The tests also show that the gear is capable of withstanding breaking forces [3-5].

The bending stress $(\sigma_b)_P$ is influenced by the tangential load, the material, and the geometry of teeth, which can undergo bending. The ratio of the tangential force to the bending stress defines the safety factor (SFP). The chance of failure is less, and the performance of the elements suffers less.

3. Results and discussion

In this study, we tested the pressure angle and the face width in two different ways. I tried to find out the load capacity of a single gear tooth and the effect of changing its pressure angle. Both the deformation and von Mises stress were calculated using SolidWorks tangential load and other methods, and the results are shown in the graphs below.

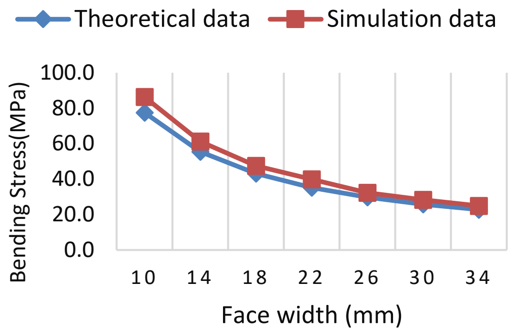
3.1 Effects of face width on bending stress

The graphs section illustrates the product's output as a function of different factors. The first graph shows how the pressure angle is held constant by varying the face width of the gear. We can observe how face width alone affects carbon scores. The other graph is obtained by varying the pressure angle but keeping the face width constant. The results can be seen regardless of face width, but the pressure angle has an effect. The respective change, followed by complete experimental data, is shown in the two graphs. The data offers an enhanced relationship of face width vs. pressure angle and their overall effect. The relationship between face width, pressure angle, and pinion gear bending stress is illustrated in the four graphs.

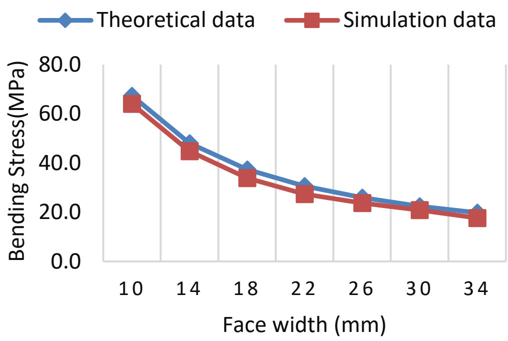
Fig.3.1 shows that for a 20° pinion gear, bending stress decreases with an increase in face width. The theoretical data starts at 77.552 MPa, and at 10 mm it drops to 22.809 MPa at 34 mm. SolidWorks data starts at 86.345 MPa and after that, decreases by 24.906 MPa. This shows that wider faces distribute load better, which reduces stress. SolidWorks appears to consistently yield higher values of bending stress due to more realistic modelling. The Pinion is examined for Bending Stress against Pressure angle [16]. The Face Width is fixed at 18 mm for this graph. Both theoretical and SolidWorks data show that as the pressure angle increases, the bending stress decreases. Based on theoretical values, numbers in the data dropped from 47.872 MPa to 41.547 MPa, while in SolidWorks, the data dropped from 67.970MPa to 41.547 MPa. This indicates higher pressure.

3.2 Effects of face width on bending stress

Angles help distribute loads more evenly, which reduces bending stress. SolidWorks takes care of more complex things.

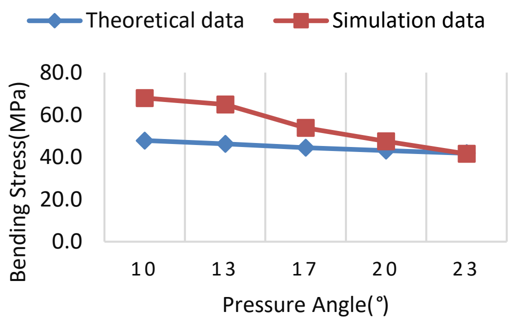


a)

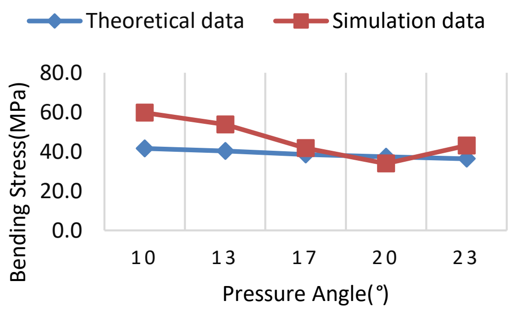


b)

Fig. 3.1: Change in bending stress with respect to face width for a) pinion, b) gear, for a pressure angle of 20 degrees



c)



d)

Fig. 3.2: Variations by bending stress with respect to pressure angle for a) pinion, b) gear, for a pressure angle of 18 mm

The graphs in Figure 3.2 also present ordinary gear results. As the face width and pressure angle increase, neither one has less bending stress.

The SolidWorks data is always more than the theoretical data. Because of this, SolidWorks analysis studies that it is mainly the gear material property and gear tooth shapes that cause the bending stress to increase. In conclusion, we can obtain minimum bending stress and also increase the strength of the gear. Face-width and pressure angle are properly selected for achieving the above purpose. The stress values given by SolidWorks are more reasonable but slightly higher. If the pressure angle of a cylindrical gearbox design is greater, then tooth strength and load-carrying capacity improve. However, this focuses the effort at the base of the tooth, causing an increase in the bending stress and an increased risk of failure from high loads [14]. The design is changed for tooth change and material optimization, which assists in stressing [15].

3.3 Optimization

The pinion bending model has an R-sq. value of 98.74%. The bending of pinions has nearly all its variability explained. A 97.85% value of adjusted R-squared, given the number of predictors, suggests that the probability of overfitting decreased. The model is also quite strong for predicting out-of-sample observations since the predicted R-sq. is 91.07%. The regression model is well-suited for the data.

Table 3: ANOVA Analysis for Pinion

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	5	5408.19	1081.64	110.02	0.000
Linear	2	4754.05	2377.02	241.78	0.000
Face Width	1	4181.79	4181.79	425.35	0.000
Pressure Angle	1	572.25	572.25	58.21	0.000
Square	2	591.51	295.75	30.08	0.000
Face Width*Face Width	1	533.46	533.46	54.26	0.000
Pressure Angle*Pressure Angle	1	20.62	20.62	2.10	0.191
2-Way Interaction	1	62.64	62.64	6.37	0.040
Face Width*Pressure Angle	1	62.64	62.64	6.37	0.040
Error	7	68.82	9.83		
Lack-of-Fit	3	68.76	22.92	1578.57	0.000
Pure Error	4	0.06	0.01		
Total	12	5477.01			

From the ANOVA analysis, we conclude that overall regression is highly significant for both models as their p-value is less than 0.05 and thus their predictor defines an important contribution to the pinion bending. According to Table 3, the linear effects of face width and pressure angle, as well as the square terms and the interaction between face width and pressure angle, are statistically significant. The interaction between face width and pressure angle was significant at the 0.040 level, while the pressure angle quadratic term was not significant at the 0.191 level.

As seen in Table 4, linear terms of face width and pressure angle still remained significant, along with square terms and face width squared interaction. The term of a squared pressure angle ($p = 0.225$) and the two-way interaction of face width and pressure angle ($p = 0.096$) are not significant. According to Tables 1 and 2, face width, pressure angle, square terms, and face width squared interaction are significant. Moreover, Table 1 indicates a significant interaction between face width and pressure angle.

The regression model explains the fit of pinion bending (BB) by 96.66 %, which is a good fit. The adjusted R-sq. (R-squared) of the model is 94.28% indicating a good model. The produced estimated R-sq. is 76.44 per cent, suggesting a moderate predictive accuracy, while the background data has a good fit as per reasonable standards. Overall, this model has a good performance but low generalization ability.

Table 4: ANOVA Analysis for Gear

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	5	3945.10	789.02	40.56	0.000
Linear	2	3447.51	1723.76	88.61	0.000
Face Width	1	2881.04	2881.04	148.11	0.000
Pressure Angle	1	566.48	566.48	29.12	0.001
Square	2	425.83	212.92	10.95	0.007
Face Width*Face Width	1	415.31	415.31	21.35	0.002
Pressure Angle*Pressure Angle	1	34.51	34.51	1.77	0.225
2-Way Interaction	1	71.76	71.76	3.69	0.096
Face Width*Pressure Angle	1	71.76	71.76	3.69	0.096
Error	7	136.17	19.45		
Lack-of-Fit	3	134.94	44.98	146.08	0.000
Pure Error	4	1.23	0.31		
Total	12	4081.27			

4. Conclusion

As per the study, the external surface width and tooth angle of a gear considerably affect the functioning of the gear in question. The experimental results showed that the rise in pressure and the gear face expansion are two developments that improve the efficiency, stress distribution, and load capability of the gear. South Grove gear employs very carefully designed features to enhance its durability and performance. They discovered a method to make the cars reach their destinations faster while making the body of the car more dependable. Designed for revolutions, allowing gear designs to branch out and adapt to the complications of organization in general industry.

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Nomenclature

Symbol	Description
σ_{allow}	Allowable bending stress (MPa or N/mm²)
σ_y	Yield strength of the material (MPa or N/mm²)
F_s	Safety factor (dimensionless, typically between 1.5 and 3)
$\sigma_{c,allow}$	Allowable contact stress (MPa or N/mm²)
$\sigma_{c,y}$	Material’s contact fatigue strength (MPa or N/mm²)
F_s	Safety factor (dimensionless, typically 1.5 to 2.5)
δ	Displacement or deformation (in units like meters or millimeters)
F	Applied force (in units like Newtons)
k	Spring constant or stiffness (in units like Newtons per meter, N/m)