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Original Article

An Approach for Narrowing the Method of Selection of Profile Shift Coefficient for A Given Helical Gear Pair

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***Abstract** - Profile shift or Addendum modification is provided in gear teeth to avoid undercut, improve strength and running properties or adjust the centre distance. Along with this, it is generally recommended for gears with a critical number of teeth, non-standard entre distance, obtain balances specific sliding or obtain reduction in sizes. However, there is no specific ready data that advises how to use it and what value can be opted. This research work is performed to investigate the effect of addendum modification on various parameters like contact ratio, tooth root strength, specific sliding, and undercutting. A pair of helical gears with 24 teeth and 97 teeth with 20-degree pressure angle and 13.4022-degree helix angle is considered for study purposes. The results are compared with the kissSoft data, and charts are prepared to establish the selection of addendum modification parameters for both gear and pinion. As an output of this research work, a graph is produced to help in narrowing the selection of profile shift coefficient with an excel tool for performing various functions. This work was specific to the sample case, whereas the same methodology can be implemented for other pairs of gears.*

***Keywords** - Addendum modification, KissSoft, Specific sliding, Undercut.*

1. Introduction

Profile shift plays a crucial role in gear geometry and its working. There are various research work available which shows the selection of profile shifts based on centre distance sliding speed. Some are defining these values based on load-carrying capacity. However, none of the research work is actually showing a method to collaborate al the necessary parameters and find a suitable profile shift which can meet all design and geometrical constraint criteria. This research work provides a noble method to find out the profile shift based on consideration of design and geometrical parameters in one go. To proceed with this, it is important to understand the basics of profile shift. A hob is, in effect, a rack cutter. The meshing process cutter and the workpiece are discussed here. For understanding, the datum line can be considered as the cutter centrode, and the reference diameter of the gear can be considered as its centrode such that both have having same pitch. The position of the gear cutter is shown in Figure 1, with various possible positions.

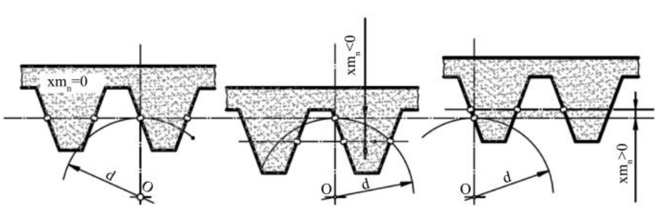


Fig. 1 Position of rack cutter [16]

In the left figure of Figure 1, the datum line of the cutter is shown as tangent to the reference diameter of the gear. This means the datum line is rolling over the pitch circle diameter of the gear. This is known as nominal position and the gears generated by this method have no profile shift coefficient ($x = 0$) and are also known as null gears. This action does not affect gear tooth thickness.

In the middle of Figure 1, the datum line of the cutter is shown as offset to the reference diameter of the gear such that it is interfering with the reference diameter. Such shifting of the cutter is known as negative profile shift in gear terminology, and the equivalent profile shift coefficient is negative ($x < 0$) in such case. This action reduces the tooth thickness.

On the right side of Figure 1, the datum line of the cutter is shown as offset to the reference diameter of the gear such that it is not interfering with the reference diameter and away from it. Such shifting of the cutter is known as a positive profile shift in gear terminology, and the equivalent profile shift coefficient is negative ($x > 0$) in such a case. The positive profile shift increases the tooth thickness.

A mesh version of the cutter with gear is shown in Figure 2. So, in general, to define the geometry of involute gear, the parameters required to be considered: number of teeth (z), profile shift coefficient (x), and parameter of basic tooth profile (pressure angle αn, module mn, factors ha, and c), the non-dimensional magnitude of tip radius ρko of the tool.

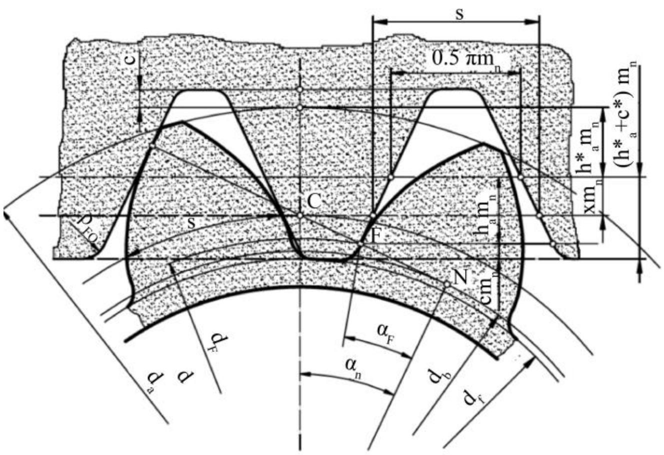


Fig. 2 Mesh of rack cutter [16]

This research work is about how to select the profile shift value and what parameters need to be considered. To investigate this to the in-depth level Literature review is presented in the next section.

2. Literature Review

To understand the profile shift and its effect more closely, various literature and available research materials were deeply studied. The details of the few critical papers which has really helped in the learning are detailed below:

M. Natraj, S. Sankar, and M.S. Raj [1] worked on profile modification for increasing the tooth strength in spur gear using CAD. This research work was mainly related to spur gear tooth failure. Gears with undercutting issues are addressed here such that if the root radius profile can be modified, then the tooth root failure issue can be resolved to a certain extent. The trochoidal root fillet concept was proposed here rather than the circular root fillet. To confirm further the outcome and its comparison ANSYS version 11 was used for CAE analysis of tooth geometry. The conclusion of this research work was lower stress value in root reason with trochoidal curve than circular curve thus helping in reducing the tooth fracture issue.

Philips D. Rockwell [2] worked on Profile Shift in External Parallel-Axis Cylindrical Involute gears. This research was extremely helpful in understanding the effect of profile shifts on various performance parameters. The discussion was there related to its profile shift effect on balances specific sliding, balanced bending fatigue life, undercut, and gear tip thickness. This work also advised the range for profile shift selection of gears such as $-0.5 \leq x \leq 1.0$. This paper also provides a detailed way of calculating the tooth parameters, including the backlash effect. Overall this paper has disclosed various performance parameters getting affected due to profile shift change.

Gunay et al. [3], worked on The effects of the Addendum modification coefficient on tooth stresses of spur gear. This research work was about the investigation of stresses in tooth roots of spur gears because of Profile shift. The finite element method is used to investigate the tooth root stress for various values of profile shift with both positive and negative. Four distinct values of profile shift are considered as -0.5, -0.3, 0.3 and 0.5. Various results were plotted to find the effect of profile shift. It was observed that with an increase in profile shift, the tooth root stress was improving, whereas the tooth tip thickness was reducing simultaneously. Undercutting was also discussed which could be reduced or improved by considering suitable profile shift value.

Gultekin Karadere Ilhan Yilmaz [4] worked on the Investigation of the effects of profile shift in Helical gear mechanisms with Analytical and Numerical methods. This research w ork numerical and analytical approaches for calculation for investigating the profiler shift effect in helical gears. The idea was to compare both methodologies, Numerical and Analytical, for a given set of examples. Excel sheet was used to calculate analytically by using various profile shift values, whereas Ansys software was used for the Numerical solution. It was concluded that the effect of profile shift is more on root stresses as compared to contact stress. It was also concluded that the negative value of profile shift has a detrimental effect on the tooth root stress as compared to positive profile shift.

Baglioni et al. [5] worked on the influence of the addendum modification on spur gear efficiency. In this research work, the efficiency of the gear pair is analyzed with an approach of friction coefficient reduction by using Niemann's approach and past research work. In order to vary the friction, the addendum coefficient value was varied to get different friction values by changing the tooth profile. Various charts were plotted to verify the outcome. It was found that with an increase in profile shift coefficient as a total, the efficiency of the system also increases.

Zoltan Tomori [6] worked on An optimal choice of Profile shift coefficient for spur gears. This research work targeted a different approach to selecting the profile shift values for a gear pair. The paper initiates the discussion of stress in teeth, considering the gear ratio of 1 and summation of profile shift. Further, in the next section, the limiting values of profile shift are discussed with undercut consideration. Discussion about pitting was also there to correlate the profile shift with frictional effect and surface related failures.

B. Samya, M. Boudi, A. Bachir,Y. Amadane [7] considered the bending stress and contact stress as the measuring parameters to know how profile shift factors are affecting it. Three different profile shift coefficient values are taken as 0, 0.1, and 0.2. Analytical calculations were done as per ISO standards and Lewis's theory concept. Later, the results were cross-verified in Ansys software by using the finite element method. The tooth form factor and tooth tress factor played an important role in the analysis of the tooth root stress to know the effect of profile shifts on these parameters. Various charts were plotted with respect to the contact ratio load sharing factor, with profile shift as the x-axis, to know the characteristics curve. This research work was performed for a sample spur gear set. It was concluded that the higher the profile shift factor, the lower the chances of spur gear root failure and contact failure. Both analytical and FEA results were in good agreement for this.

M. Geberemariam, Ashish Thakur, E. Leake, and Daniel Tilahun [8] worked on the Effect of change of the contact ratio on contact fatigue stress of involute spur gears. This research work was related to the contact stress of spur gears in relation to the contact ratio. Six different sets of contact ratios were considered, starting from 1.6 to 2. Critical points and scenarios were examined for each contact ratio to calculate the contact stress. CATIA and Solid Works software were used to develop the gears, and then Finite element analysis was done in ANSYS Workbench. It was concluded that stress values were changing with the change in contact ratio, and the relationship between them was observed as linear. With the help of the contact ratio the load sharing factors were also plotted.

Wang et al. [9] worked on the performance of spur gear with respect to the profile shift modification factor. A time-varying mesh stiffness model is prepared in this research with respect to varying profile shifts. A change in profile shift may reduce the mesh stiffness of the gear pair. It was found that for individual shifts, a positive profile shift reduces the gear stiffness. This increases the tooth thickness to increase the fatigue life.

Rajesh et al. [10] worked on the optimization of profile shift co-efficient for the highest contact ratio in non-standard gearing. This research work deals with the effect of a high contact ratio on the load-sharing factor. Nonstandard gears here mean the gears have profile shift. The work also detailed various options of profile shift selection as zero, negative and positive. The effect of profile shift is also corelated along with the tooth summation. Various values of profile shift co-efficient are considered, ranging from -1.250 to -0.250. The value of tooth summation was modified to get the highest contact ratio value.

Ali Raad Hassan [11], the computer program developed to plot spur gear tooth profile with respect to various geometrical parameters like base circle, root circle, pressure angle and profile shifts. The influence of the cutter data is also considered in this work by the author. It was found that the program software was very useful in generating various spur gear curves for different parameters. These geometries were further analyzed in ANSYS software for modal and natural frequency analysis. Various curves were drawn to identify the nature of profile shift and pressure angle variation with natural frequencies.

Pedrero and Artes [12] worked on the development of the analytical method for the estimation of profile shift coefficient value, which can be used with all Pressure angles and addendum values. The method developed did not need any hectic iteration cycle, which resulted in a highly efficient speed computer program. Equations for the specific sliding were generated from a very basic form by using the relationship diagram. In terms of geometrical values, the condition of balanced specific sliding was derived. Three different groups of gears with geometrical parameters were considered for reviewing the formula. Addendum modification curves were generated for the three different cases with respect to the no. of teeth of gears with different pressure angles.

J. I. Pedrero, M. Artes, and J.C. Garcia [13] worked on the Determination of the addendum modification factors for gears with pre-established contact ratio. The authors have worked on approximation of the equation for the determination of the relationship between the gear and pinion addendum modification factor. Specific sliding condition is also used here in this analysis. The formula developed was free of iteration thus saving much time. The developed formulae were too user-friendly for analytical hand calculation. These were developed from very basic geometrical figures to the desired form. Various curves were generated to determine the effectivity of the developed formulae with respect to addendum modification coefficient, no of teeth and various speed ratios. The equation was found to be compatible with every combination of pressure angle, tooth combination and contact ratios.

Sachidananda et al. [14] worked on Sliding velocity in profile-corrected gears. The researcher studied the sliding velocity for a defined gear pair having 100 no of the tooth having a specific center distance. Contact stress, along with specific speed values, were calculated for the same pair for various profile shift coefficients. The process started with an estimation of contact stress, contact area, sliding speed and the multiple of Contact pressure with sliding speed. Sliding speeds were calculated at various points along the contact path. Various curves were plotted to know the relation of pressure angle with respect to the sliding speed contact stress. It was concluded that profile shift changes the gear geometry and sliding speed.

Abderazek et al. [15] worked on the algorithm of tooth profile optimization with respect to balancing specific sliding coefficients of involute gears. The author discussed the various charts available in previous literature and standards. Here conclusion was that the charts and graphs sometimes difficult to predict the exact intermediate values. The researcher proposed a differential algorithm to resolve the profile shift coefficient selection. This algorithm was related to an optimization technique developed for exact, specific, balanced sliding conditions. A flowchart was produced during the research activity.

Wang et al. [9] performed the research on time-varying mesh stiffness and dynamic characteristics of gears with respect to profile shift. The author discussed S, and So gearing. A case study was also performed, and various charts were produced to observe the effect of profile shift.

3. The Outcome of the Literature Survey

Past works have given a very good picture to understand the effect of various profile shift values in relation to the operating function of the gear pair. The finding is detailed below:

- Profile shifts have a considerable effect on gear tooth root strength.
- Profile shifts need to be selected in accordance with tip thickness.
- Specific sliding has an important relation with profile shift value.
- Addendum coefficient values as a sum of gear pairs have a good relationship.
- Sliding speed is directly related to the profile shift coefficient.
- The tooth stiffness value is also dependent on the tooth profile shift factor.
- There is no study material available which advises directly how to select a profile shift factor for a gear pair.

4. Gap Found in the Literature Review

Most of the work was done by considering some parameters to derive the required profile shift. However, the methods did not give a holistic view of how to define a profile shift, which can consider all the necessary parameters like sliding velocity, tip thickness, safety factor, and contact ratio altogether in one go. Present research work has considered all these parameters and provided a way to find a profile shift which can meet all criteria.

4.1. Parameters for Detailed Work

To do the analysis more deeply, undercutting, tooth root stress, safety factors, specific sliding parameters, and tooth tip thickness parameter needs to be studied. The sample example to be studied will be as per Table 1.

4.2. Software used in this Research

KissSoft 2020 [17] is used for gear and shaft calculation; the Excel tool is developed locally to perform various calculations analytically, as mentioned in this paper.

Table 1. Subject case

Parameter	Pinion	Gear
No of teeth	24	97
Normal Module	1	
Helix Angle (degree)	13-24'-8"	
Pressure Angle(degree)	20	
RPM	160	39.6
Material-Case hardened and Ground	20MnCr5	

4.3. Tooth Tip Thickness

Gear tip thickness is given by the relationship as shown below: [18]

$$S_{ta} = d_a\left(\frac{\pi}{2z} + \frac{2x\tan(\alpha_n)}{z} + inv\alpha_t - inv\alpha_{ta}\right) \quad (1)$$

Where,

$$inv\alpha_t = \tan(\alpha_t) - \left(\alpha_t * \frac{\pi}{180}\right)$$
$$inv\alpha_{ta} = \tan(\alpha_{ta}) - \left(\alpha_{ta} * \frac{\pi}{180}\right)$$
$$S_{na} = S_{ta} * Cos(\beta_a) \quad (2)$$

4.4. Tooth Root Undercutting

Tooth flanks are broadly classified into two parts- the involute and the non-involute part. The involute part of the tooth flank is in between the tip circle diameter and the limiting point F, as shown in Figure 3.

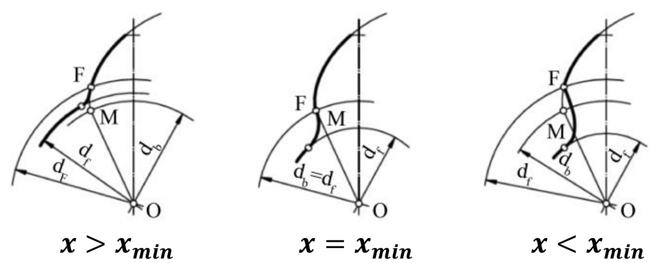


Fig. 3 Involute profile range for different values of x [16]

At point F, the Pressure angle is shown as:

$$\tan(\alpha_F) = \tan(\alpha_t) - 4 * \frac{1+c-x}{z_n * sin2\alpha_t} \quad (3)$$

α_F need to be zero to have the minimum value to avoid any undercut., so the minimum value of the profile shift can be derived from Equation -3 as below:

$$x_{min} = h_a - \frac{z*(sin\alpha_t)^2}{2*Cos\beta} \quad (4)$$

Figure 4 shows the variation of tip thickness at various profile shift values. As per Equation 4, the minimum value of profile shift can be considered as 0.4, whereas as per Equation 2, the tip thickness value can be evaluated, and for a minimum desired value of 0.4 times the normal module as per global standard practice, the possible applicable range looks like to be in between -0.4 to 0.7 value.

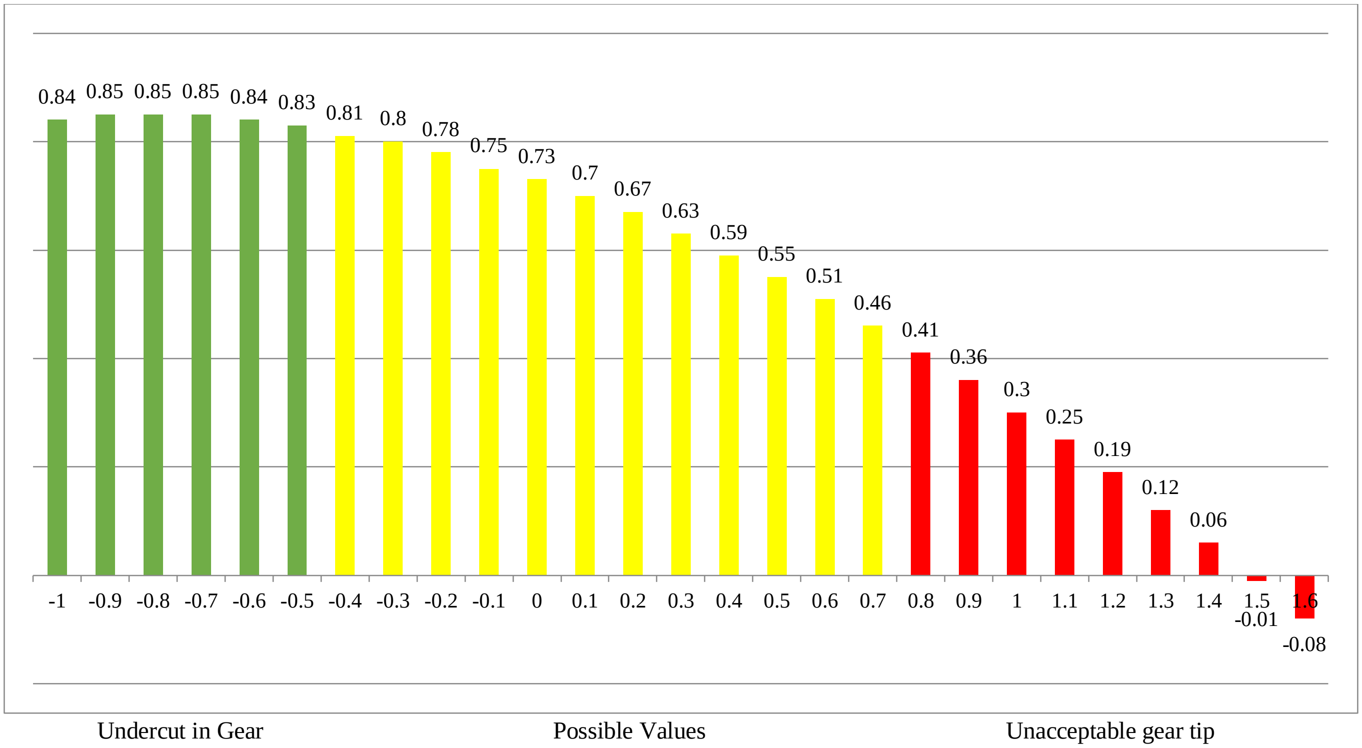


Fig. 4 Tip thickness for 24 teeth

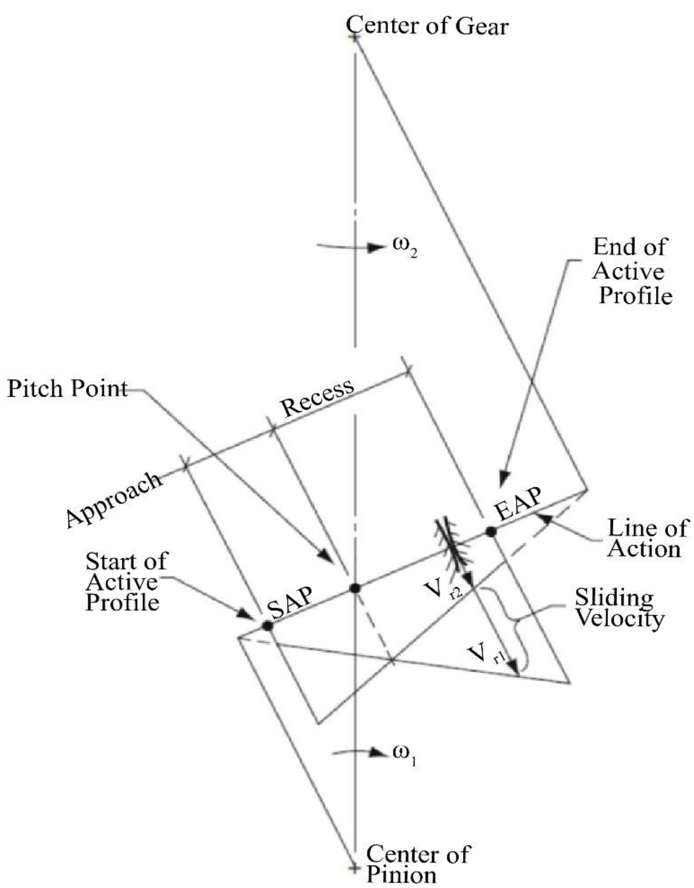


Fig. 5 Sliding velocity along the contact length [19]

Table 2. Calculated values example

Location	ρ1 (mm)	ρ2 (mm)	Vr1 (m/min)	Vr2 (m/min)	Vs1 (m/min)	Vs2 (m/min)	VSS1	VSS2	Roll Angle (Radians)
A	2.39	19.405	0.04	0.08	0.08	-0.04	-1.00	0.207	0.207
B	4.198	17.597	0.07	0.073	0.073	0.00	-0.04	0.363	0.363
C	4.323	17.472	0.072	0.072	0.072	0.00	0.00	0.374	0.374
D	5.415	16.38	0.091	0.068	0.068	-0.02	0.25	0.469	0.469
E	7.223	14.572	0.121	0.06	0.06	-0.06	0.50	0.625	0.625

4.5. Specific Sliding

Gear tooth pair acts against each other with a combination of rolling and sliding velocities. The motion is pure rolling at the pitch point, whereas, as the contact point moves away from the pitch point, the motion between the pair is a combination of rolling and sliding. Sliding velocity plays a critical role in terms of surface-related failures. It is important to know its magnitude over the entire contact length at various points which will start from the start of the active profile to the end of the active profile. Rolling velocity at any point is calculated as follows:

$$V_{r1(i)} = \frac{\omega_1 * \rho_{1(i)}}{1000} \qquad (5)$$

$$V_{r2(i)} = \frac{\omega_2 * \rho_{2(i)}}{1000} \qquad (6)$$

The sliding velocities at different points can be given as follows:

$$v_{s(i)} = v_{r1(i)} - v_{r2(i)} \qquad (7)$$

$$v_{s2} = -v_{s1} \qquad (8)$$

Specific sliding at any point is calculated as follows:

$$v_{ss1} = \frac{v_{s1}}{v_{r1}} \qquad (9)$$

$$v_{ss2} = \frac{v_{s2}}{v_{r2}} \qquad (10)$$

An Excel tool was prepared to calculate the required values as per Equations 7,8,9 and 10. The values are tabulated for a profile shift value of +/- 0.3. The positive value is considered for pinion as a negative value reduces the tooth thickness.

4.6. Profile Shift Factor Effect on Contact Ratio

To ensure smooth meshing in a gear pair, a contact ratio value of 1.6 as the minimum value is generally accepted industry-wise. It can be reduced to 1.4 but in very specific and uncontrolled conditions. For acceptance value in this research work contact ratio of 1.6 value is considered as the minimum limit.

The contact ratio for the gear pair can be calculated by Equations 11,12 and 13 as below:

$$g_\alpha = 0.5\left(\sqrt{d_{a1}^2 - d_{b1}^2} + \sqrt{d_{a2}^2 - d_{b2}^2}\right) - a sin\alpha_{\omega t} \qquad (11)$$

$$p_{et} = \frac{\pi d_b}{z} \qquad (12)$$

$$\varepsilon_\alpha = \frac{g\alpha}{p_{et}} \qquad (13)$$

The previous section identified the range of profile shift coefficient values from -0.4 to 0.7. The negative profile shift coefficient causes the reduction of tooth thickness in the gears. It is not recommended for pinion until and unless it is compulsion. By using the value of 0 to 0.7, the different values of the contact ratio are plotted in Figure 6.

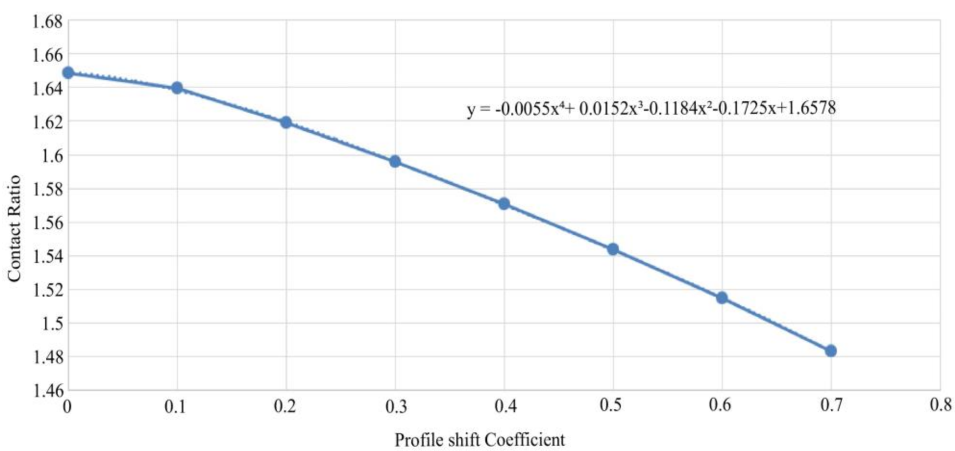


Fig. 6 Variation in contact ratio

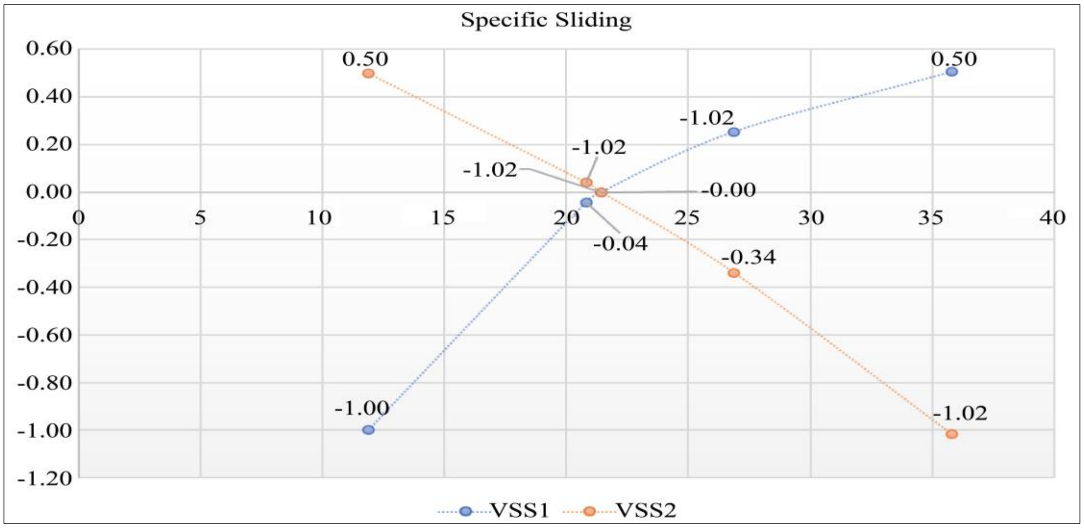


Fig. 7 Specific sliding for the subject gear pair

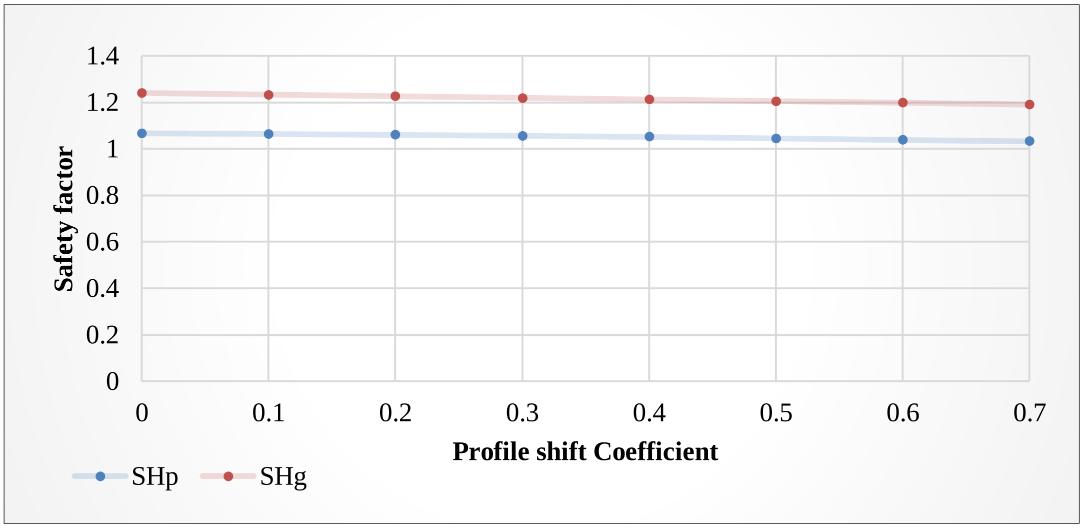


Fig. 8 Tooth stress contact safety factor

The curve equation is also shown in the Figure. By solving the equation for 1.6 contact ratio value the possible value of profile shift comes to be 0.3. By considering the same value, the specific sliding is calculated for the given gear pair the result is plotted in Figure 7. This shows the balanced specific sliding.

4.7. Profile Shift Factor Effect on Tooth Stress Service Factor

ISO:6336 [20] is used for the calculation of tooth root stress and contact stress service factor. In order to calculate the service factors, Equations 14 and 15 are used as shown below:

$$S_H = \frac{\sigma_{Hlim}*Z_N*Z_L*Z_R*Z_V*Z_W*Z_X}{\sigma_{He}} \geq S_{Hmin} \qquad (14)$$

$$S_F = \frac{\sigma_{Flim}*Y_{ST}*Y_N*Y_\delta*Y_R*Y_X}{\sigma_{Fe}} \geq S_{Fmin} \qquad (15)$$

Where,σ_{He} and σ_{Fe} are equivalent stress of constant amplitude for contact stress and tooth root stress. Figure 8 shows the respective safety factors for Contact stress. The chart values are calculated with respect to various profile shift coefficients ranging from 0 to 0.7. Figure 9 shows the safety factor for the given gear pair for root stress. These are also plotted against the value of various profile shift coefficients.

4.8. Chart for the Selection of Profile Shift Coefficients

Based on the previous learning profile shift coefficient selection graph is prepared based on the tip thickness, root stress, contact stress and contact ratio.

In Figure 10, the hatch region is the best location from which the profile shift value shall be selected for the gear pair of this research. The range is from -0.2 to 0.3 to meet all the requirements of global design standards.

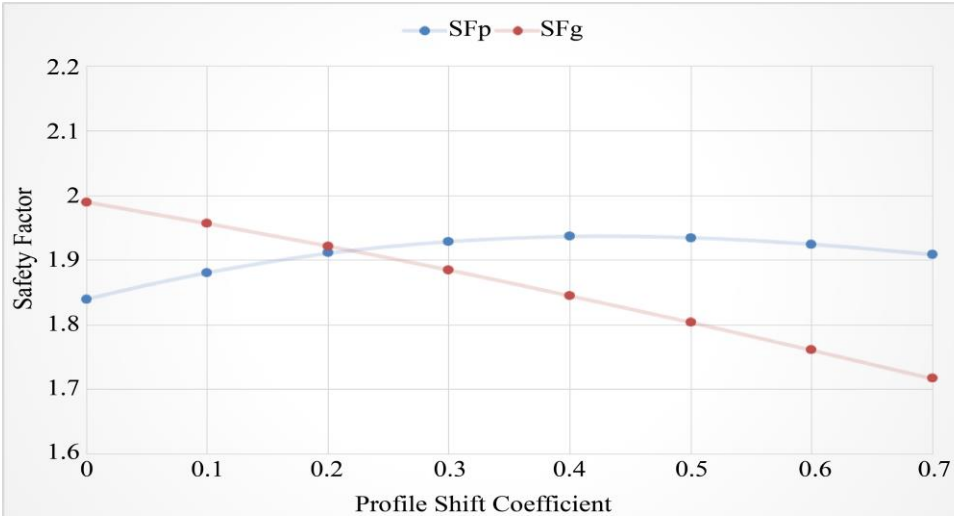


Fig. 9 Tooth stress bending safety factor

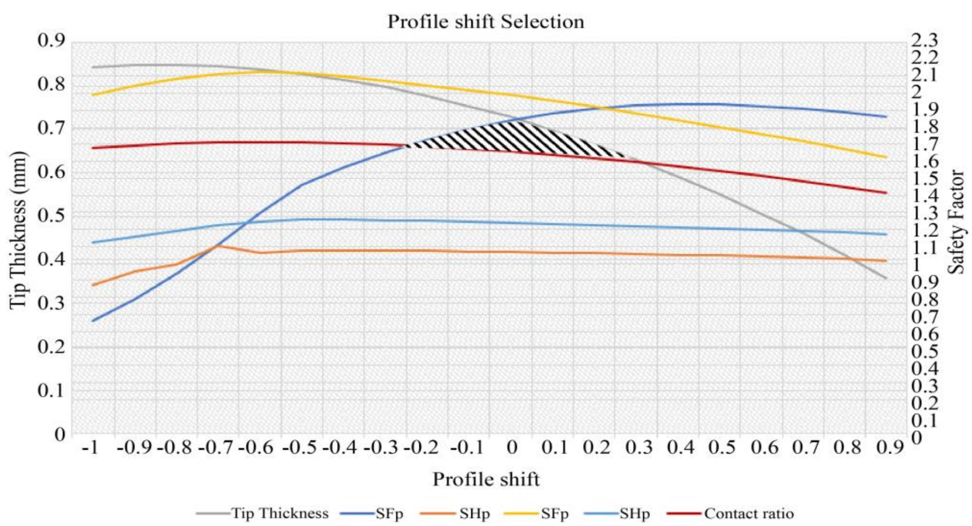


Fig. 10 Profile shift selection chart

4.9. Results and Finding

In this section by considering the range of profile shift coefficient from -0.2 to 0.3 is used to derive the results in KissSoft software and they are compared with the methodologies advised above.

5. Contact Ratio Determination

Figure 11 shows the plot of the result of Kisssoft against the considered profile shift coefficient values. The lowest value of contact ratio is 1.6, which is obtained at 0.3, whereas all other values give a higher contact ratio, thus matching the Figure 10 Characteristics.

6. Safety Factor Plot

Figure 12 shows the plot of the result of KissSoft against the considered profile shift coefficient values for safety factors of root and contact strength. The lowest values of root and contact stress safety factors are 1.84 and 1.056, respectively, which meet the global design standards need of 1.6 and thus match the Figure 10 characteristics.

7. Specific Sliding Plot

A specific sliding curve is generated through Kisssoft, and all the values are plotted in Figure 13. The specific sliding graph is generated for all values ranging from -0.2 to 0.3 and is arranged in a clockwise sequence from the first curve.

The last curve in Figure 13 clearly shows the balanced specific sliding values as compared to others. Thus, 0.3 is the best value to get specific sliding in a balanced form. Profile shift coefficient values show a strong relationship with contact ratio, safety factors and specific sliding.

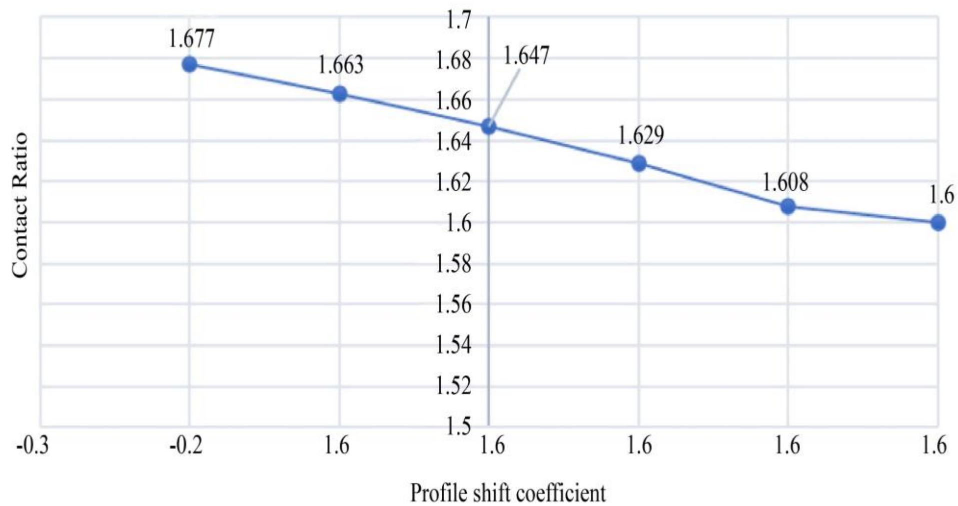


Fig. 11 Contact ratio plot

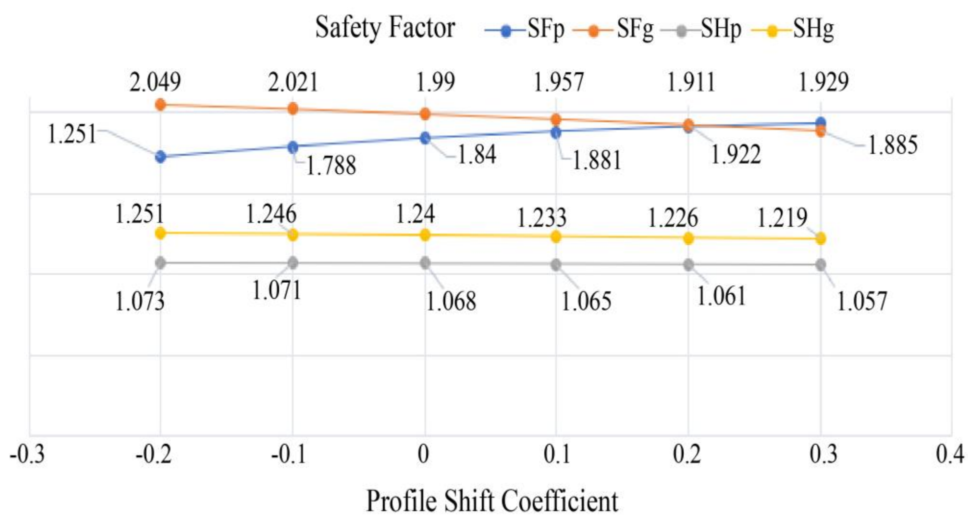
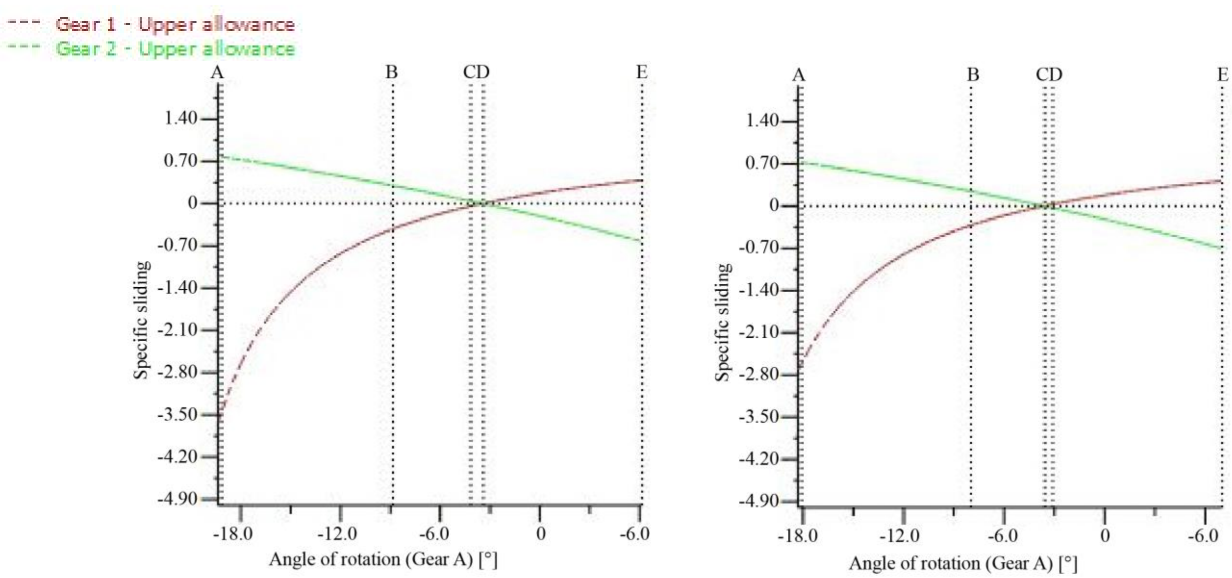


Fig. 12 Safety factor plot



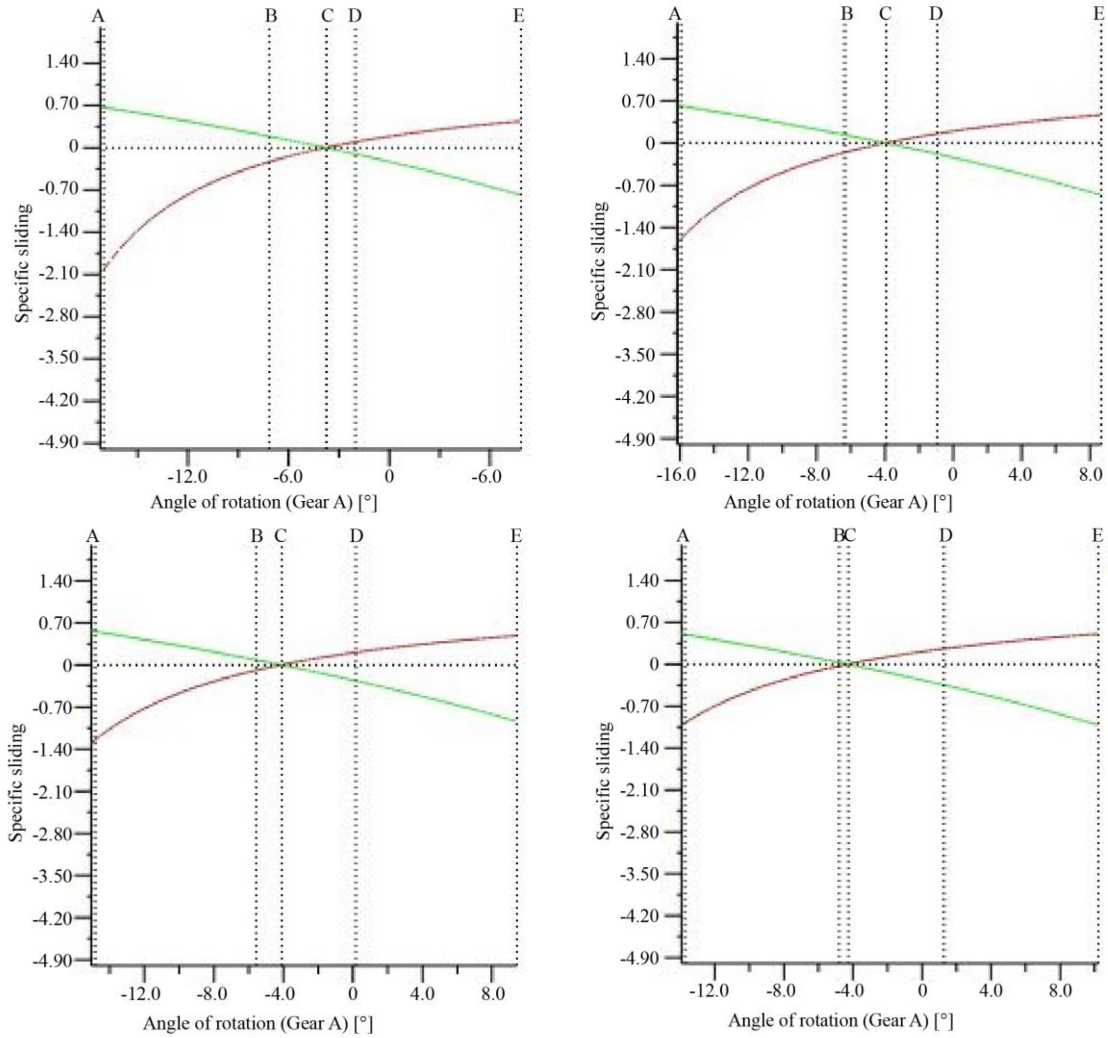


Fig. 13 Specific speed distribution

8. Conclusion and Future Scope

This research work has provided a good way to select the profile shift value for any gear pair. The methodology of selection is as below:

- Calculate the minimum value of the profile shift coefficient for no undercut.
- Calculate the specific sliding values for balanced conditions by using the Excel tool developed during this research work. Specific sliding shall be calculated for hatch zone profile shift coefficient values.
- Use Figure 10 and the required safety factors to determine the suitability.
- The same method and curve can be generated for any no of gear pairs by following the same methods.

A good Future scope could be developed as Figure 10 for a wide range of gears, with scuffing and micro pitting being also included in it.

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