Analysis of Maximum Stress due to Bending in Helical Gears: Comparison Between the Agma Method and Finite Element Analysis

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This study presents a methodology for designing helical gears based on Shigley's calculation approach and AGMA standards, including the calculation of maximum bending stresses and the application of safety factors. The analytical results were compared with numerical results obtained from a structural analysis using the Finite Element Method (FEM), applying the von Mises yield initiation failure criterion. The methodology involved defining geometric and operational parameters, calculating the tangential load, and considering overload factors to achieve more realistic simulations. The maximum stress obtained from the finite element analysis showed a 3.85% difference relative to the analytical calculation from the AGMA standard, demonstrating good agreement between the methodologies.
Keywords: Helical Gears. AGMA Standard. Maximum Bending Stress. Finite Element Method.

Helical gears used to transmit small powers and short center-to-center distances [1] have greater face width in contact than spur gears of the same size, which gives them a higher load-carrying capacity [2]. In addition, the load is distributed among several teeth, allowing smoother transmission and more consistent elastic flexibility [2], as occurs in rolling mill gearboxes, as illustrated in Figure 1. Due to these characteristics, the design of helical gears requires detailed study and conservative methods to ensure greater reliability and component life.

This study aims to explore three main aspects: (1) the design of helical gears based on Shigley's book [3] and AGMA standards, using a virtual example with fundamental parameters, including the calculation of maximum bending stresses and their respective factors; (2) the modeling of the gear designed following the AGMA methodology for defining tooth dimensions; and (3) the simplified static structural analysis of the pinion (using finite elements) to compare with the maximum bending stress through the von Mises criterion.

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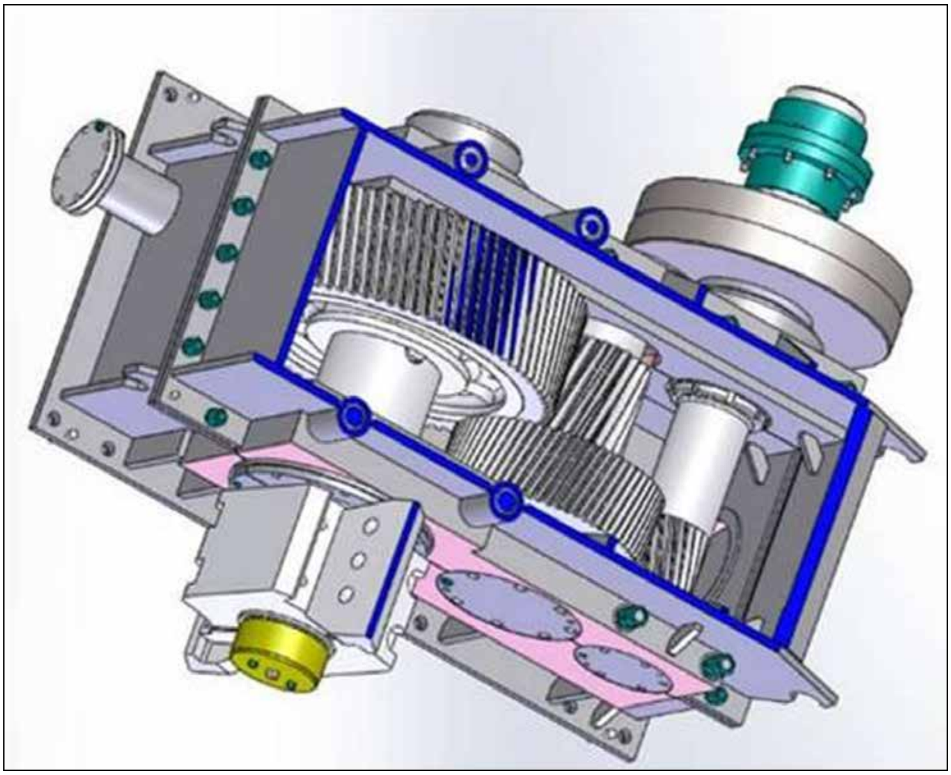
Rolling, one of the primary steel forming processes, involves reducing the material's cross-section to predetermined dimensions and shapes [4]. During the process, rotating work rolls, driven by motors and gearboxes (Figure 1), generate enough energy to deform the rolled material [5]. According to Santi and Trazzi (2015), due to the large forces present in the rolling process, the drive system must be robust enough to withstand the imposed loads [6].

Materials and Methods

The helical gears designed in this study had their initial parameters defined in Kutz's book [7] and are presented in Table 1, which outlines the geometric characteristics of the gears and their operating conditions. Based on these specifications, the necessary dimensions were calculated for modeling the gear and pinion in SolidWorks, as illustrated in Figure 2, along with the boundary and loading conditions used in Ansys.

During the gear modeling, a fillet with a 1.5 mm radius was added at the tooth root. This region received a more detailed mesh refinement to ensure accuracy in the load application area. For analysis, the gear wheel was suppressed, and a tangential load equivalent to 4500 [N] was applied directly to the pinion tooth face. Table 2 presents the factors used for the analytical calculation of maximum bending stresses, the safety factors for the pinion and gear,

Figure 1. Rolling mill gearbox – CAD model.

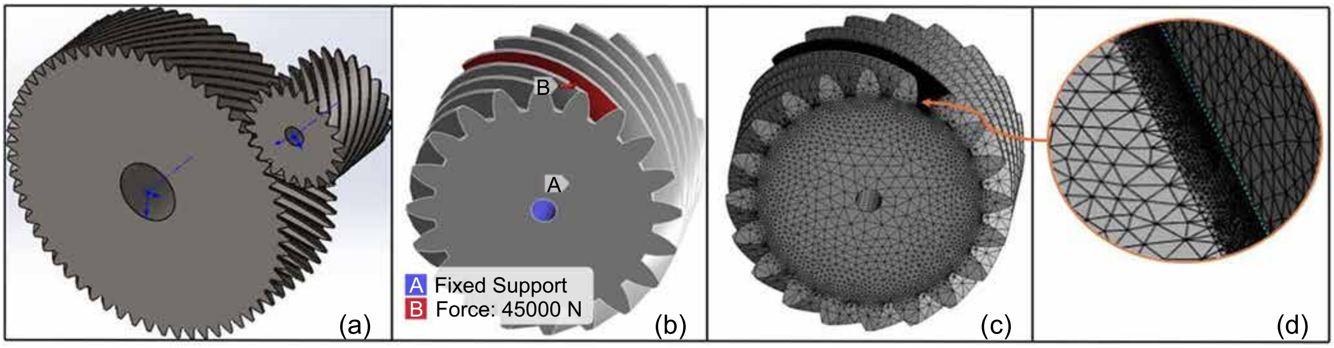


Source: GMB Heavy Industries, 2025 [8].

Table 1. Gear parameters and operating conditions.

ParametersValues					
Number of pinion teeth (*Np*)	20	Gear hardness	300 HB	Reliability	99%
Number of gear teeth (*Nc*)	60	Modulus of elasticity (pinion and gear)	205 GPa	Operating temperature	120 °C
Metric module (m)	10 mm	Poisson's coefficient (pinion and gear)	0.29	Quality standard	Qv = 7
Transmission ratio	3:1	Maximum torque on input shaft	2,000 Nm	Pinion pitch diameter (dp)	200 mm
Pressure angle (ϕ)	20°	Pinion rotation	1000 rpm	Gear pitch diameter (dg)	600 mm
Helix angle (ψ)	30°	Machine load	Heavy shock	Transverse angle (ϕt)	22.08°
Pinion hardness	370 HB	Power source overload	Medium shock	Face width (b)	157.08 mm

Figure 2. (a) CAD gear model, (b) boundary and loading conditions, (c) finite element model, and (d) detail of the fillet region at the pinion root.



and the results obtained, calculated using equations (1), (2), and (3).

The comparative methodology between the AGMA and Finite Element Methods adopted is similar to that of Queiroz de Jesus and colleagues (2024) [9], differing in the type of gear used (spur gears) and in its specific application (high-performance automotive), resulting in efforts of different magnitudes.

$$\sigma = W_t K_o K_v K_s \frac{1}{b_m} \frac{K_h K_b}{Y_j} \quad \text{Eq. 1}$$

$$\sigma_R = \frac{S_t Y_N}{K_T K_R} \quad \text{Eq. 2}$$

$$S_F = \frac{\sigma_R}{\sigma} \quad \text{Eq. 3}$$

An important aspect of the analysis is the overload factor (*Ko*), applied conservatively in the AGMA calculation. This factor incorporates external loads that exceed the nominal tangential load Wt in specific applications, accounting for variations [10] in torque and operational shocks that can amplify the forces acting on the pinion.

Thus, the maximum bending stress was calculated considering the worst-case scenario, ensuring a more realistic approach. For the simulation, the tangential load applied to the tooth was defined as *KoWt* = 45,000 N. To reduce computational cost, the simulation was performed only on the gear subjected to the highest maximum stresses generated by contact with the mating gear, maintaining the conservative approach without compromising result accuracy.

Results and Discussion

The results presented consist of the analytical calculation of the pinion tooth root stress and the stress field calculated using the finite element method. Table 2 presents the results of maximum bending stresses and analytically calculated safety factors, obtained using SMath and Excel software, along with the dimensioning factors employed to achieve these results.

Figure 3 shows the von Mises stress plot in the region where maximum bending stress occurs on the pinion tooth. It is observed that the maximum equivalent von Mises stress at the tooth root of the gear was approximately 3.85% lower than the value calculated analytically according to the AGMA standard. The error obtained is consistent with the values found in the works of Gidado, Muhammad and Umar (2014) [10] and Queiroz de Jesus and colleagues (2024) [9].

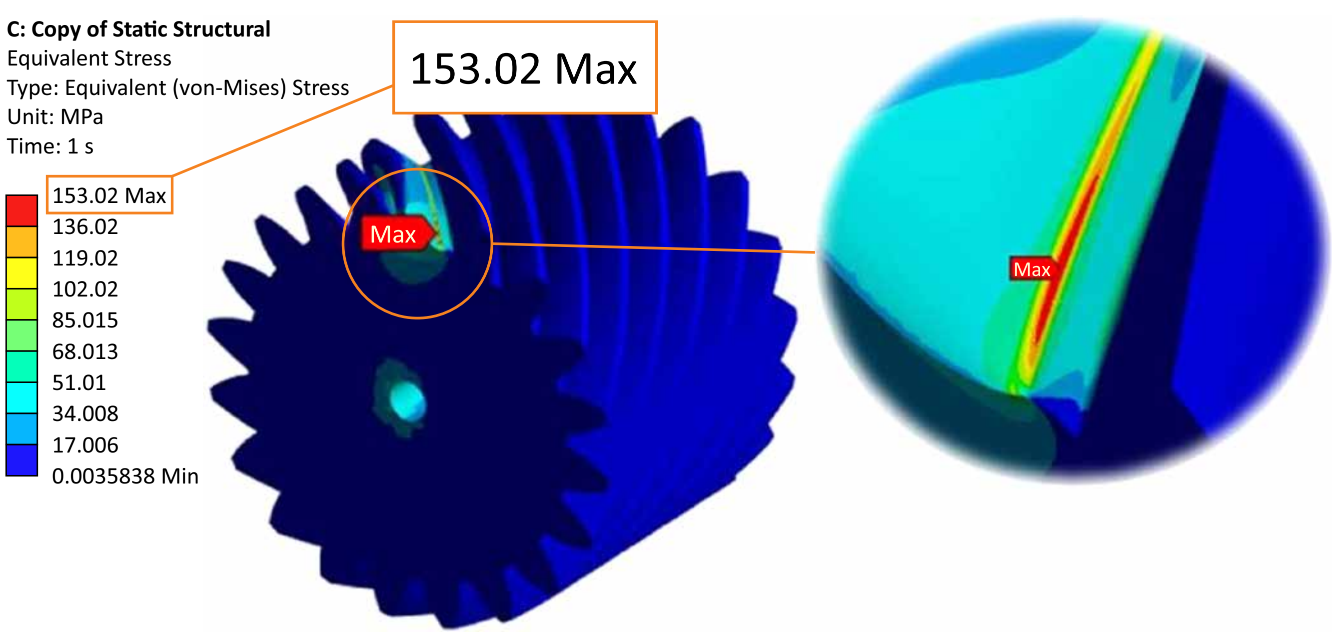
Conclusion

The study corroborated the effectiveness of the finite element method in validating the analytical design of helical gears, in accordance with AGMA standards, highlighting its applicability in high-demand projects. The simulation performed in Ansys® showed good agreement with the analytical calculations, while the inclusion of the overload factor provided a more conservative forecast under adverse scenarios.

Table 2. Maximum bending stress and safety factors.

Bending Strength of Helical Gears		**Safety Factors of Helical Gears**	
Tangential force (W_t)	20,000 N	Pinion bending strength (σ_R)P	275.60 MPa
Dynamic factor (K_v)	1.476	Gear bending strength (σ_R)G	320.00 MPa
Overload factor (K_o)	2.25	Stress-cycle factors (Y_N)P	0.94
Size factor pinion (K_s)P	1.21	Stress-cycle factors (Y_N)G	0.96
Size factor gear (K_s)G	1.22	Reliability factor (K_R)	1.00
Load distribution factor (K_m)	1.35	Temperature factor (K_T)	1.00
Rim thickness factor (K_b)	1		
Geometric bending strength factor (Y_j)P	0.4371		
Geometric bending strength factor (Y_j)G	0.5307		
Results			
Maximum pinion bending stress	158.90 MPa	Pinion safety factor (SF)P	1.73
Maximum gear bending stress (σ) G	131.81 MPa	Gear safety factor (SF)G	2.43

Figure 3. Result of the analysis in ANSYS.



For future studies, it is recommended to investigate gear tooth fatigue by estimating the number of cycles until crack initiation and propagation through numerical and analytical analyses.

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