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Experimental Study of Linear Profile Modification in Spur Gear Leg

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Abstract. Gears with the involute profile are assumed to be rigid and good at producing mating gear. They are also deformable and deflectable, indicating additional space is required to avoid interference from the unpredictable shaft, misalignment, and dynamic factors. In addition, load and rotational speed cause frequent mild to severe interference leading to a significant reduction in its lifespan. This study conducted an experimental comparison between standard and modification involute gear profiles in a ratio of 1:2 to reduce interference. It was carried out using 18 pinion teeth with a 72 mm base diameter at three and four levels of torque load and speed. The load was also mounted a strain gauge on pinion gears to conduct strain measurement on the gear leg. Finally, it was applied using a mechanical brake measured by a dynamic torque meter. The experimental test obtained microstrain when the modified spur gear contacted for 3.04 – 6.67 milliseconds depending on the rotation speed and torque load. Modification of the linear involute profile significantly reduced the tooth leg strain at the rate of 5.6% to 13.99%. Meanwhile, the maximum microstrain reduction of 13.99% occurred at a speed of 1100 rpm and a torque load of 65%.

Keywords: Interference; Involute profile; Modification profile; Spur gear; Strain reduction

1. Introduction

Modern-day mechanical gears are widely used in transmission systems to transmit torque included in the wave energy converter (Ariefianto, Hadiwidodo, and Rahmawati, 2022). Despite manufacturing new transmission systems with magnetic gears, no physical contact between materials has been identified (Rahman, Hassan, and Ihsan, 2022; Niguchi, Hirata and Zaini, 2013). (Li and Bird, 2018) stated that these various benefits of mechanical gears prevent the need for lubrication, high durability, and a high-speed-reduction ratio. Its main shortcoming includes lower torque density. Therefore, continuous efforts are carried out to extend the service life and reduce mechanical gear vibrations. When a backlash is overly narrow, it causes overheating, overload, noise, and jamming, while a loose one leads to a significant impact load and excessive vibrations (Dyaneshwar and Mangrulkar, 2016).

The amount of backlash can be determined by considering various factors. The first factor is based on the module range or diametral pitch, while the second factor involves evaluating the diametral pitch and the centre of the distance axis. A third factor utilizes the second parameter of diametral pitch and line velocity. However, it is often difficult to determine the exact amount of backlash required due to manufacturing imperfections. (Karba *et al.*, 2019).

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Planned backlash aims to increase the distance between the shafts or cut the side of the gear. The study by (Sharma, Moorthy and Kumar, 2015) used the finite element method to investigate strain reduction in the spur gear leg. Unplanned backlash with unpredictable magnitude cause problems involving various parameters, such as tooth and shaft deflections, load, rotational speed, bearing clearance, temperature, and lubrication. The concept of a second factor caused by material stiffness and contact stress was further developed to address contact stress between non-metal and non-metal (Rahman, Shoukaku, and Iwai, 2021). Meanwhile, (Eritenel *et al.*, 2003) conducted a numerical study using the finite element method. The study found that the fillet radius caused bending deflection, which influenced the base stiffness as defined by (Joshi and Karma, 2011), and this in turn affected the magnitude of the backlash.

(Guo, Keller, and LaCava, 2012) conducted a study on the effect of bearing clearance distance on planetary stress. The study found that increased bearing clearance leads to a higher non-torque workload. Preliminary studies have been conducted using numerical simulations and approximation equations to determine the minimum gap width in the elastohydrodynamic lubrication (EHL) channel (Dowson and Higginson, 2014; Lubrecht, Venner and Colin, 2009). (Li and Kahraman, 2011) selected a similar approach involving the pre-calculation of dynamic tooth strength. Meanwhile, (Jeon *et al.*, 2011) designed the last factor using an inaccurate profile of 0.015 to 0.033 EDM.

This study addressed the requirement for adequate backlash by implementing a critical profile modification. Prior research has explored the transformation of the tooth profile using the modeling and dynamic simulation (Liang, Zuo, and Feng, 2018), artificial neural networks (Devendiran and Manivannan, 2015b), and genetic algorithms (Devendiran and Manivannan, 2015a). One of the critical problems in the industry is determining what to do, assuming the service life of the gear is shorter than expected. According to studies, the easiest and most expensive solution is replacing the gear or modifying its tooth through lead modification processes in the axial direction to reduce the effect of load concentration due to misalignment. Profile modification targets tooth deflection due to mechanical loads. Therefore, the available backlash does not provide free space, which causes low to high interferences, leading to jamming.

The tooth profile modification on the gears occurs in the dedendum and addendum areas. Addendum modifications, also known as profile tips or relief modifications, are more common than dedendum due to their inability to reduce bending strength. The amount (Δ) and length of profile modification (L_n) are the parameters used to modify the profile tip, as presented in Figure 1. Amount profile modification is an important parameter determined by the actual operating conditions. It is estimated by determining the mechanical and thermal loads deflection and the error profile of the manufacturing process.

This study was conducted using the tooth flexural deflection and contact deformation with a linear tip relief profile (Marković and Vrcan, 2016; Buljanovic and Obsieger, 2009). Modifications to the linear tooth profile led to a lower dynamic response compared to the unmodified tooth profile. Furthermore, the modifications involved different gear types, including spur (Ghosh and Chakraborty, 2016; Li *et al.*, 2016), helical (Wu, Wang and Han, 2012), and planetary gears (Bahk and Parker, 2013) studied through finite element modeling and simulation.

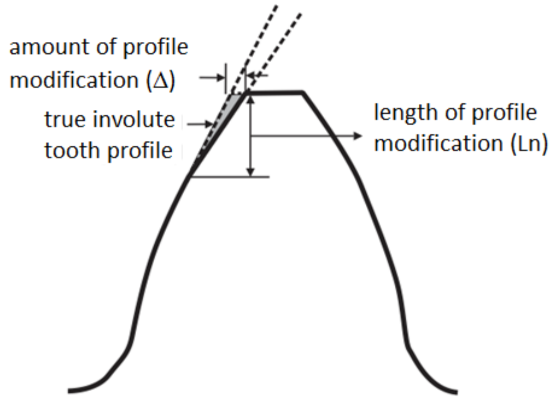


Figure 1 Profile of modification parameters

Experimental methods were employed to measure the stress-strain in the root tooth, including the use of strain gauges (Lisle, Shaw, and Frazer, 2019) and photoelasticity (Raptis and Savaidis, 2018). The experimental fatigue method was conducted by calculating the cycle and administering the load in stages (Concli, 2021; Liang, Zuo, and Feng, 2018). It is more challenging to take dynamic robot tooth measurements in real-time because it requires data logger equipment capable of speedily recording dynamic data and writing the entanglement problem. Most dynamic photoelasticity measurement methods use a stroboscope to produce quasi-static or momentary photos (Patil, Patil, and Nulke, 2018). Few related studies existed on using a high-speed camera to record real-time data using transient dynamic photoelasticity measurements. In the symmetric gear tip relief profile, it can not only reduce the bending stress but also reduce the contact stress (Molnár, Csoban, and Zwierczyk, 2021).

Another way to increase bending strength is to use asymmetric gears formed from two involute profiles with different contact angles on the drive and coast sides. Several studies have been conducted on these gears. For instance, (Pramono and Rizal, 2021) researched the influence of the asymmetric factor to reduce bending stress, using the contact-angle cosine ratio of the drive and coast sides as the asymmetric factor, representing the symmetric spur gear. The greater the asymmetric factor, the higher the contact angle difference and decreased bending stress. Furthermore, (Mallesh *et al.*, 2009) studied the effect of profile shift on the bending stress of asymmetric gears with x values of 0, 0.1, 0.2, 0.3, 0.4, and 0.5. The results showed that the larger the profile shift, the smaller the bending stress, applicable to different asymmetric values with contact angles of 20^o to 30^o. (Mallesh and VG, 2020) also investigated modeling using the asymmetric factor and profile shift. The increase in pressure angle on the drive side leads to a decrease in contact ratio and tooth thickness on the addendum circle and a rise in pressure angle. The tooth thickness on the addendum circle decreases for gears with increased profile shifts as the pressure angle on the drive side rises. Consequently, the bending stress at the critical section reduces significantly.

(Mo *et al.*, 2022) compared the analytically obtained meshing stiffness increased with a rise in asymmetric factors. In addition, the modification of teeth allows for a smoother transition between single and double ones.

Besides reducing bending stresses, asymmetric gears with tip relief can also lead to a decline in contact stresses, as (Karpat and Ekwaro-Osire, 2008) using Archard's analytical equation. As the amount of tip relief increases, the wear depth, particularly at the beginning and end of the mesh, decreases. Similarly, as the number of wear cycles increases, the effect of the tip relief modification on wear depths decreases slightly.

(Yılmaz, Dogan, and Karpat, 2017) proposed the asymmetric trochoid profile to decrease maximum bending stress and move the upper critical point. It is evaluated that effect of rim thickness on bending stress is more significant for those less than 1.3×module because of higher rim deformations.

Although asymmetric gear has the advantage of increasing bending and contact strength, it also has the disadvantage of rotating in only one direction, indicating it cannot be used for two-way transmission. When the drive side is worn, it must be replaced, unlike the symmetric gear, which can be reversed. Therefore, further study is needed to investigate the performance difference between symmetric and asymmetric gears.

This experimental study involved a linear comparison between the standard and modified involute gear profile at a ratio of 1: 2, where the number of pinion teeth and diameter are 18 and 72 mm. Strain measurements were taken using a strain gauge mounted on one pinion tooth, with slip rings used to avoid entangling the cable during rotation. Load torque was measured using a mechanical brake.

2. Methods

The various experimental stages include making standard tooth profiles and modifications, designing, manufacturing, and setting up test equipment or benches. The last experiment tested both the standard and the modified profile by following procedures, as shown in Figure 2.

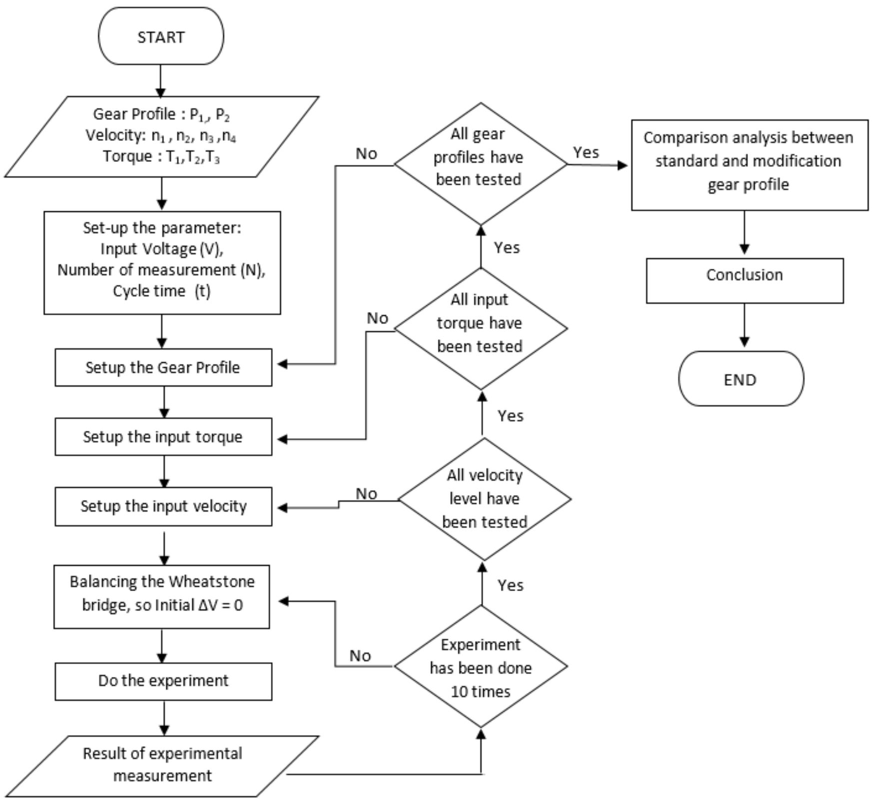


Figure 2 Flowchart of experiment procedures for spur gear with profile modification

2.1 Determination of the Number of Pinion Teeth

Involute gears can be readily generated by rack-type cutters, such as hob. Gear generation can also be also accomplished with gear-type cutters using a shaper or planer machine. In the production of standard straight gears, a small number of teeth can lead to undercutting at the dedendum. The undercut not only weakens the tooth with a wasp-like waist but also removes some of the useful involute adjacent to the base circle. To determine the minimum number of teeth required to avoid undercutting, one can refer to Equation (1) (Michalec *et al.,* 2009).

$$Z_c \geq \frac{2}{sin^2\theta} \tag{1}$$

where :

Z_c is the minimum number of teeth needed to prevent undercutting, and

θ represents the pressure angle or contact angle.

However, if a smaller number of teeth is desired without undercutting, standard spur gears may require modifications such as profile shifting, which is described by Equation (2) (Michalec *et al.,* 2009).

$$x = 1 - \frac{Z_c}{2} sin^2\theta \tag{2}$$

where :

x is the shift coefficient.

In this study, standard spur gears without undercutting were used, following Equation (1), and with a contact angle of 20 degrees, the number of pinion teeth was determined to be 18.

2.2. Profile Modification

Straight gears are made of a standard involute surface gear profile and a linear modified involute, which start at distances of 1 1.8 mm and 1.6 mm, respectively. The cut at the tip of the tooth is 0.104 mm and 0.091 for the pinion and gear, respectively, as shown in Figure 3.

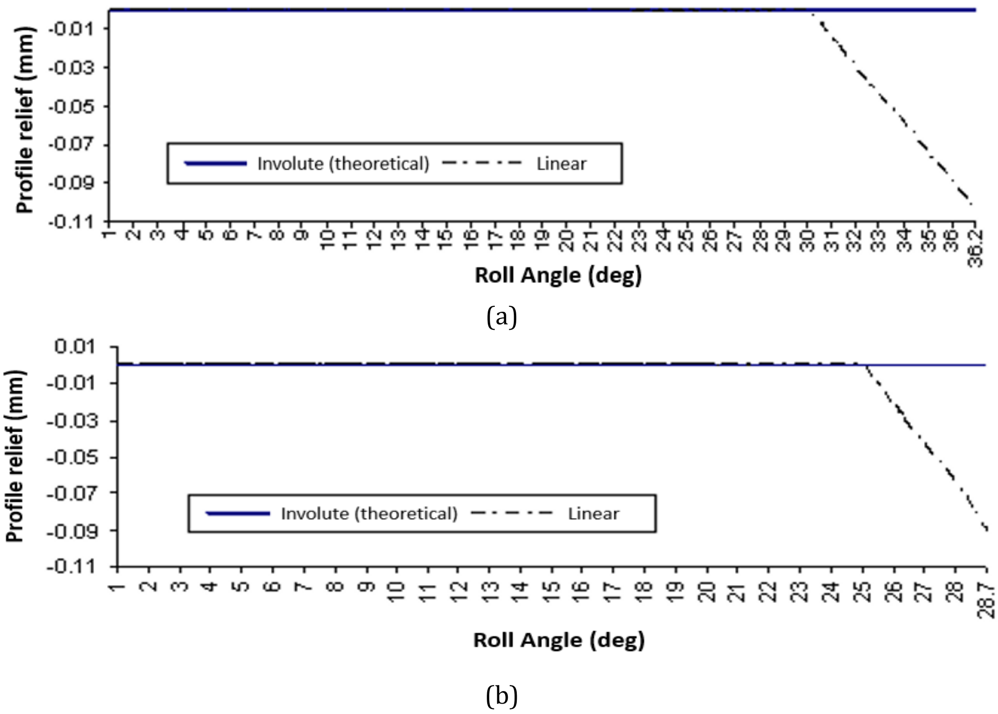
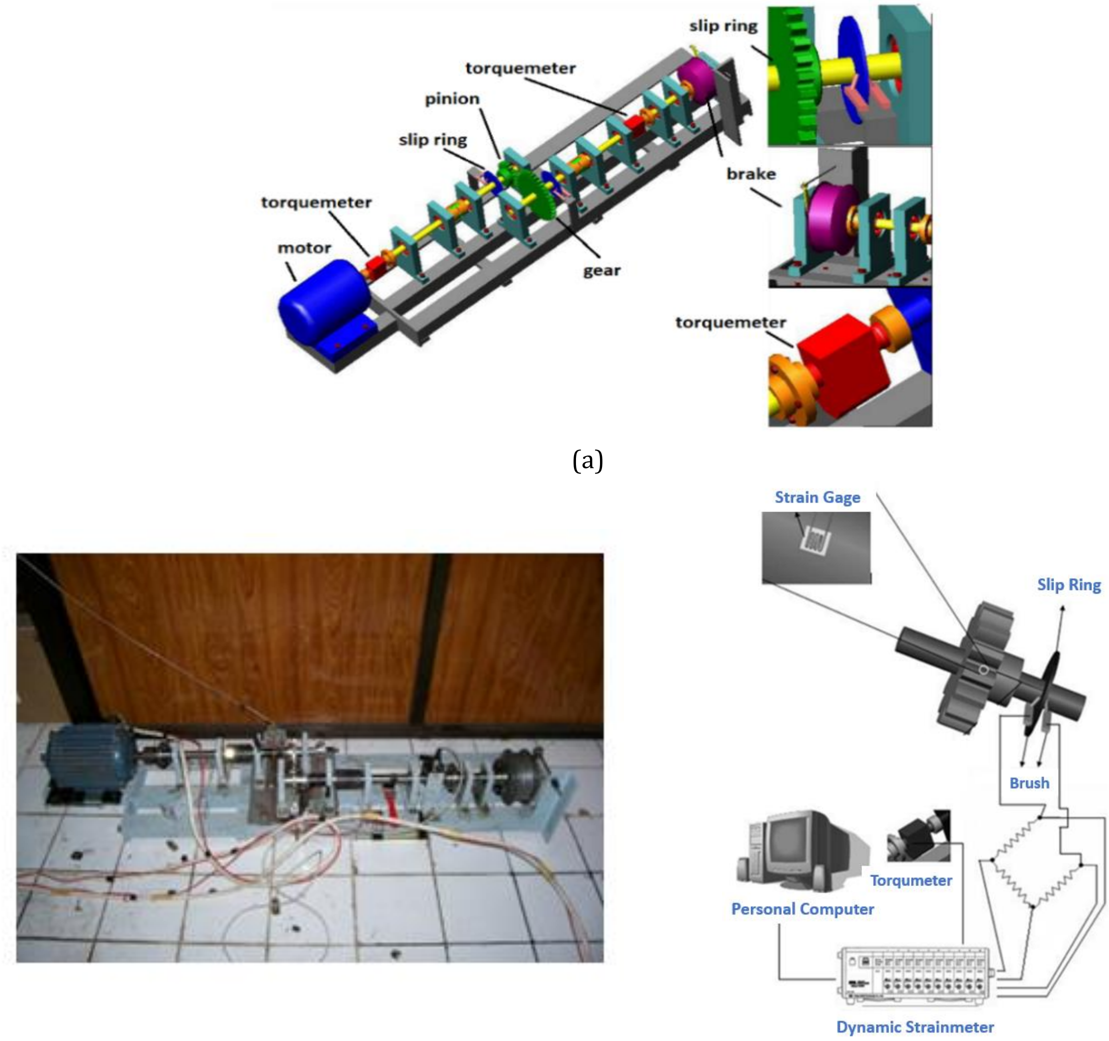


Figure 3 Profile of modification (a) pinion and (b) gear

2.3. Design and Prototype of Test Bench

A test device was designed to measure the stress on the tooth root, as shown in Figure 4. The device is powered by a 1500 rpm motor with 1.5 horsepower. A Dynamic Torquemeter with a maximum torque of 160 Nm is mounted on the motor shaft and connected to the pinion spur gear shaft. A slip ring is installed near the pinion spur gear with a strain gauge attached to one of the root teeth. The slip ring is further connected to a series of bridge stones and a dynamic data logger linked to a PC. Once the pinion spur gear is paired with the driven spur gear, slip rings are installed. Finally, a brake system is installed as a load at the drive shaft's end.



(b) (c)

Figure 4 Experimental study: (a) Design, (b) Set up, (c) Strain measurement

2.4. Determination of Time for One Contact Gear

Due to the varying rotating speeds, it is necessary to calculate the time of one contact gear for each rate. It takes 1/8.33 = 0.12 seconds for the motor shaft speed of 500 rpm or 8.33 rps to completely rotate for 18 teeth. Therefore, one gear contact takes 0.12/18 = 0.0066 seconds or 6,67 milliseconds. The same calculation is carried out for other speeds, and the results are presented in Table 1.

Table 1 Contact duration on one teeth pair

Angular speed		Contact duration of one teeth pair (ms)
rpm	rps	
500	8.33	6.67
700	11.66	4.78
900	15.00	3.72
1100	18.33	3.04

3. Results and Discussion

The stain gage mounted on one of the root teeth is a uniaxial strain gage that follows the bending direction with microstrain units. Following Hook's law, the bending stress in the elastic zone can be obtained, where the strain is proportional to the stress with a constant of proportionality.

The measurement resulting from a strain gauge is shown on a graph where the abscissa and the ordinate axis represent time and strain, respectively. Figure 5 shows the pinion with a rotation of 500 rpm. The maximum strain decreases due to linear modification for all load levels, which reduces the level of dynamic interference.

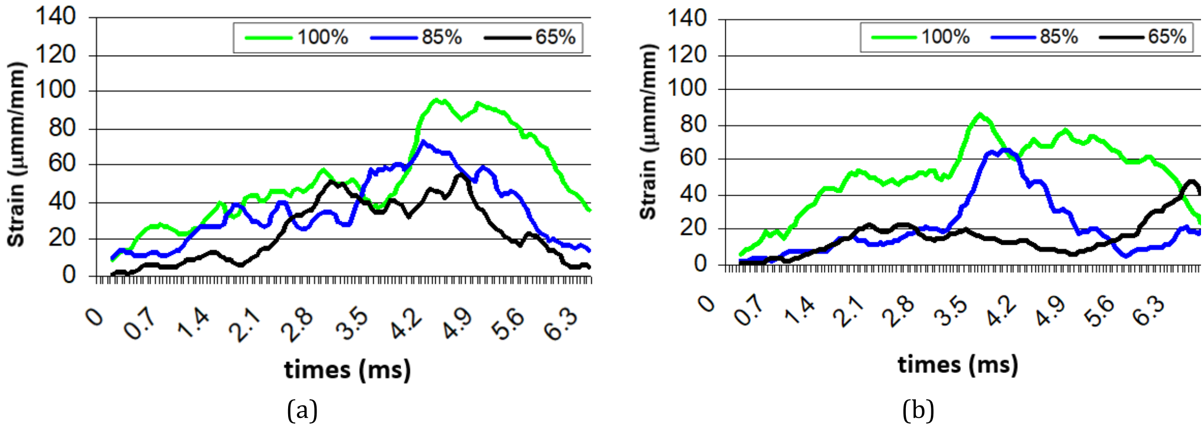


Figure 5 The results of the strain measurement on one teeth pair in contact at 500 rpm: (a) pinion involute, (b) pinion linear

The data involved maximum strain in facilitating comparison and further analysis. Figure 6 shows that the increase in torsional load raises the strain in the gear leg for all speeds in standards and modified involute profiles. This is in line with a theory that increasing the load will enlarge strain on the tooth foot and raise the tooth deflection, thereby causing interference.

In Figure 7, an increase in rotation at a constant load led to a rise in the strain and all load levels in accordance with the characteristics of an AC motor using a speed controller with an inverter. Furthermore, an increase in rotation led to a rise in the motor shaft and a slight rotation of the AC.

At 500 rpm rotation and 65% load, the profile modification reduced the strain by 9.44%, 8.55%, and 8.08% for 65%, 85%, and 100% loads, respectively. Meanwhile, at 1100 rpm rotation and 65% load, a maximum strain reduction of 13.99% was obtained, while at 85% load, it decreased by 11.48%. However, at 100% load, the strain decreased again to 9.41%, indicating that interference starts with increased tooth deflection.

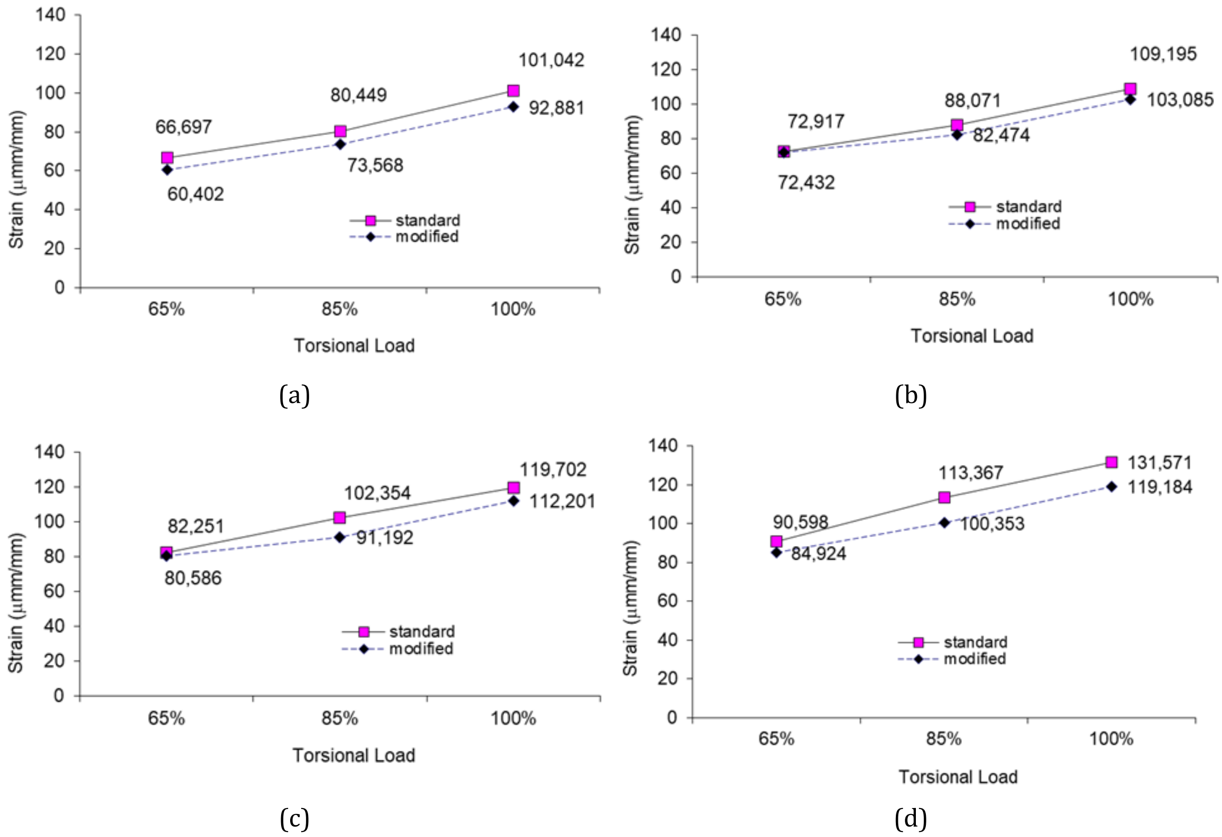


Figure 6 Comparison of strain as a function of load at (a) 500 rpm, (b) 700 rpm, (c) 900 rpm, and (d) 1100 rpm

An evident increase in load leads to a decrease in strain reduction. This is because the increase in load raises the deflection, thereby increasing the level of interference. An increase in load raises interference levels in all rotations by 500, 700, 900, and 1100 rpm, causing strain to decrease.

Other studies mostly came from simulations since experiments are rarely conducted. Even though experiments were performed, they did not carry out variations in loading and rotation. Therefore, the effectiveness of modifying the involute spur gear in various conditions experienced by the gear pair cannot be determined.

This study confirms the findings of (Li *et al.*, 2016) that modifying the involute spur gear can reduce impact and noise at the contact point by decreasing the strain value on the modified pinion gear. This reduction in strain helps to achieve an appropriate amount of backlash to prevent overheating, noise, impact, and excessive vibration.

This study proves that the reduction in strain value can occur in various loading and rotation conditions. Preliminary studies on tooth profile modification (Dai, Cooley, and Parker, 2016; Ghosh and Chakraborty, 2016; Bahk and Parker, 2013; Wu, Wang and Han, 2012) only focused on specific loading conditions. Therefore, the impact of profile modification is unknown for various conditions in spur gear pairs.

This dynamic experimental study also has advantages compared to other studies focused on maximum conditions. The contact between the spur gears for 3 - 6 ms can be well observed using the measuring instrument. Furthermore, the measurement results in microstrain can be graphed as shown in Figure 8. This cannot be found in the photoelasticity experimental method (Raptis and Savaidis, 2018).

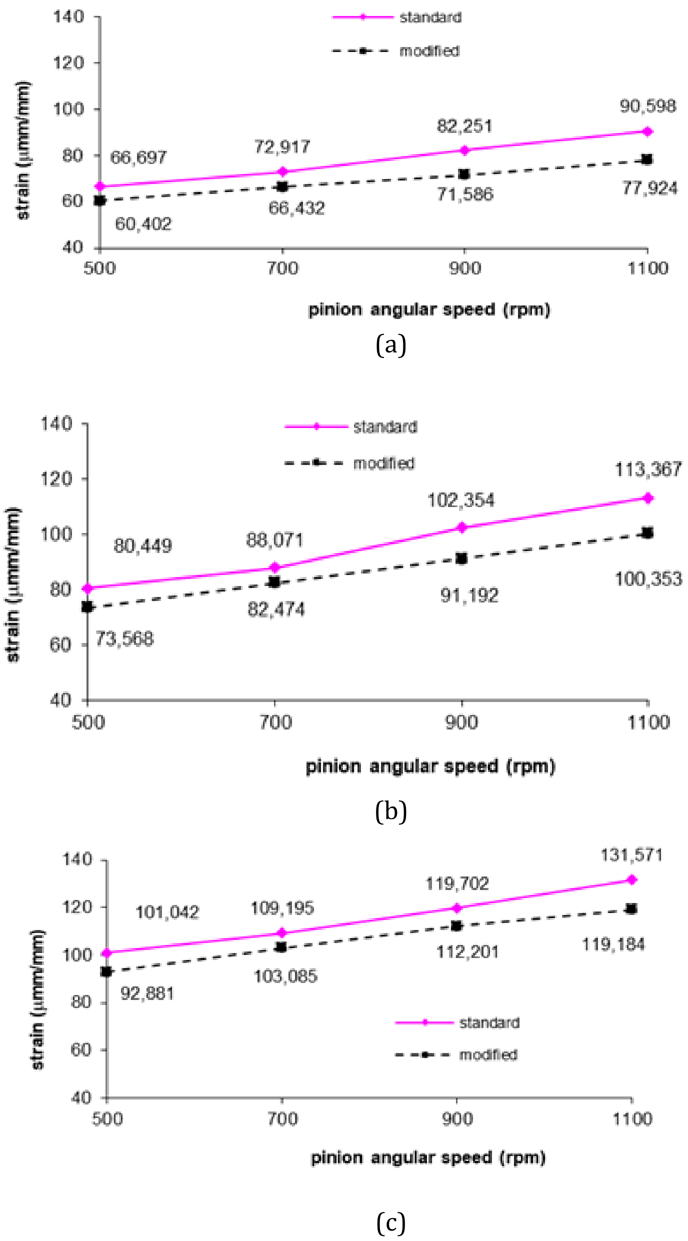


Figure 7 Maximum strain ratio as a function of speed at (a) 65%, (b) 85%, and (c) 100% load

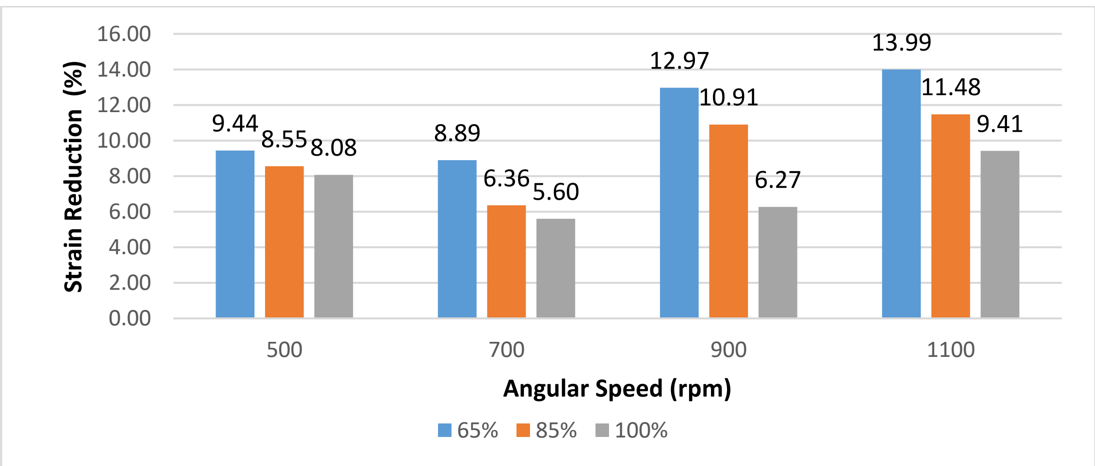


Figure 8 Reduction of strain between standard and modified linear profile

4. Conclusions

In conclusion, several experimental studies determined the effect of tooth profile modification on standard and linearly modified gear profiles by comparing and analyzing the results. The reduction in micro-strain was observed for a duration of 3.04 – 6.67 milliseconds, depending on the rotation speed and torque load. Modifying the linear involute profile significantly reduced tooth leg strain, ranging from 5.6% to 13.99%, depending on the load and speed. The maximum reduction in micro-strain of 13.99% was achieved at a rate of 1100 rpm and a torque load of 65%, increasing tooth strength against bending loads. It is important to note that reducing tooth leg strain is crucial for ensuring the longevity and durability of gears in various real-world scenarios. Further studies could explore the impact of tooth profile modification on other gear types and sizes and investigate the effect of different parameters, such as lubrication, on gear performance. Overall, the findings of this study have a valuable contribution to understanding modified gear profiles and their impact on gear strength and durability.

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