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Effect of Profile Modification on Noise in Involute Gear Pair

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Abstract

Ensuring regular movement and power transmission in gear wheels is possible with a suitable tooth profile. Although involute profiles have a structure that can provide conjugate movement in accordance with the law of gearing, transmission errors that cause vibration and noise occur due to the flexibility of the elements in the gear system under dynamic conditions. Transmission errors occur depending on many parameters from production to the oil and lubrication system. In this study, the vibration and noise state generated during transmission in gears were investigated in relation to geometric parameters. In the reference gear pair assumption, the noise levels were evaluated for the four cases created due to the change of different geometric parameters. The effect of profile modification on vibration and noise was addressed, and optimum profile modification parameters for the reference gear pair were analyzed with KISSsoft commercial software. The effects of geometric parameters and profile modification on noise level are shown in graphs. With the application of the Tip Relief method, a significant decrease in the noise level has been observed.

Keywords: Gear noise, Profile modification, Tip relief, Transmission error, KISSsoft

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1. Introduction

Gears, which have a long history, are cylindrical wheels that are used to transmit motion and power from one rotating shaft to the other with tooth sets with appropriate profiles placed on their surfaces. Motion and power transmission are provided by gears in almost every machine, especially in automotive and aviation. Depending on factors such as working area, transmitted torque, and speed, motion transmission type, operating conditions, different types of gears made of different materials have emerged.

Gear wheels must meet the motion conditions to function properly. The kinematic condition can only be achieved with a suitable tooth profile. The pitch circles of the gears in contact position are tangent to each other and contact at a single point, and this point is called the rolling point. For gear wheels transmitting at a constant speed, the linear velocities must be equal at the rolling point and the common normal drawn from the contact point of the gear wheels at any time must pass through the rolling point. This transmission condition is expressed as the law of gearing. The involute tooth profile is the most common tooth profile that provides conjugate movement.

During the transmission of gear wheels, each tooth is exposed to static and dynamic loading. Movement and power transmission are provided by the force on the line of action that the tooth pair

in contact exerts on each other at the contact points. The applied force creates a friction force between the tooth surfaces sliding on each other, apart from the rolling point. These operating conditions in gear wheels create dynamic loads on tooth profiles that are greater than static loads.

Dynamic loads caused by vibrations at high speeds cause noise [1]. Reducing dynamic loads on gear wheels will increase efficiency, decrease damage and noise [2]. The main causes of dynamic load formation are transitions between tooth contacts, deformation of teeth, shaft, bearing and other elements under load, production and assembly errors [3, 4]. Reducing the dynamic force and noise of the gear system is an important consideration in gear design. Tooth profile modification is one of the most well-known and effective methods applied in the literature to minimize dynamic vibration and noise [5-9]. Profile modification applied to minimize transmission errors is generally deliberate changes made in the involute tooth profile.

Palmer and Fish evaluated the valid theories and experimental studies on tip relief methods applied in the control of transmission errors of spur gears. They developed a calculation method in order to evaluate the effect of the modification to be applied during the design phase by establishing a connection between transmission errors, noise and vibration [10]. Tesfahunegn et al. studied the effects of linear and non-linear profile modification applications on transmission errors and stresses. They stated that profile modification is not independent of shape, and different transmission errors and stresses are observed in linear and non-linear applications [11]. Pleguezuelos et al. investigated the effect of profile modification application on semi-static transmission error in high contact ratio gears. A pre-developed model was used for load distribution and quasi-static transmission error based on the equal delay interval hypothesis on all teeth in synchronous contact. Optimum profile modification dimensions are expressed as a function of contact ratio [12]. Ma and his colleagues examined the effect of type relief on vibrations in the gear pair working under 150 Nm torque, neglecting friction force and tooth cavity. The mesh stiffness varying with time under different amounts of tip relief is calculated according to the finite model. Then, a gear rotor system finite element model is developed by MATLAB, taking into account the time varying mesh stiffness and load torque. It was stated that the tip relief application softens the transfer because it compensates the elastic deformation at the root and balances the transition to single tooth contact[13].

Dhokale et al. Conducted a literature study on the working noise generated in gear wheels and stated in the results section that the main reason for the noise is the vibration caused by transmission errors [14]. Akerblom and Parssinen experimentally investigated the noise and vibration occurring in gears for eleven gear pairs that differ in terms of manufacturing and geometric parameters. In their studies, they concluded that profile modifications made in gears reduce the noise level, surface roughness is an effective factor on noise, and grinding application reduces noise. [4] Petra conducted an optimization study with KISSsoft software, taking into account the contact ratios in order to improve the noise level in helical gear pairs [15]. Sánchez et al. studied the contact ratio gears under load. They investigated the effect of profile modification application on transmission errors and load distribution [16].

2.1. Masuda gear noise equation

Masuda et al. studied the simple and accurate calculation methods of the noise generated during transmission in gears. By adding a dynamic term to the Kato [17] equation, which determines the general noise level using the gear's specification data, a semi-empirical equation has been developed that takes into account the effect of the tooth side surface finishing method. It was stated that the data obtained from the test gears through the experiments were in accordance with the data obtained from the equation with 5 dB tolerance [18]. KISSsoft commercial software calculates the noise level in gear wheels based on the Masuda equation. Kato equation is shown in Eq. (1) and Masuda equation in Eq. (2).

$$L_K = \frac{20(1-tan(\beta/2))\sqrt[8]{u}}{f_v.\sqrt[4]{\varepsilon_a}}.+20.logW \qquad dB(A) \qquad (1)$$

$$L_M = \frac{20\left(1-\tan\left(\frac{\beta}{2}\right)\right)\sqrt[8]{u}}{\sqrt[4]{\varepsilon_a}}.\sqrt{\frac{5,56+\sqrt{v}}{5,56}} + 20.logW + 20.logX + 20 \qquad dB(A) \qquad (2)$$

L is the total noise level from a gearbox in dB (A) at 1 meter. β, helix angle; u, transmission ratio; εα, contact ratio; W, power delivered in hp; v, circumferential velocity in m / s; fv speed factor; X is the vibration displacement amplitude normalized by static deflection.

2. Effect of geometric parameters on noise

Vibration and noise generation in gears can be significantly reduced by selecting appropriate design parameters and modifying the tooth profile. The effect of geometric parameters on noise generation is analyzed in this part of the study. A spur gear pair that has a 150 mm distances between the axes, a face width of 34 mm, 20° pressure angle, a 3 mm module, and quality grade 6 was taken as reference. Input conditions are accepted as 500 Nm torque and 4000 rpm speed the selected parameters of the reference gear were included in the solutions within the specified range and their relationship with noise was investigated. The cases created due to the change of pressure angle, helix angle, tooth quality, and tooth width parameters are given in Table 1.

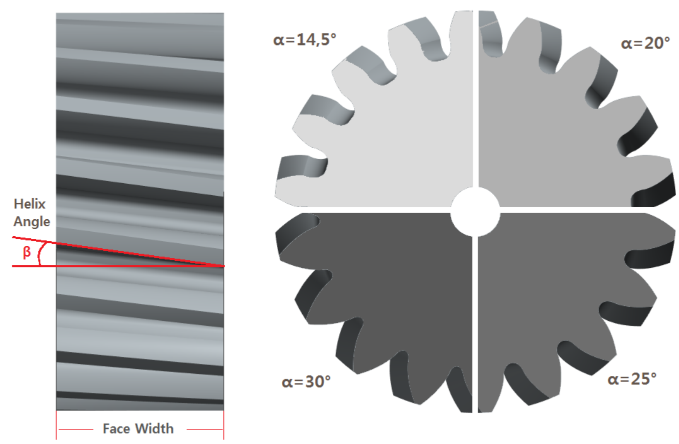

Fig.1. Geometrical design parameters

Table 1. Parameters of case studies

	Case 1	Case 2	Case 3	Case 4
Module	3 mm	3 mm	3 mm	3 mm
Pressure angle	*14,5°-30°*	20°	20°	20°
Helix angle	0°	*0°-30°*	0°	0°
Quality [ISO-1328]	6	6	*1 - 12*	6
Face width	34 mm	34 mm	34 mm	*26-42 mm*
Center distance	150 mm	150 mm	150 mm	150 mm
Pinyon Number of teeth	24	24	24	24
Wheel Number of teeth	74	74	74	74

Case 1

The change of the values obtained in the solution made in the range of maximum and minimum pressure angles used in gear wheel design according to the profile contact ratio is shown in Fig. 2. Increasing the pressure angle decreases the contact ratio of the tooth profiles and causes an increase in the noise level. The pressure angle in gear design is most commonly used as $\alpha = 20°$. In some special cases where tooth root strength is more important, the pressure angle is chosen larger than 20° to produce gear without undercuts and to increase the number of limiting teeth [19].

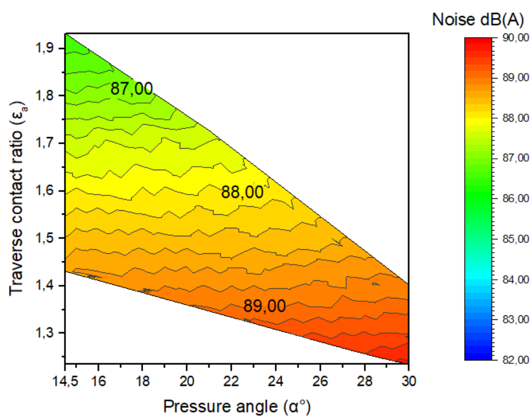

Fig.2. The change of the noise level due to the pressure angle change

Case 2

The effect on noise due to the change of the helix angle in the reference gear in the range of 0°-30° is discussed in this section. The data obtained as a result of the solution are shown in Figure 4. The determination of the helix angle results in a helix contact ratio in addition to the profile contact ratio. The number of teeth in constant contact during load transfer defines the contact ratio. When the transmission is evaluated in two dimensions, the load is not constantly shared by the two gear teeth under the limit condition of the engagement ratio X <2, a part of the transmission is provided by a single tooth. This situation is shown in Figure 3 by specifying the contact areas and limits of the tooth in the transmission. Due to the transitions between regions and the increasing load, deflection in the single tooth area causes noise formation.

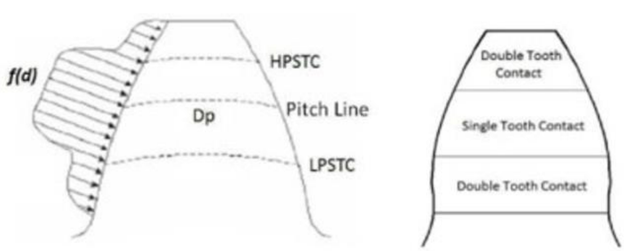

Fig.3. Tooth-contact areas and limits [20]

As the helix angle increases, the amount of helix engagement starting from the tip of the tooth and continuing along the surface increases along the width of the tooth. The gear tooth surface, which comes into contact gradually, makes the contact transitions smooth with the increase of the helix angle. Figure 4 shows the change in the total contact ratio and the noise level depending on the change of the helix angle. With the increase of the helix angle, the helix contact ratio and the total contact ratio increase accordingly. Since the load is shared by more than one tooth, it reduces the deflection in the tooth profile, resulting in reduced vibration. The reduction of dynamic transmission errors also reduces the noise.

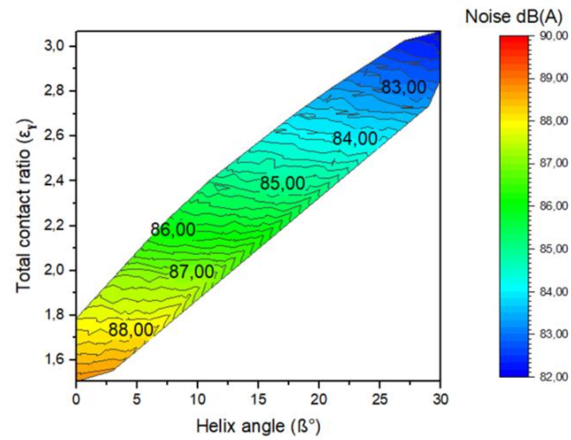

Fig.4. Variation of the noise level with the helix angle

Case 3

Face width affects the load-bearing capability of gears, load distribution across the width and operating ability. While the increase in the face width increases the load capacity, it causes the load distribution to become uneven due to manufacturing errors or shaft deformations [21]. In Table 2, noise values are listed for the solution made with the limit condition of ± 8 mm from the reference gear face width. Face width has a directly proportional effect on noise. The increase in face width also increases the noise level.

Table 2. Noise level according to tooth width

Face width [mm]	Noise [dB(A)]
26	89,40
28	89,44
30	89,51
32	89,52
34	89,61
36	89,69
38	89,73
40	89,86
42	89,93

Case 4

The 1-11 quality was created according to the ISO-1328-1: 2013 standard and the associated noise levels are shown in Table 3. Processing with low tolerances in gear production reduces the margin of error, therefore transmission error in gears and related noise levels are reduced. The quality group with the lowest tolerances is determined as 1 and the highest 11[22].

Table 3. Noise level according to tooth quality

Quality [ISO-1328]	Noise [dB(A)]
1	88,92
2	89,07
3	89,13
4	89,23
5	89,36
6	89,61
7	89,86
8	90,27
9	90,81
10	91,57
11	92,69

3. Noise optimization with profile modification

Profile modification is a common method applied to reduce the sudden change of static transmission errors that cause a stimulating effect on the system under dynamic operating conditions. Load amplification is reduced by limiting the impact effect by profile modification during transitions between contacts. It is the process of removing material from the root or tooth tip. Since the effects of the treatment applied on the tip and the root are the same, modifications made at the tip are common to avoid a reduction in stiffness. The process is applied linearly and parabolically. Fig. 5 shows the profile modification applied to the tip of gears. "C" refers to the tip relief in µm. "L" is the width factor [23].

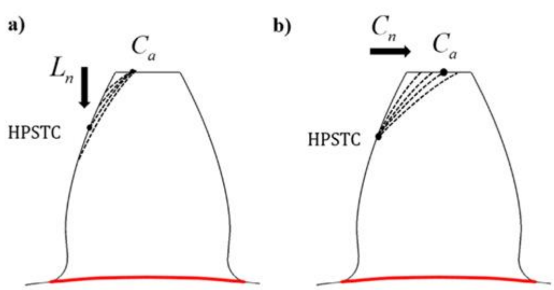

Fig.5. Profile modification [24]

Linear profile modification in the tooth tip area was applied in four stages with KISSsoft commercial software according to the solution intervals given in Table 4. In order to determine the optimum region, the intervals were kept wide at first. The value range was solved in 6 steps in each stage, 1297 variations were obtained and the determined optimal regions formed the solution area of the next stage. With this strategy, the solution areas were gradually narrowed and the solution was brought closer to the optimum. Optimal modification values applied for wheel and pinion are shown in the conclusion section. In the second stage, value ranges were applied separately for pinion and wheel, and the 400 solutions obtained are shown graphically in Fig. 6 and Fig. 7. In both graphs, dark blue areas represent the region where transmission errors and related noise levels are the lowest. The relationship of transmission errors with vibration and noise is shown in Fig. 8.

Table 4. Profile modification solution range

	Pinion		Wheel	
No	Cαa [µm]	LCa	Cαa [µm]	LCa
1	0-100	0-3	0-100	0-3
2	10-30	1-2	20-40	1,25-2,25
3	15-20	1,1-1,7	30-35	1,6-2
4	16-17	1,4-1,6	33-34	1,75-1,95
Re-sult	16,8103	1,533	33,8421	1,8556

Fig. 6 shows the change of the peak to peak transmission error according to the parametric values for the tip relief applied for the wheel. It is undesirable for the contact points of the tooth profiles to form outside the line of action during the motion and power transmission of the gears. Theoretically, it is not possible to transfer the motion over this line continuously under dynamic conditions due to tooth deflection. Appropriate profile modifications from the tip and the root are important for the continuity of the transmission over the line of action. The range of 1.7-1.9 for the width factor and the tip relief range of 30-35 µm indicate the appropriate modification range for the reference gear for the specified operating conditions. The peak to peak transmission error value expressed with the color scale corresponds to the low values in the specified range and is expressed in dark blue in the graph.

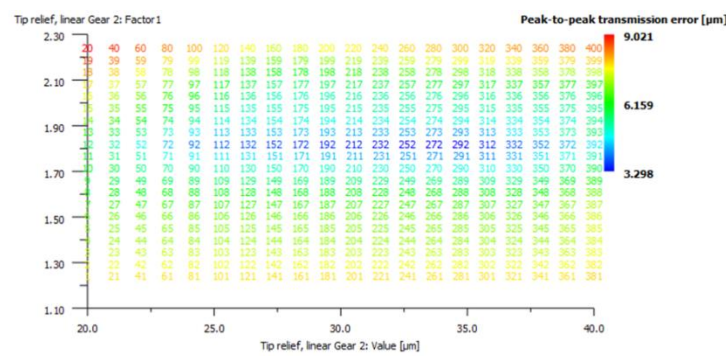

Fig.6. Transmission error according to profile modification variables for wheel

The method applied for the gear is similarly applied for the pinion. Fig. 7 shows the change of the peak to peak transmission error according to the parametric values for the tip relief applied for the pinion. The range of 1.1-1.7 for the length factor and the tip relief range of 15-20 µm indicate the dark blue region where transmission errors are minimal. The boundary conditions for the next solution were obtained by determining the appropriate intervals in each solution step for the gear and pinion. With this method, the solution area was narrowed and the geometric values of the profile modification that could be applied were brought closer to optimum.

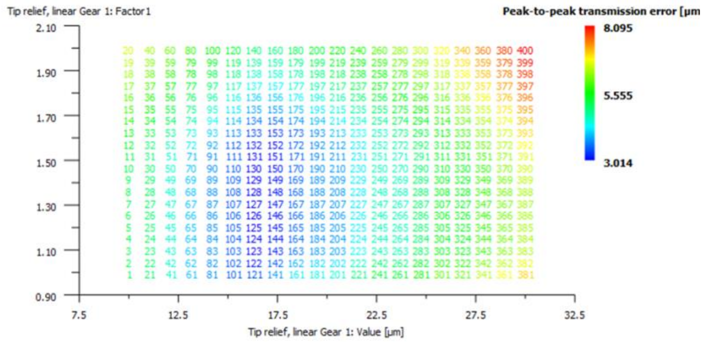

Fig.7. Transmission error according to profile modification variables for pinion

Based on the conjugate motion complying with the law of gearing, the tooth pairs in contact state apply force to each other in the line of action axis. In addition, the transmission ratio must remain constant throughout the operation. It is one of the design objectives that gears are able to meet these requirements under static conditions as well as under dynamic operating conditions. The flexibility of the system elements in dynamic operating situations in gears disrupts these conditions and creates transmission errors. Transmission errors are defined as the angular deviation of the drive gear with respect to the driven gear and the displacement of the associated contact point on the line of action. The difference between the maximum and minimum values of the transmission error in dynamic operating conditions refers to the peak-to-peak transmission error. In Fig. 8, the amplitude of the first harmonic of the vibration due to the change in the peak-to-peak transmission error is given. The inability of the gears to perform the conjugate motion during the transmission and the deviation of the transmission over the line of action creates an excitation force that causes a vibration which also depends on the tooth stiffness. In dynamic operating conditions, the transmission is provided through only one tooth in the time between a tooth coming out of the engagement and a new tooth entering the engagement depending on the contact ratio. The stiffness of these load transitions in transmission is related to the excitation force. As a result of the increase in transmission errors in the graph, the increased excitation force triggered the vibration and increased the noise.

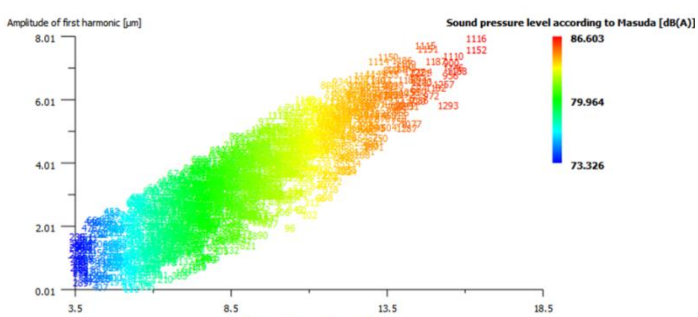

Fig.8. Vibration and noise level due to transmission errors

In Fig. 9. Red line refers to the profile without tip relief, and the blue dashed line shows the modified profile.

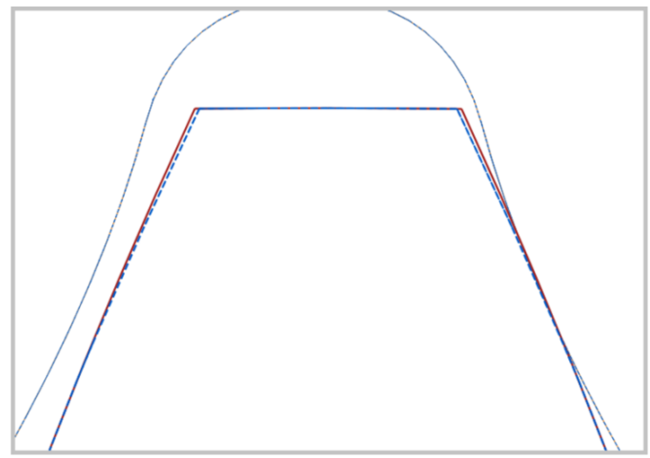

Fig.9. Wheel profile modification

3. Conclusion

In this study, noise analysis was made for four different cases created by changing geometric parameters in the reference gear pair. Also, the effect of profile modification on noise was investigated by applying a linear relief process to the tooth tip. The effects of the change of the geometric parameters on noise are shown in graphics by correlating with the contact ratio. Since the increase in the contact ratio reduces transmission errors, it lowers the noise level.

In the first case, the variation of the noise depending on the pressure angle was examined. Since the increase in the pressure angle reduces the line of action length and the contact ratio decreases accordingly, the noise increases.

In the second case, the gear was transformed into a helical gear, resulting in an additional helical contact ratio to the profile contact ratio. Since the transition between contacts occurs more smoothly in gears, the noise decreased.

In the third case, the noise conditions of gears were evaluated depending on the face width. Since the increase in the gear width will enhance the effect of manufacturing errors, the noise increased.

In the fourth case, the relationship between the manufacturing quality and the noise is examined. It is known that the noise increases with the decrease in quality.

Profile modification constitutes the largest part of the study. To determine the optimal tip relief parameters for the reference gear, a four-stage solution was implemented, and the optimal solution area obtained from each solution was used as the input data of the subsequent solution. With the gradual approach, the solution area was narrowed and the values that make the noise level minimum were determined. As a result of the solutions, it was found out that the sound level was minimum at 16.8103 mm tip relief and 1.533 width factor for the pinion, 33.8421 tip relief, and 1.8556 width factor for the wheel. The effects of geometric parameters and profile modification on noise were compared. For the gear with quality level 1, pressure angle 14.5°, helix angle 30°, face width 26 mm, the minimum noise level is found to be 81.5 dB (A), while the tip relief applied to the reference gear alone reduced this value to a level of 71.57 dB (A).

Conflict of Interest Statement

The authors declare that there is no conflict of interest.

CRediT Author Statement

Regaip Menküç: Conceptualization, Methodology, Software, Writing-original draft, Visualization,
Latif Kasım Uysal: Writing-original draft, Visualization, Writing-review & editing,
Tolga Topgül: Supervision, Project administration.

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