ÖMER YILDIRIM   M.Sc. in Mechanical Engineering   AUGUST 2019

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GRADUATE SCHOOL OF NATURAL & APPLIED SCIENCES

DEVELOPMENT OF A USER FRIENDLY SOFTWARE FOR ASYMMETRIC SPUR GEARS INCLUDING MACRO AND MICRO GEOMETRY CONSTRUCTION

M.Sc. THESIS

IN

MECHANICAL ENGINEERING

BY

ÖMER YILDIRIM

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Supervisor

Prof. Dr. Nihat YILDIRIM

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Name of the Student : Ömer YILDIRIM

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Approval of the Graduate School of Natural and Applied Sciences

Prof. Dr. A. Necmeddin YAZICI
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

Prof. Dr. Mustafa GÜNAL
Head of Department

This is to certify that we have read this thesis and that in our consensus opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

Prof. Dr. Nihat YILDIRIM
Supervisor

Examining Committee Members:	Signature
Prof. Dr. Nihat YILDIRIM	…………………..
Prof. Dr. M.Cüneyt FETVACI	…………………..
Asst. Prof. Dr. Abdullah AKPOLAT	…………………..

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Ömer YILDIRIM

ABSTRACT

DEVELOPMENT OF A USER FRIENDLY SOFTWARE FOR ASYMMETRIC SPUR GEARS INCLUDING MACRO AND MICRO GEOMETRY CONSTRUCTION

YILDIRIM, Ömer
Ms.C. in Mechanical Engineering
Supervisor: Prof. Dr. Nihat YILDIRIM

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The aim of this thesis is to develop user friendly software for macro and micro generation of asymmetrical tooth profile spur gears.

This thesis is a part of Indigenous Helicopter Program of Rotary Wing Technology Center (DKTM) [1] which was launched at METU Technopark Facility within the scope of contract signed between Undersecretariat for Defence Industries (SSM) and Turkish Aerospace Industries, Inc. (TAI) on June 26, 2013.

A user friendly software has been developed to generate macro geometry of asymmetric gears for checking geometric limits. Analysis and design of asymmetric spur gears for tooth root bending stress with micro profile design for minimum peak to peak transmission error (PPTE) can be fulfilled by using this software. Tooth pair and mesh stiffness values can also be calculated and curves can be constructed.

Key Words: Asymmetric spur gears, Gear design, Quasi static transmission error, Tooth pair stiffness, Mesh stiffness, Tooth root bending stress, Helicopter gears.

ÖZET

ASİMETRİK DÜZ DİŞLİLERİN MAKRO VE MİKRO GEOMETRİLERİN TÜRETİLMESİ İÇİN KULLANICI DOSTU BİR BİLGİSAYAR PROGRAMI GELİŞTİRİLMESİ

YILDIRIM, Ömer
Yüksek Lisans Tezi, Makina Mühendisliği Bölümü
Danışman: Prof. Dr. Nihat YILDIRIM

Ağustos 2019
106 sayfa

Bu çalışmanın temel amacı asimetrik düz dişlilerin makro ve mikro geometrilerinin türetilmesine olanak sağlayan kullanıcı dostu bir bilgisayar programı geliştirilmesidir.

Bu tez, TAI (Türk Havacılık ve Uzay Sanayi A.Ş) ile Savunma Sanayi Müsteşarlığı (SSM) arasında 26.06.2013 tarihinde imzalanan anlaşma ile ODTÜ Teknoparkta kurulmuş olan DKTM (Döner Kanat Teknoloji Merkezi) tarafından desteklenen Özgün Helikopter Programı [1] kapsamında yürütülmüştür.

Asimetrik düz dişlilerin diş çifti direngenliklerini, statik iletim hatalarını ve diş dibi gerilmelerini hesap edebilen ayrıca dişlilerin geometrik limitlerinin kontrolüne imkan tanıyan bir program geliştirilmiştir.

Anahtar Kelimeler: Asimetrik düz dişliler, Dişli tasarmı, Statik iletim hatası, Diş çifti direngenliği, Dişli temas direngenliği, Diş dibi gerilmesi, Helikopter dişlileri.

''Dedicated to my family''

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CHAPTER 1

INTRODUCTION

1.1 General Introduction

This thesis has been conducted as a part of Indigenous Helicopter Program of Rotary Wing Technology Center (DKTM)[1] which was launched in the scope of contract signed between SSM (Undersecretariat for Defense Industries) and TAI (Turkish Aerospace Industries).

This study mainly is based on macro and micro design-analysis of asymmetrical tooth spur gears which can be applied to helicopter transmission units because of their high load carrying advantages.

1.2 An Overview of Asymmetric Spur Gears

Gears are very essential machine elements used for power transmission between parallel or non-parallel shafts. Several gear types are in use for transmitting power, torque and speed between power source and machine/systems. Due to significance of gears and their important tasks for power transmission, gear design has been given great importance by gear designers and engineers. Gear design requires two main criteria; static and dynamic performances.

Static performance is based on geometrical necessities and stress analysis of gear materials during operation whereas dynamic performance usually deals with vibration and noise. Gears experience contact stress at tooth flank and bending stress tooth root.

Safety, reliability and service life for gears are influenced by gear geometry and loading conditions. Gear engineers need to have a strong background for designing of gears and gearboxes owing to plentiful of parameters and complex structure of gearboxes. There are several design procedures such as Lewis bending equation, AGMA equation and international gear standards.

A standard spur gear has same pressure angle at both left and right sides of a gear tooth flanks because of symmetry in tooth profile and same level of performance in terms of bending and contact stresses. This symmetry of both tooth sides stems from gear generation methods but it is not an obligation.

In most practical cases, both the forward and backward rotations are not always used for power transmission. Both flanks of gear tooth have the same geometry/shape and hence similar bending and contact strengths. However, in most practical cases, both the forward and backward rotations are not always used for power transmission [2-3].

In many gear transmissions, a tooth load on one flank is significantly higher and is applied for longer periods of time than for the opposite one; an asymmetric tooth shape reflects this functional difference. With asymmetric gears, the standard symmetric tooling gear rack is modified by altering the pressure angle of one of its flanks [4].

Two sides of the gear tooth are different in terms of functions for most gears. Whereas one side (drive side) is highly loaded for longer durations, the opposite side (coast side) is unloaded or slightly loaded for short time. Thus asymmetric tooth (Figure 1.1) is well suited for cases where the torque is transmitted mainly, in one direction, [5] as in the case of most aerospace applications.

Asymmetric gears can be a good alternative to symmetric ones because of loading situation differences of both flanks of almost all gears. Asymmetrical tooth gears are mainly suitable for unidirectional torque transmission. These kinds of gears provide less space for same transmitted torque value or more torque transmission at same level of space [6].

Figure 1.1 Asymmetric involute gear tooth profile

Gear mass, noise and vibration level can be significantly reduced by designing and producing gear tooth in asymmetric form (in macro level and by applying proper profile modifications in micro level). Similarly by providing standard pressure angle at coast side and higher pressure angle at drive side, both bending and contact strength can be significantly improved with less mass [5].

While symmetric gears are being designed with standard methods, decision of suitable pressure angles at coast side and drive side for asymmetric gears are more important. Although pressure angle increase produces stronger gear in terms of bending, this increase is not limitless because of its effects on contact ratio and tooth tip thickness decrease.

Decision on pressure angle cause to change of many factors namely tooth thickness at root form factor, stress concentration factor, load factor, and moment [6]. Therefore a software is required to predict the abovementioned significant factors and design asymmetric gear having bending (and contact) stresses for specific applications.

Designing gears for a good dynamic performance, minimizing transmission error and obtaining smooth transmission error curves are important at least as much as static design regardless of symmetric or asymmetric tooth spur gears.

All requirements expected to be satisfied by gears for static and dynamic performances necessitates more efforts on asymmetric gear designs and following tasks are planned to be studies:

- ✓ Macro geometry construction of asymmetric spur gears,
- ✓ Design and analysis of asymmetric spur gears based on tooth root bending stress theoretically,
- ✓ Micro geometry design of asymmetric gears for minimized transmission error by applying profile relief to tooth tip.

1.3 Purpose of the Thesis

The purpose of this thesis is to:

- ➢ Construct gear macro geometry of asymmetric spur gears,
- ➢ Conduct a theoretical study on asymmetric spur gears to determine bending stress,
- ➢ Calculate tooth pair and mesh stiffnesses,

- Conduct a theoretical study on asymmetric spur gears to minimize peak to peak transmission error (PPTE) values and obtain smooth transmission error curves.

1.4 Structure of the Thesis

Structure of thesis is explained in this section briefly.

Chapter 2 includes literature survey on asymmetric spur gears for macro profile generation, micro profile design, and analytical efforts for root bending stress.

Chapter 3 is based on macro geometry generation of asymmetric tooth profile spur gears. Cases studies are presented for symmetric and asymmetric gears by comparing gear profiles with KissSoft [7].

Chapter 4 includes root bending stress calculations for symmetric and asymmetric tooth spur gears. Some modifications of root bending stress equations are also presented for asymmetric gears.

Tooth pair and mesh stiffness calculations are carried out. Micro geometry design for symmetrical and asymmetrical spur gears to minimize transmission error and obtain transmission error curves are presented in Chapter 5.

Chapter 6 includes discussion and conclusion of thesis.

CHAPTER 2

LITERATURE SURVEY

Gears are power transmission units used for many vehicles and machines. Gear pairs are expected to provide safe meshing for required service life under required level of torque. They need to be evaluated for bending stress at tooth root section and contact stress at tooth flank surface for accurate design. In addition, obtaining minimum transmission error value and constructing smooth transmission error curves under design and off-design loads by tip relieving of gears is an important task for gear engineers. Several studies are available in literature for different design criteria; macro geometry generation, tooth root bending stress calculation and micro profile design for minimized transmission error.

2.1 Macro Geometry Generation of Asymmetric Spur Gears

Asymmetric tooth spur gears are different than symmetric ones in terms of pressure angle of drive (highly loaded for long time) and coast side (unloaded or lightly loaded for short duration) of tooth. Romax [8], Dontyne [9] and KissSoft [7] are well known gear software to evaluate gear performances for obtaining macro geometry of gears and evaluation of tooth stresses and transmission error values. During literature survey, no commercial software has been encountered for asymmetric profile gears. It is an important task to generate macro geometry of asymmetric spur gears for stress analysis based on finite element works. In some researches, macro geometry generation of asymmetric spur gears are presented.

A mathematical model was presented to generate asymmetric spur gears in the study of Fetvaci and İmrak [10]. A computer program was developed for macro geometry generation of gear and geometry of gear cutting tools. Finite element analyses were carried out to evaluate the effect of coast side pressure angle on root stress for different case studies and stress reduction was obtained by this means.

Fetvaci [11] presented a study for macro geometry generation of asymmetric spur gears by modeling trajectory of cutting tool with different pressure angle and tip radius alternatives during gear generation for hobbing and shaping processes.

Alipiev et al [12] presented realized potential method for determining lowest value of minimum teeth number of both symmetric and asymmetric spur gears by using standard and non-standard racks. Parameters of rack cutter namely addendum, tip radius, pressure angle and fillet radius restrictions were taken into consideration for asymmetric gears with four or five teeth. Areas of existence were determined for gears with undercut and undercut-free.

Alipiev [13] intended the method of realized potential to design spur gears with profiles of symmetric and asymmetric shapes. The method aims to determine existence areas of gears with involute profile and realized potential for minimum number of teeth regardless of symmetric or asymmetric gears.

Alipiev [14] proposed a theory for gear meshing conditions based on generating asymmetric gears by using two different generalized basic racks.

Vojtkova [15] developed software to construct geometric profile of asymmetrical gears by checking thickness of top land and interference. Asymmetrical gears were presented by taking limits of pressure angle, top land thickness and contact ratio into account. Reduced radii of curvature and specific sliding curves were constructed.

2.2 Bending Stress Calculations of Asymmetric Spur Gears

There are two significant concerns in terms of gear design; bending stress at tooth root section and contact stress on tooth surface flank. Stress calculations and comparing them with regarding strengths are necessary for whether gears are safe or not. Some international gear standards such as ISO [16], DIN [17] and AGMA [18] are presented for performance evaluations of symmetrical tooth spur gears. In contrast, no international standard is encountered for calculations of geometrical parameters or stress values of asymmetric tooth profile spur gears. Some literature works for root bending stress calculations are discussed in this part of the thesis.

Kapelevich [19] established a geometrical theory for asymmetric gears. Finite element analyses were conducted in addition to experiments for asymmetric gears used in a planetary gear reduction unit of a turbo-prop engine. Author suggested that asymmetric gear with greater pressure angle at drive side of tooth yields stress reduction.

Kapelevich and Shekhtman [20] proposed a method for evaluation of bending and contact stress calculations for asymmetric tooth gears based on stress calculations for equivalent symmetric gears. 2D and 3D finite element analyses were conducted to obtain conversion coefficients to calculate tooth and root stress of asymmetric gears based on symmetric ones.

Kapelevich [21] presented a study on bending and contact stress calculations of symmetrical and asymmetrical gears for unidirectional and bidirectional loadings. Symmetric spur gear with 25 degree of pressure angle was accepted as reference for comparisons. When greater pressure angle for drive side was selected in analyses, stress reductions of 6-12% were obtained for contact stress whereas stress increase of 7-8% occurs for bending stress.

Kapelevich [22] proposed three different methods such as random search, trigonometric functions approximation and finite element to calculate root bending stress for asymmetric gears. A non-traditional method of direct gear design was developed for root stress evaluation for asymmetric external and internal gears of two stage planetary gear box of TV7 – 117S turboprop jet.

Sekar and Muhtuveerappan [23] modified equations presented in ISO standards [16] to calculate root stress modifying factors, tooth thickness at gear root, bending moment arm and root bending stress for asymmetric spur gears. In addition to calculations, finite element analyses were carried out. Load sharing and root stress were evaluated by loading asymmetric gears at tooth tip and highest point of single tooth contact. Authors recommended using greater pressure angle for drive side flank to obtain stress reduction.

Cavdar et al [24] carried out some finite element analyses for symmetrical and asymmetrical tooth spur gears to calculate tooth root bending stress. It was claimed that stress reduction of 35-39% was obtained by using asymmetric gear instead of symmetric one with 20 degree pressure angle.

Francesco and Marini [3] adapted method presented in ISO [16] for root stress calculations of asymmetric tooth spur gears. It was aimed to present a procedure for bending stress evaluation of asymmetric gears. Stress values calculated were compared finite element works result and 13% difference was obtained.

Francesco and Marini [2] underlined that asymmetry in gear tooth profile produces higher load carrying capacity or higher bending strength based on greater value of pressure angle for drive or coast side respectively. Stress reduction of 18.5% in bending was obtained in the case of drive (17.5degree) and coast (30degree) side pressure angles.

Karpat [25] performed analytical and numerical investigation concerning tooth bending stress analysis of symmetrical and asymmetrical tooth spur gear pairs. Asymmetric gears with greater pressure angle on drive side were evaluated in terms of bending stress.

Karpat et al [26] investigated asymmetric spur gears in terms of root bending stress, deflection and mesh stiffness. Profile shifted symmetric and asymmetric (with no relief) tooth spur gears were compared for root bending stress based on DIN [17] standards and finite element analyses.

Karpat et al [27] conducted a number of finite element analyses to determine bending stress for both flanks of gear tooth. Root thickness at a distance from two times module from gear tooth tip was calculated for asymmetric gear pairs. In addition, contact stress and stiffness were determined.

Marimuthu and Muhtuveerappan [28] made effort to determine optimum profile shift for asymmetric gears with normal contact ratio and high contact ratio based on direct gear design. Multi pair contact analysis was carried out to estimate load sharing and finite element analyses were conducted for gears with full rim.

Marimuthu and Muhtuveerappan [29] investigated the effects of addendum, teeth number and module on load sharing for asymmetric spur gears with greater drive side pressure angle by conducting finite element analyses for 3-teeth gear models. Stress reduction in bending and contact stresses were obtained by increase of teeth number while addendum and normal module increase yields to bending stress increase.

Sekar and Muhtuveerappan [30] estimated load sharing, bending stiffness and bending stress of asymmetrical profile helical gears designed conventionally and by direct method. Area of existence diagram was used to determine gear pair requirements and finite element analyses were carried out to evaluate the influence of gear ratio, transverse contact ratio, top land thickness and number of teeth on load sharing, stiffness and stress.

Spitas et al [31] made a parametric investigation to obtain a stronger gear tooth by evaluating combined effect of gear cutting tool tip radius and dedendum on gear root clearance. Analytical calculations and finite element works were carried out to determine root stress by taking those all abovementioned into consideration.

2.3 Stiffness Evaluations, Micro Profile Design and Transmission Error Prediction of Asymmetric Spur Gears

Transmission error (TE) has long been established as main source of vibration and unpleasant noise of gear pairs. It is highly unwelcome due to its negative effect on gear performance. TE has to be minimized in amplitude and optimized in shape to get a smooth TE curves. It is strongly related to gear dynamic performance and influence noise, vibration and harshness (NVH) characteristics adversely. There are some studies to minimize peak to peak transmission error values for gear pairs and are summarized here.

Munro et al [32] presented a study of transmission error calculation at outside of path of contact for spur gears. Theoretical and experimental works results were presented in paper. Theoretical calculations differs from actual measurements of TE values; with an amount of changing from 9.5 to 12.0 micron.

Yildirim and Munro [33] proposed a systematic approach for designing micro profile relief for low and high contact ratio spur gears. Calculations were verified by comparing results of experiments for TE values under several loads. Minimum peak to peak transmission error was obtained for design load and off design loads by applying intermediate relief to optimize TE value.

Yildirim and Munro [34] presented double relief by applying short and long relief together. It was mentioned that this new type of relief is superior to other relief types for many parameters such as PPTE, tooth load sharing and smooth TE curves. In reference study, theoretical calculations were supported by experiments.

Yildirim et al [35] designed spur gears with high contact ratio applied double relief for helicopter transmissions. In this paper, double relived spur gear pairs were tested; low PPTE and noise reduction of 7-11 dB were obtained.

Palmer and Fish [36] designed micro geometry of spur gears with and without adjacent pitch error values. Gears with different relief parameters were evaluated in terms of no relief, very short, short, intermediate, long and very long tip relief. It was underlined that a correctly designed micro profile, gear produces low transmission error.

Houser et al [37] compared theoretical and experimental results of transmission error for parallel axes gears. Experimental works for spur and helical gears were carried out by using a gear test rig whereas load distribution program (LDP) was used for theoretical calculations.

A limited number of literature works have been found on the design of micro geometry of asymmetric gears. An optimum design for micro geometry of asymmetric spur gears combined with the advantage of gear in terms of bending stress can compete with helical gears for load carrying capacity. Minimization of transmission error is very important to improve dynamic performance of asymmetric gear pairs and to keep vibration level as low as possible.

Karpat et al [38] performed 2D finite element analyses to determine stiffness by varying and holding constant (20degree) pressure angle values for drive and coast side respectively. Teeth numbers were also changed with applying different degree of asymmetry to tooth profile to see its effect on tooth stiffness.

Kapelevich and Shekhtman [39] made effort to find an optimum solution for asymmetric gear pairs with low and high contact ratios to obtain low transmission error, high load carrying capacity and high efficiency. It was stated that manufacturing tolerances with deflection of tooth under load affects TE and contact ratio. But deflections under the effects of bending and contact were taken into account.

Karpat and Ekwaro-Osire [40] presented a study for asymmetric spur gears by relieving gear tooth tip to investigate the effect of relief on wear. Reduction of wear depth and dynamic load was obtained with increased tip relief. By taking all these studies and requirements into account, it is aimed to develop a user friendly software which can fulfill following tasks in this thesis:

- Macro geometry generation of symmetric and asymmetric spur pairs,
- Tooth root bending stress calculations of symmetric and asymmetric spur gears,
- Tooth pair and mesh stiffness calculations of symmetric and asymmetric spur pairs,
- Micro profile design for minimization of quasi static transmission error for symmetric and asymmetric spur gears.

CHAPTER 3

MACRO GEOMETRY CONSTRUCTION OF ASYMMETRIC GEARS

There are some formulae and principles for macro geometry generation of symmetric involute tooth profile gears. Procedures for modifying these formulae to generate macro geometry of asymmetric spur gear are explained in this chapter. To prove correctness of these formulae modifications, some case studies have been carried out for symmetric and asymmetric tooth spur gears. Software which fulfils tasks of macro geometry generation for both symmetric and asymmetric tooth profile spur gears has been developed in the scope of this thesis. This software and some case studies are presented in this chapter. Same case studies have been also carried out by using KissSoft [7] for symmetric gears directly and asymmetric based on comparable symmetric tooth profile for drive and coast sides.

3.1 Symmetric Tooth Profile Spur Gears

Power cannot be used directly where it is produced for many applications. Therefore power transmission between power source and machines is very significant. Gears are important power transmission elements and their design requires safeness and accuracy for transmitting power and motion between parallel or non-parallel shafts through teeth.

Gear tooth profile is expected to provide a uniform motion / power transmission between shafts with a nonslip and smooth drive. Because of reasons abovementioned, gear tooth size and profile is very important. There are many different gear profiles namely cycloidal, circular and involute which perform required tasks such as uniform motion transfer with constant angular velocity. Involute profile is in universal use for gear profiles because it produces constant angular velocity and allows center distance variation without transmission ratio change.

$$\theta = \tan\emptyset - \emptyset = \mathrm{inv}\,(\emptyset) \tag{3.1}$$

Where θ and ∅ are vectorial and pressure angles respectively in radians, and inv is involute function. The involute curve can be expressed in mathematical form of vectorial angle.

R, R_B and R_C are any radius on involute curve, base radius and radius of curvature related to Radius of R respectively as presented in Figure 3.1. R_C and R depend on base circle radius of R_B. Relations are given in (Eq. 3.2-3.3).

$$\cos \emptyset = \frac{R_B}{R} \tag{3.2}$$

$$R_C = \sqrt{R^2 - R_B^2} \tag{3.3}$$

R_B is the radius of the base circle which involute curve generates from (Figure 3.1 and 3.3). Pressure angle is the angle between line of contact and the common tangent to the gears pitch circles in mesh. Pressure angle is not a constant value for each point on involute curve and differs from one radius to another. Three gear teeth models which having same module and teeth number and differ from each other in terms of pressure angle are presented in Figure 3.4. Increase in pressure angle produces stronger gear root section and thinner tooth tip (Figure 3.4).

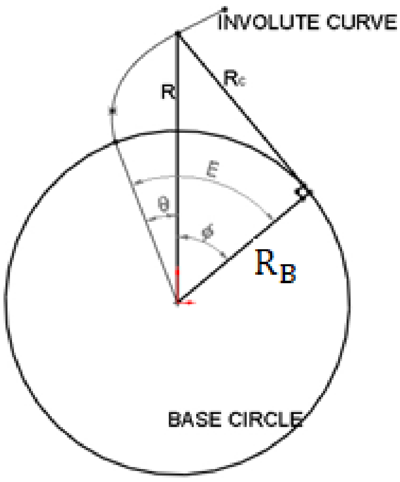


Figure 3.1 Involute tooth profile

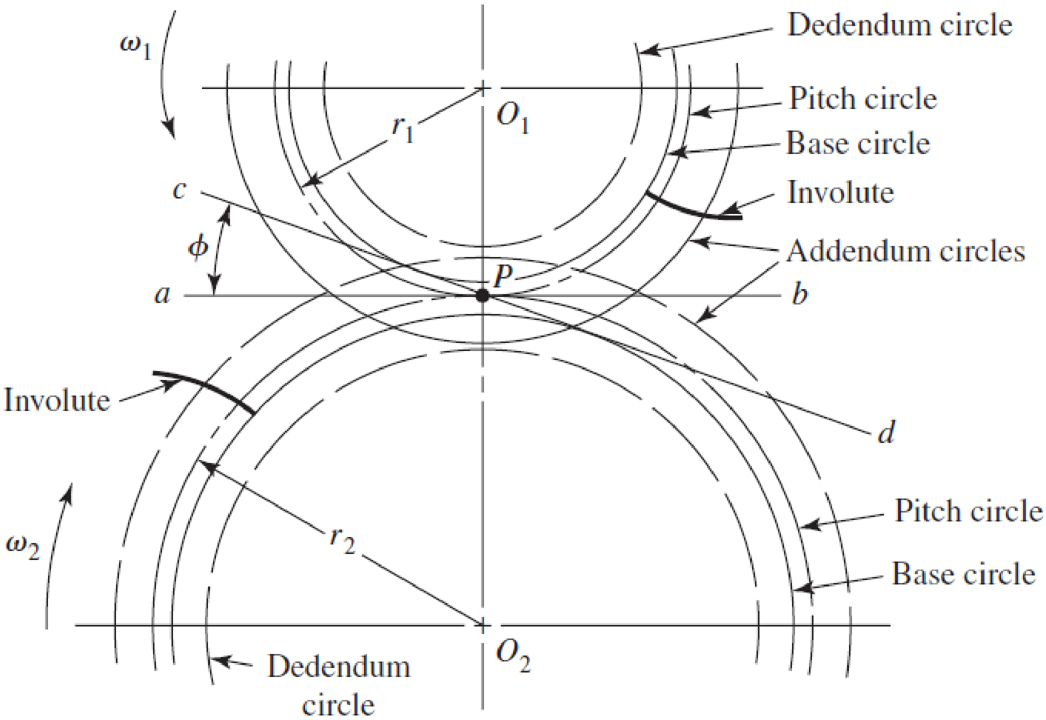


Figure 3.2 Pressure angle, base circles, dedendum circles and pitch circles [41]

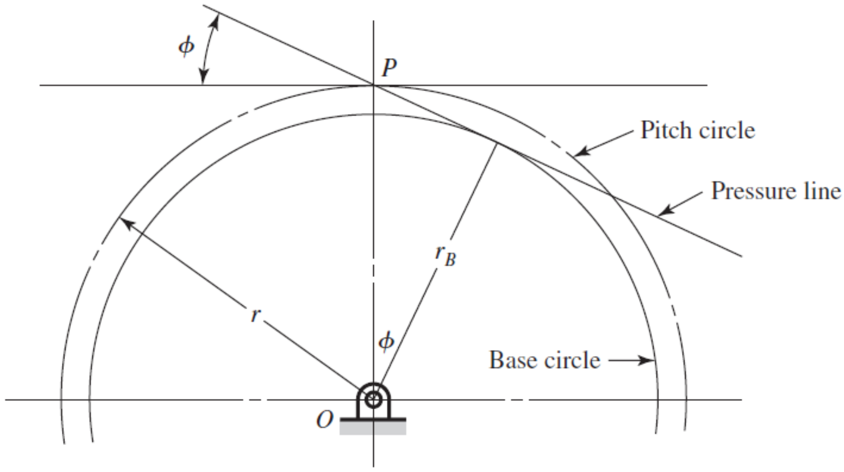


Figure 3.3 Base circle and pressure angle [41]

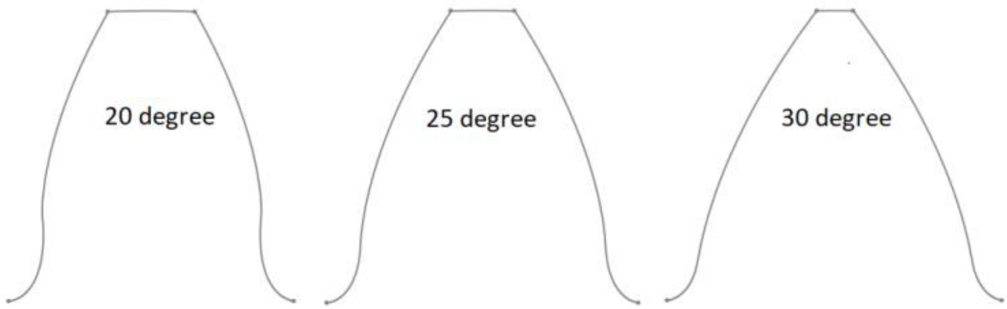


Figure 3.4 Symmetric gear teeth with pressure angle values of 20, 25 and 30 degree

3.2 Asymmetric Tooth Profile Spur Gears

Pressure angle is main gear parameter for symmetry or asymmetry of gear tooth profiles. Both flanks of gear tooth for spur gears with symmetrical profile are identical in terms of geometry, and strengths against bending and surface contact loading. In almost all applications, gears rotate in same direction without reversing. Therefore, generating gears in identical symmetric profile for both flanks is not an obligation; it is outcome of manufacturing methods. Because of all these statements given above, gears with asymmetric tooth profile can be used instead of those with symmetric tooth.

Asymmetric gears are different than symmetric ones because of pressure angle difference for drive and coast sides (Figure 3.5). This yields two different base circle radii as shown in Figure 3.6. Asymmetry in gear tooth profiles in terms of bending stress calculations are going to be evaluated in Chapter 4.

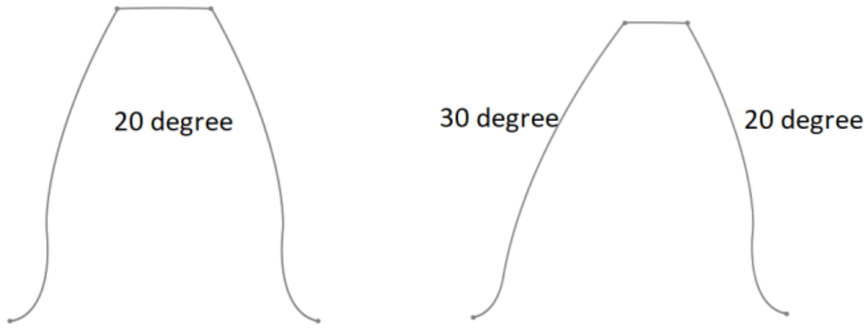


Figure 3.5 Symmetric and asymmetric tooth profile

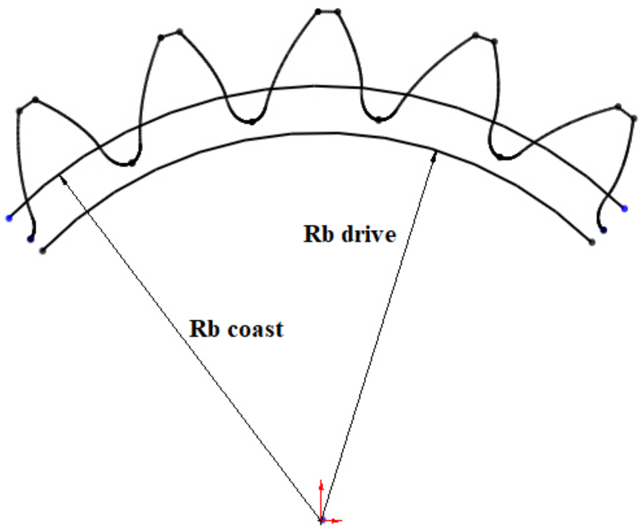
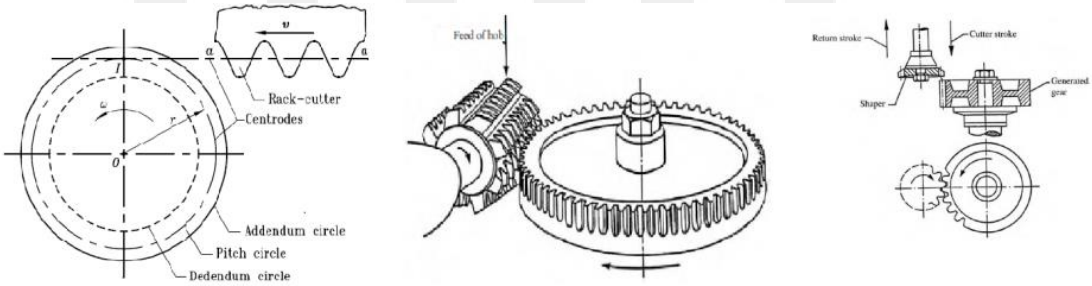


Figure 3.6 Asymmetric teeth with two different base circles

There are gear standards and well accepted formulae for evaluation of bending and contact stresses for symmetric tooth profile spur gears whereas no tool for asymmetric gears is encountered. Therefore gear designers who aim to investigate root stress of asymmetric gears can modify formulae already presented for symmetric gears or conduct finite element analyses. So macro geometry generation of asymmetric gears to analyze root stress or check geometrical limitations is very significant. Macro geometry generation software has been developed in this thesis for these aims.

3.3 Gear Geometry Generation

Gears are generated by different generation and manufacturing methods such as Maag, hobbing and Fellows shaping as shown in Figure 3.7. Hobbing is one of the gear generating processes and based on usage of gear cutter called as hob (or rack) which is straight sided cutting tool. During hobbing process, hob moves longitudinally whereas gear blank rotates in Figure 3.8. Gear tooth is generated with two different profiles of involute and trochoidal root when gear generation is completed as given in Figure 3.9.



a.Maag b.Hobbing c.Shaping

Figure 3.7 Gear generating methods [42]

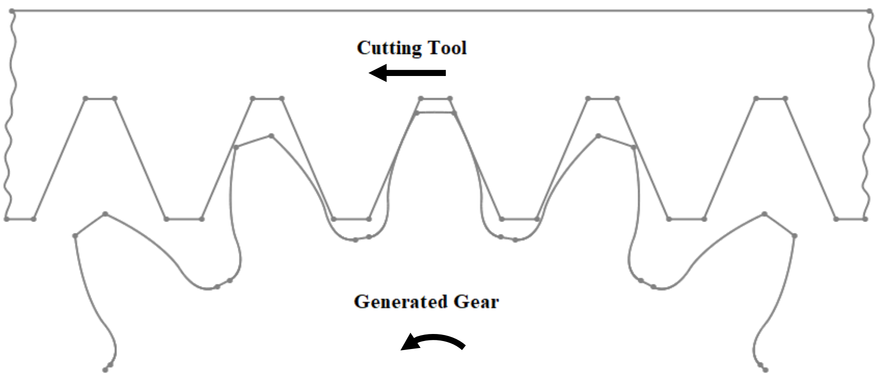


Figure 3.8 Gear generation process

A gear cutting tool, hob is a used to generate gears with identical pressure angle for drive and coast sides and module (Figure 3.10). Gear pairs generated by same hob can mesh together because of having identical same module and pressure angle.

Sum of the length of β and R equals to dedendum of gear where R is tip radius of cutting tool. TP and TH are hob tooth space and hob tooth thickness respectively. Tooth thickness of gear (on pitch circle) is equal to tooth space of hob. During generation of gears, hob moves a distance of (TP+TH) while gear rotates via an angle of (TP+TH)/R_G, where R_G is pitch radius.

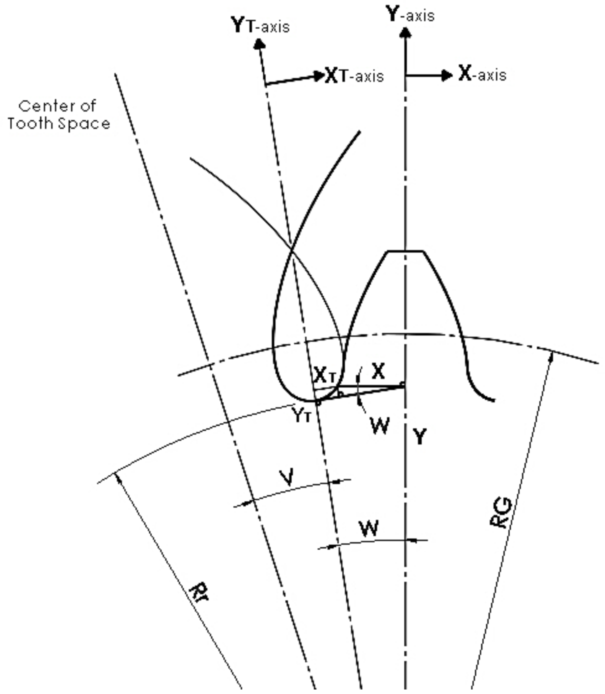


Figure 3.9 Involute and trochoid

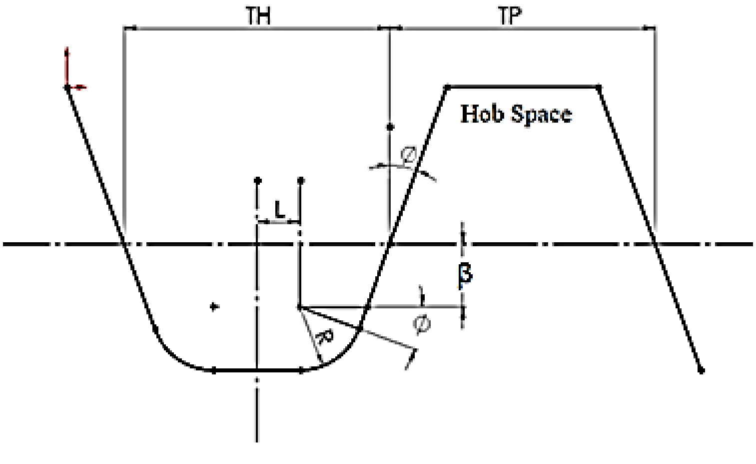


Figure 3.10 Hob geometry for symmetrical tooth profile

Gear tooth profile including involute and trochoid sections can be drawn on Cartesian coordinates. From reference book [41], equations of 3.4 – 3.27 were obtained to generate symmetric gear tooth profile and explained below.

Based on these equations and relations can be read from Figure 3.9-3.12, tooth involute profile with respect to tooth center is generated by starting at a point where tooth thickness and pressure angle are known. This point is generally selected pitch point with a pressure angle of $\emptyset_1$, pitch radius of R_1 and tooth thickness of CTT_1.

$$\theta_1 = \tan\emptyset_1 - \emptyset_1 \tag{3.4}$$

$$A = \theta_1 + \frac{0.5 * CTT_1}{R_1} \tag{3.5}$$

$$B = A - \theta_2 \tag{3.6}$$

$$\emptyset_2 = \cos^{-1}\frac{R_B}{R_2} \tag{3.7}$$

$$\theta_2 = \tan\emptyset_2 - \emptyset_2 \tag{3.8}$$

$$CTT_2 = 2 * R_2 * \left(\frac{0.5 * CTT_1}{R_1} + \theta_1 - \theta_2\right) \tag{3.9}$$

X and Y coordinates are determined for any point on involute curve:

$$X = R_2 \sin B \tag{3.10}$$

$$Y = R_2 \cos B \tag{3.11}$$

Where

$\emptyset_2$: Pressure angle at any radius,

θ_2 : Involute angle at any radius,

R_2 : Any radius on involute curve,

CTT_2 : Circular tooth thickness at any radius.

These relationships were used to prepare software for obtaining involute curve. Software routine principle is initiated based on base circle to obtain the closest point to tip with a difference of 0.125mm.

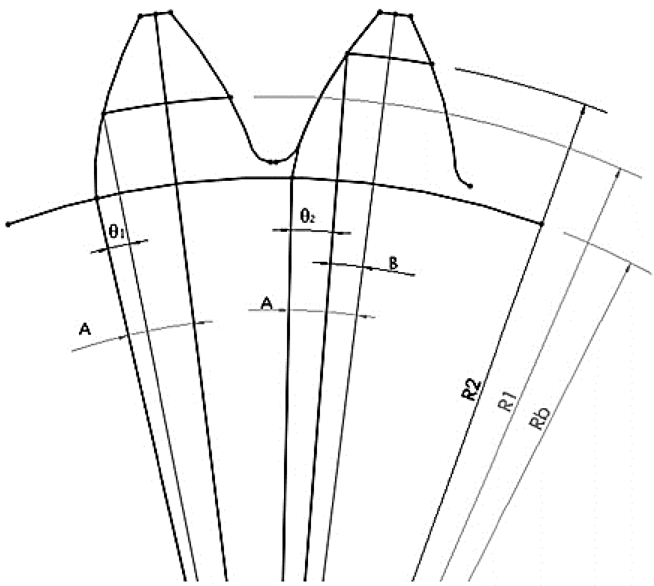


Figure 3.11 Calculating tooth thickness at any point on involute curve

Following equations are used to determine trochoid coordinates,

$$W + V = \frac{0.5*(TH+TP)}{R_G} \tag{3.12}$$

$$V = \frac{L}{R_G} \tag{3.13}$$

Where L is distance between the center of the hob tooth and point Z.

$$L = \frac{TH}{2} - \beta \tan \emptyset - \frac{R}{\cos \emptyset} \tag{3.14}$$

$$V = \frac{0.5*(TH+TP)-L}{R_G} \tag{3.15}$$

It can be seen that trochoid curve is generated by Point Z, when hob moves a distance of $R_G * E$ and gear blank rotates via an angle of E.

$$X_Z = R_Z \sin(T - E) \tag{3.16}$$

$$Y_Z = R_Z \cos(T - E) \tag{3.17}$$

$$\cos T = \frac{R_G - \beta}{R_Z} \tag{3.18}$$

$$\sin T = \frac{R_G * E}{R_Z} \quad (3.19)$$

$$X_Z = (R_G * E) * \cos E - (R_G - \beta) * \sin E \quad (3.20)$$

$$Y_Z = (R_G - \beta) * \cos E + (R_G * E) * \sin E \quad (3.21)$$

Trochoid actual value of coordinates is evaluated by adding hob tip radius to generated coordinates (Figure 3.9):

$$X_T = X_Z + R * \cos A \quad (3.22)$$

$$Y_T = Y_Z - R * \sin E \quad (3.23)$$

Trochoid coordinates with respect to tooth center is determined at last:

$$\sin W = \frac{X_T + X\cos W}{Y} \quad (3.24)$$

$$\cos W = \frac{Y_T - X\sin W}{Y} \quad (3.25)$$

$$X = Y_T \sin W - X_T \cos W \quad (3.26)$$

$$Y = Y_T \cos W + X_T \sin W \quad (3.27)$$

Thus far, macro geometry generation including involute and trochoid profiles has been explained based on Figures of 3.8 - 3.12 and Equations of 3.4 - 3.27. Hob geometry and equations for generation of asymmetric tooth profile involute gear are modified as shown in Figure 3.13.

In this thesis, an asymmetric tooth spur gear is modeled as two comparable symmetric tooth spur gears to generate gear tooth and whole gear models (Figure 3.14). These two comparable symmetric gears refer to drive and coast side of asymmetric gear in terms of pressure angle and base circle radius.

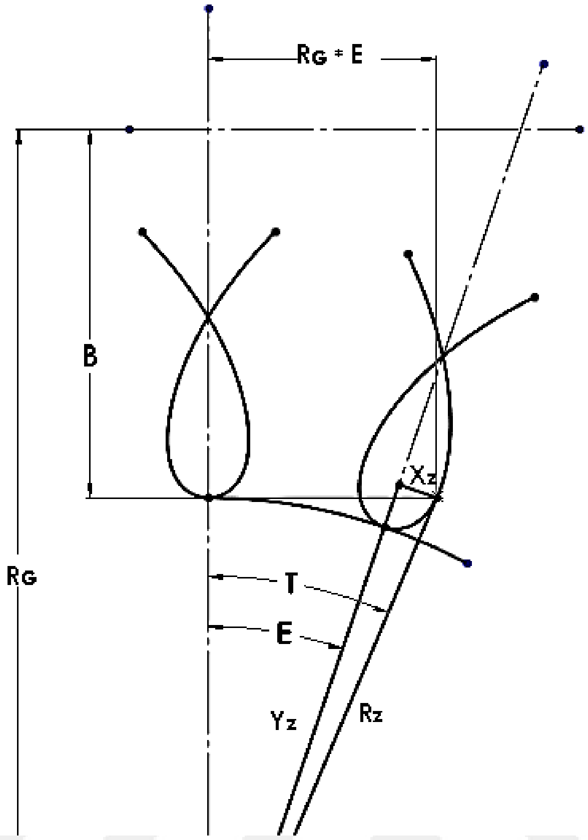


Figure 3.12 Trochoid generated by Point Z

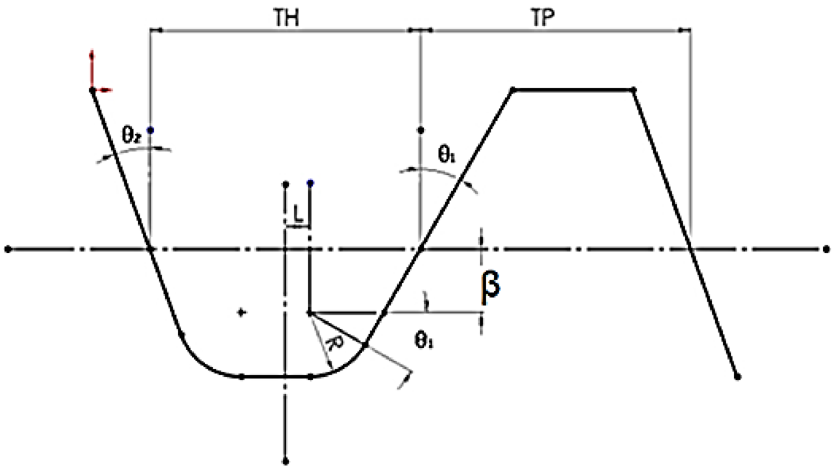


Figure 3.13 Hob geometry for asymmetrical tooth

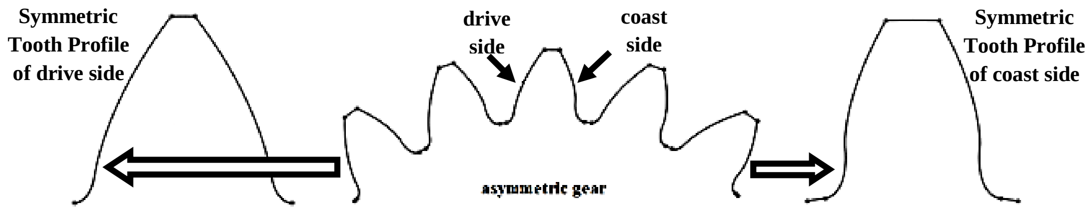


Figure 3.14 Modelling asymmetric gear as two symmetric gears

All equations given for macro geometry generation of symmetric spur gears have been modified to obtain asymmetric tooth involute profile. These equations are presented below (Eq. 3.28-3.37).

$$\theta_{1d,c} = \tan\phi_{1d,c} - \phi_{1d,c} \tag{3.28}$$

$$A_{d,c} = \theta_{1d,c} + \frac{0.5*CTT_1}{R_1} \tag{3.29}$$

$$B_{d,c} = A_{d,c} - \theta_{2d,c} \tag{3.30}$$

$$\phi_{2d,c} = \cos^{-1}\frac{R_{Bd,c}}{R_2} \tag{3.31}$$

$$\theta_{2d,c} = \tan\phi_{2d,c} - \phi_{2d,c} \tag{3.32}$$

$$CTT_{2d} = 2 * R_2 * (\frac{0.5*CTT_1}{R_1} + \theta_{1d} - \theta_{2d}) \tag{3.33}$$

$$CTT_{2c} = 2 * R_2 * (\frac{0.5*CTT_1}{R_1} + \theta_{1c} - \theta_{2c}) \tag{3.34}$$

$$CTT_2 = 0.5 * (CTT_{2d} + CTT_{2c)} \tag{3.35}$$

$$X_{d,c} = R_2 \sin B_{d,c} \tag{3.36}$$

$$Y_{d,c} = R_2 \cos B_{d,c} \tag{3.37}$$

Where

$\phi_{2d,c}$: Pressure angle at any radius for drive and coast side flank,

$\theta_{2d,c}$: Involute angle at any radius for drive and coast side flank,

R_2 : Any radius on involute curve (it is same for both flanks on involute curve),

CTT_2 : Circular tooth thickness at any radius (it is sum of half thickness of drive and coast side tooth flanks)

3.4 Software

Based on gear geometry generation principles explained above, a software has been developed for generation of symmetric and asymmetric tooth spur gears. Software has module of geometrical construction including Main Window, Input Parameters Window, Output and Tooth Profile Drawing Window, Results Window and Error

Warnings Window as shown in Figure 3.15-3.18. Symbols for gear parameters used in software and their descriptions are given in Table 3.1. When main window is obtained, sections of inputs, outputs, calculations and drawings can be seen as shown in Figure 3.15.

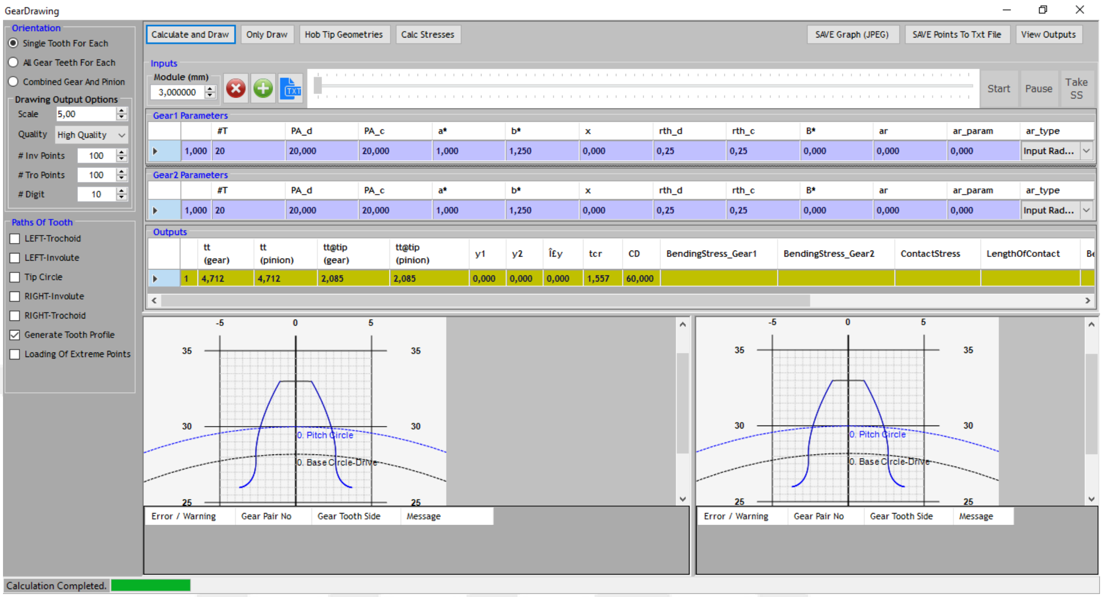


Figure 3.15 Main windows of software

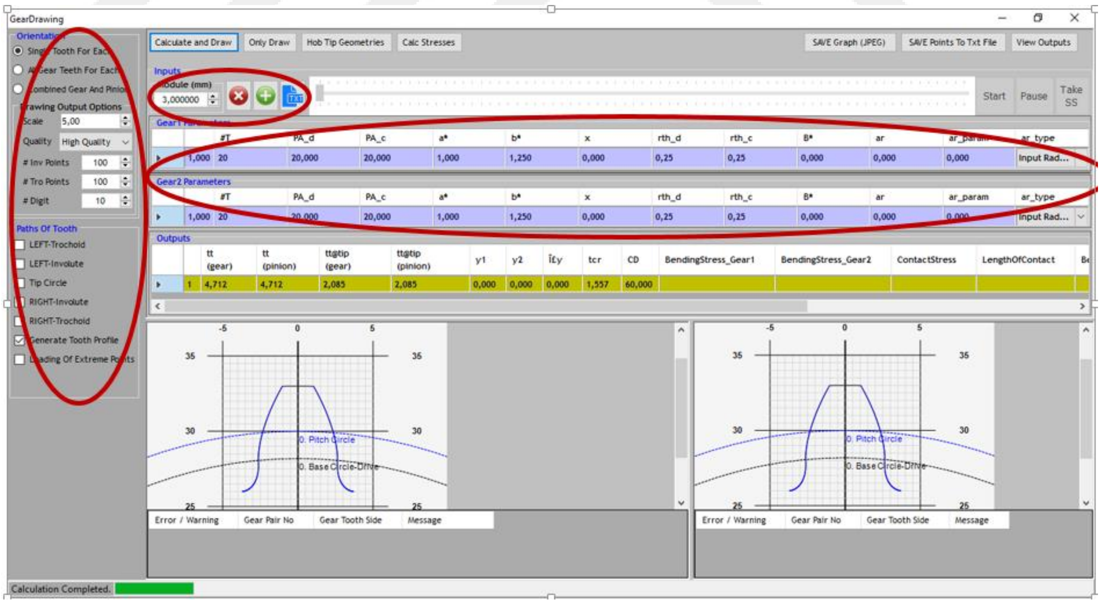


Figure 3.16 Input parameters window of software

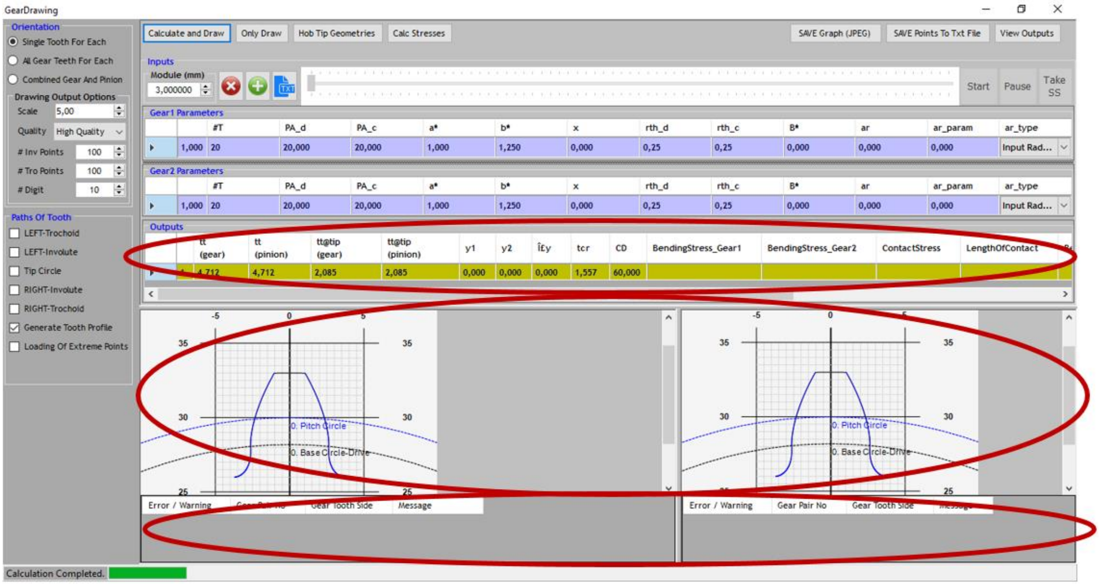


Figure 3.17 Output parameters and drawing section window of software

As shown in Figure 3.16, gear parameters such as teeth numbers, pressure angles of drive and coast side of tooth profiles, coefficients of addendum, dedendum, hob tip radius and profile shift and backlash can be selected for pinion and gear separately whereas module is common parameter for both gears. By using software, different root fillet radii coefficients can be applied for drive and coast side of trochoidal region. This enables asymmetric fillet whose advantages were deeply discussed in reference studies [43-46] for application to spur gears. Scale for gear macro geometry drawing on screen and number of points to generate involute and trochoid profiles can be changed by user. Sections of tooth profiles and root profiles for drive and coast sides of gear tooth can be selected for generation or can be omitted by designer.

Gear parameters of tooth thickness on pitch circle, tooth tip thickness, profile shift amount are calculated for pinion and gear and presented in Figure 3.17. Transverse contact ratio and nominal center distance are given for gear pair. Nominal center distance is calculated based on given gear pair parameters and can be changed by designer. It also allows checking its effect on contact ratio of gear pair. Pinion and gear macro geometries are generated and presented in drawing section of output module of software. If there is any error or warning should be taken into consideration such as low contact ratio and excessive decrease in tooth tip thickness, these are given in error-warning section of this module. For design or analysis, a new gear pair with different parameters of gears except module (module is common for all gear pairs) can be added and then can be removed from analysis list. Designed gear tooth or whole gear profiles can be imported as points in a text files (Figure 3.19).

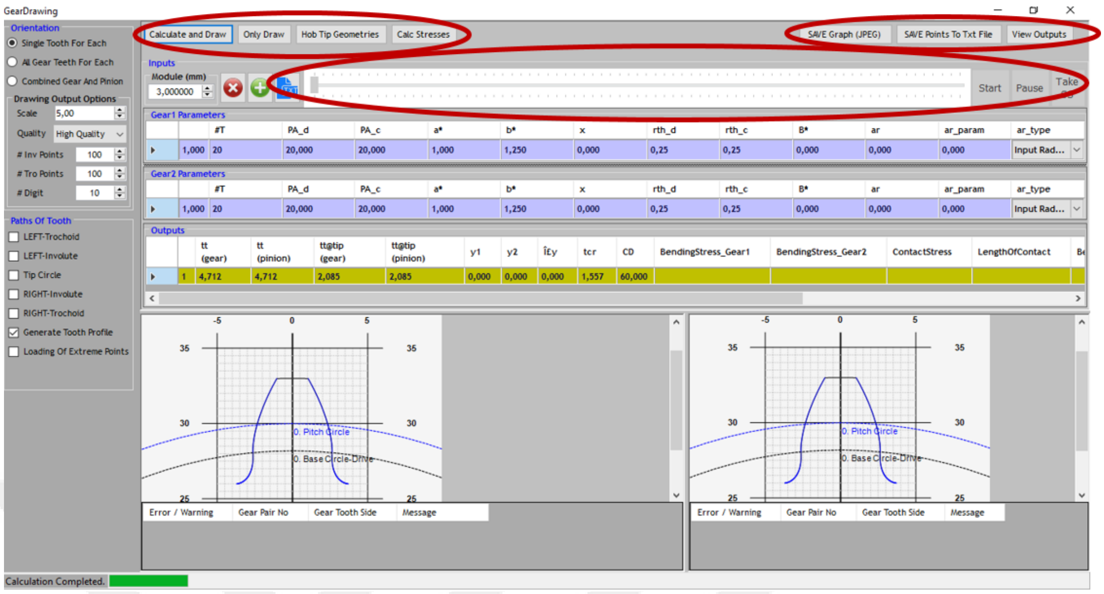


Figure 3.18 Other capability of software for stress calculation, geometry generation-drawing, saving gear profiles, viewing outputs,

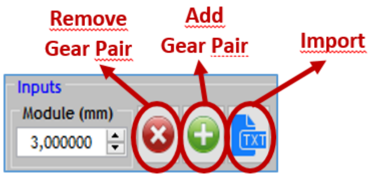


Figure 3.19 Adding new gear pair, removing and importing

Table 3.1 Gear parameters descriptions for software

<table>
<tr><th>Symbol</th><th colspan="2">Description</th><th>Unit</th></tr>
<tr><td>#T</td><td colspan="2">Teeth Number</td><td>-</td></tr>
<tr><td>PA_d</td><td colspan="2">Pressure Angle (Drive Side)</td><td>deg</td></tr>
<tr><td>PA_c</td><td colspan="2">Pressure Angle (Coast Side)</td><td>deg</td></tr>
<tr><td>a*</td><td colspan="2">Addendum Coef.</td><td>-</td></tr>
<tr><td>b*</td><td colspan="2">Dedendum Coef.</td><td>-</td></tr>
<tr><td>x</td><td colspan="2">Profile Shift Coef.</td><td>-</td></tr>
<tr><td>rth_d</td><td colspan="2">Hob Tip Radius (Drive Side)</td><td>-</td></tr>
<tr><td>rth_c</td><td colspan="2">Hob Tip Radius (Coast Side)</td><td>-</td></tr>
<tr><td>B*</td><td colspan="2">Backlash Coef.</td><td>-</td></tr>
<tr><td>ar</td><td colspan="2">Amount Of Relief</td><td>µm</td></tr>
<tr><td rowspan="3">ar_param</td><td rowspan="3">Relief Value</td><td>Input Roll Angle</td><td>deg</td></tr>
<tr><td>Input Radius</td><td>mm</td></tr>
<tr><td>Input Extend Of Relief</td><td>mm</td></tr>
</table>

Gear geometry generation and simulation of gear pair is selective for software. Single tooth, whole gear and simulation of gears (rotating in mesh) can be select by user. These gear drawings and meshing simulation can be scaled (Figure 3.20). In addition, number of points for generation of tooth and root profiles may be determined according to needs. Different selections to draw different sections of involute and trochoid for drive and coast sides of gear tooth as shown in Figure 3.20. There are some buttons of "Calculate and Draw" and "Only Draw" as presented in Figure 3.21. These buttons provides calculations and then drawing gear profiles or only drawing of gears after changing any of gear pair parameters. After clicking button of "Calculate and Draw", gears are generated as shown in Figure 3.22. To check only geometrical change of gear pairs, button of "Only Draw" is used. Additionally, gear pairs more than one can be drawn on another pair as shown in Figure 3.23.

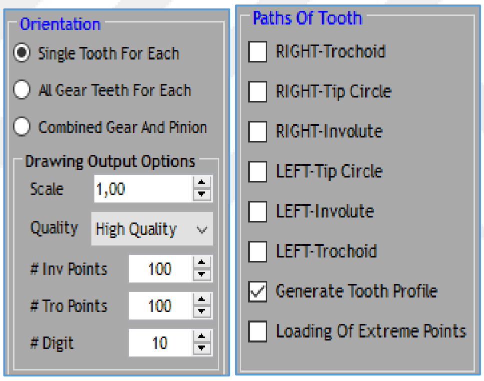


Figure 3.20 Orientation of gear geometry generation and selection of profile section generation

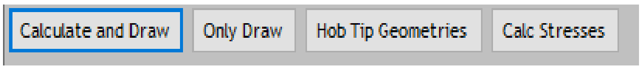


Figure 3.21 Calculation and running buttons

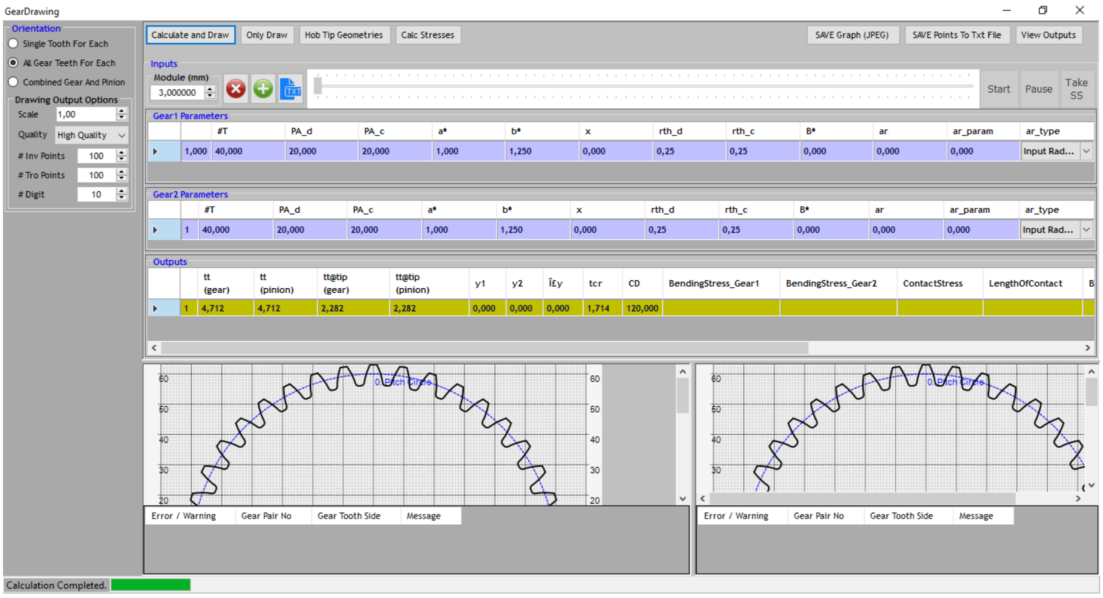


Figure 3.22 Generation of gears

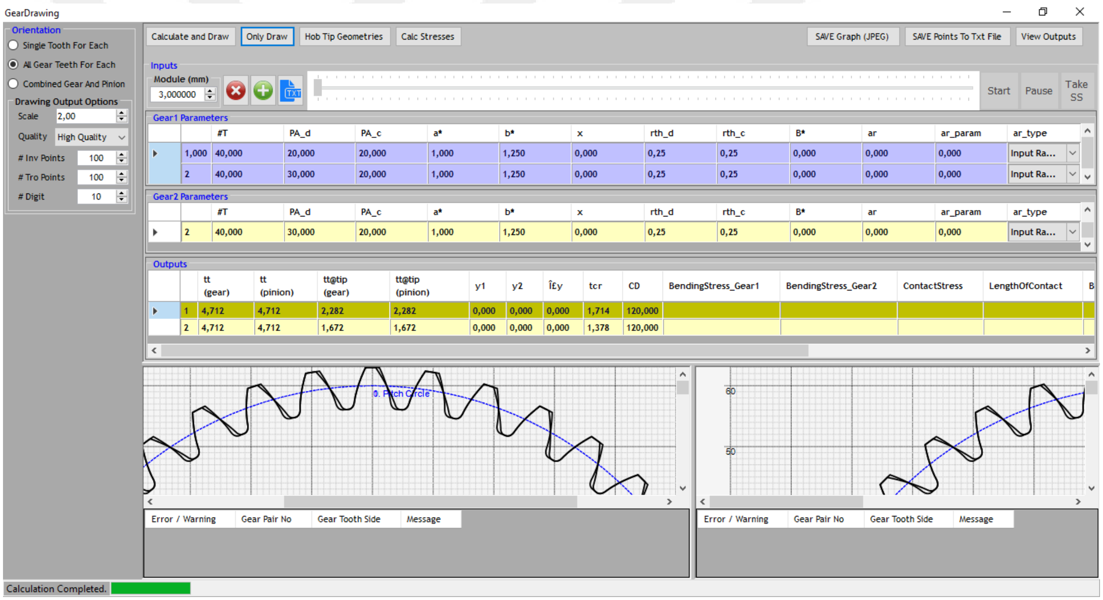


Figure 3.23 Drawing of gear pairs one over another by using button of "Only Draw"

Software allows checking hob geometry and its tip radius as given in Figure 3.24. By this means, any disorder of hob geometry generation before gear macro generation can be determined.

When radio button of "Combined Gear And Pinion" is selected (Figure 3.20), simulation can be started as presented in Figure 3.25. Then simulation of gear pair can be observes as in Figure 3.26. Buttons of Start, Pause/Play and Take SS (Figure 3.25) allows starting simulation, pausing-playing again and taking screen shot respectively.

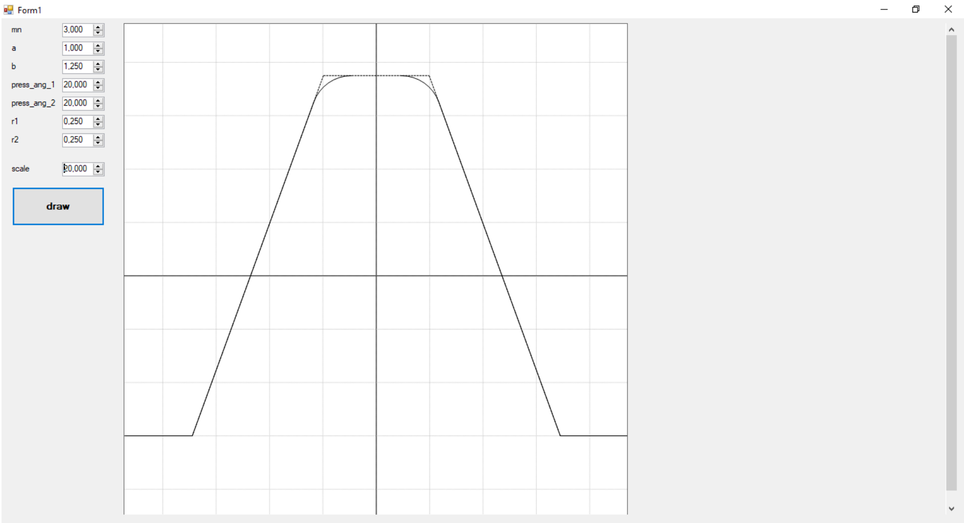


Figure 3.24 Generation of hob geometry

Figure 3.25 Simulation buttons

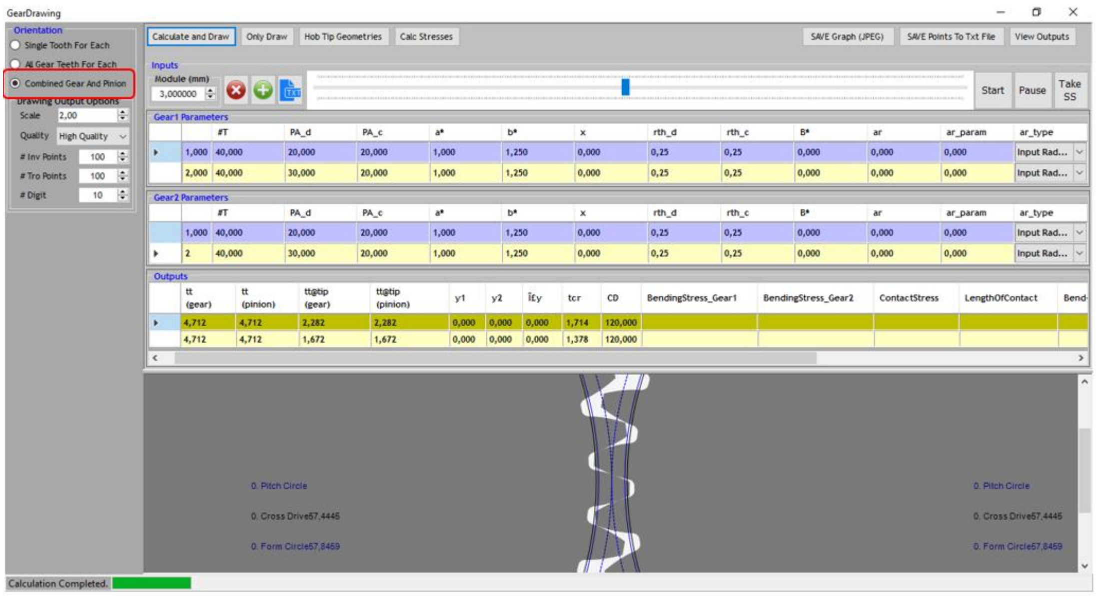


Figure 3.26 Simulation Window

Software enables to save gear macro profile, save single tooth or whole gear coordinates and view all calculated gear parameters by clicking on related button (Figure 3.27). Gears can be saves as JPEG as shown in Figure 3.28. Coordinates of points needed to gears by using Solidworks [47] can be saved in txt format (Figure 3.29-3.32).

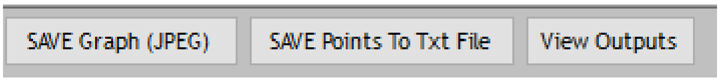


Figure 3.27 Saving outputs

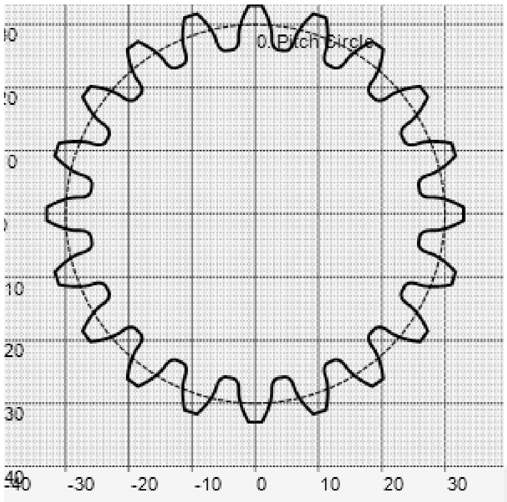


Figure 3.28 Gear figure saving as JPEG

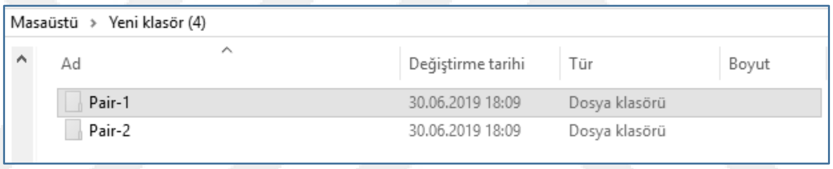


Figure 3.29 Saving gear pair profiles coordinates in file

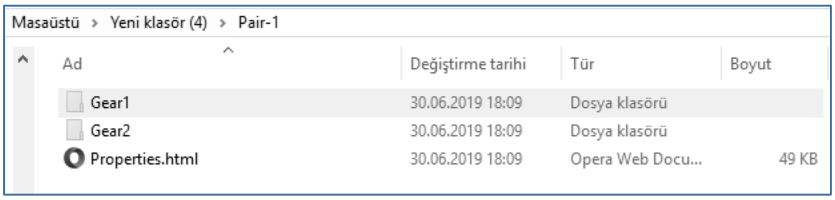


Figure 3.30 Saving pinion and gear profiles coordinates in file

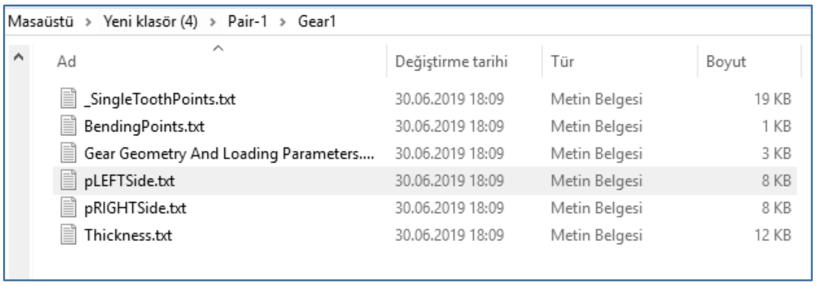


Figure 3.31 Saving gear profiles coordinates for left-right flanks

By clicking button of "View Outputs" as shown in Figure 3.27, calculated gear parameters in HTML format are presented in Figure 3.33

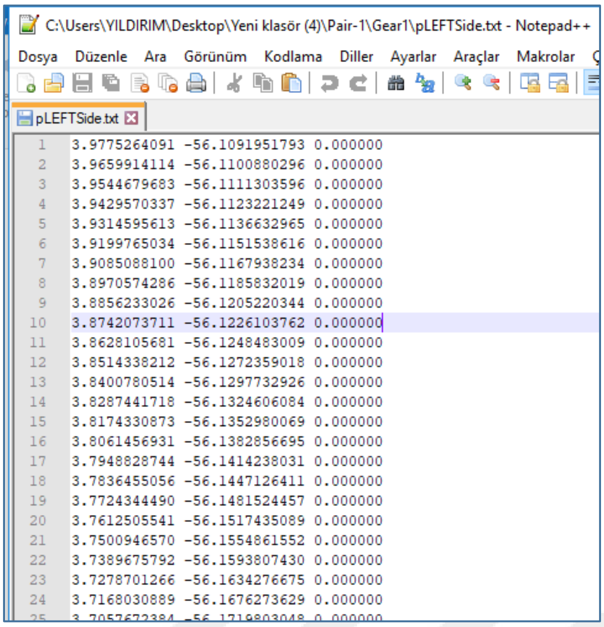


Figure 3.32 Saving gear profile coordinates for left side flank

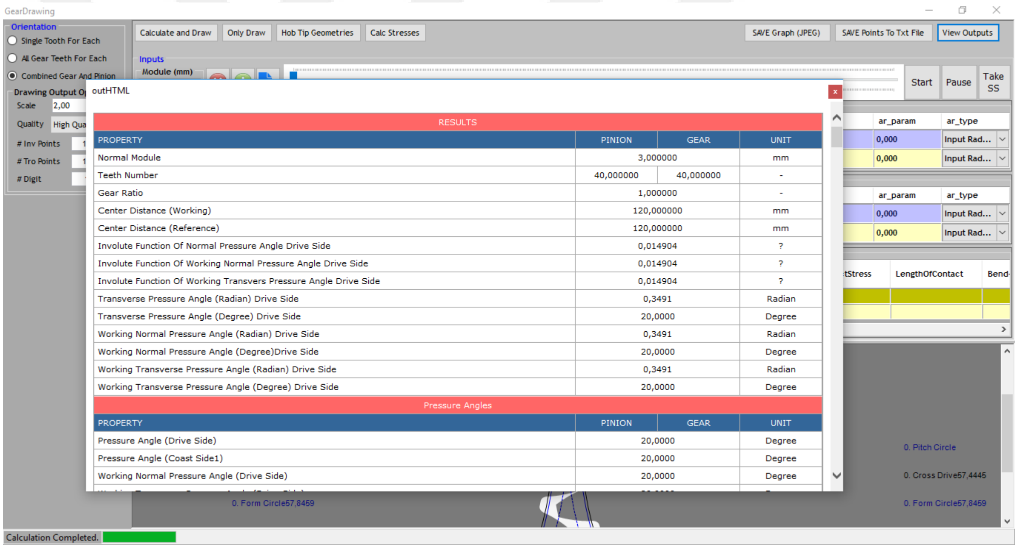


Figure 3.33 Viewing outputs in HTML format

Macro geometries of single tooth, loading positions of gear tooth, whole gear body and gear pair in mesh can be seen in Figure 3.34-3.37 respectively. Generations of single tooth profile or whole gear body for more than one pair of gear are possible by this software.

With center distance change, contact ratio change and interference can occur between gear teeth as presented in Figure 3.38. Gear parameters such as pressure angle and teeth number have significant effect on undercutting. Occurrence of undercutting for gear pairs is also determined by software and warning is presented in Figure 3.39.

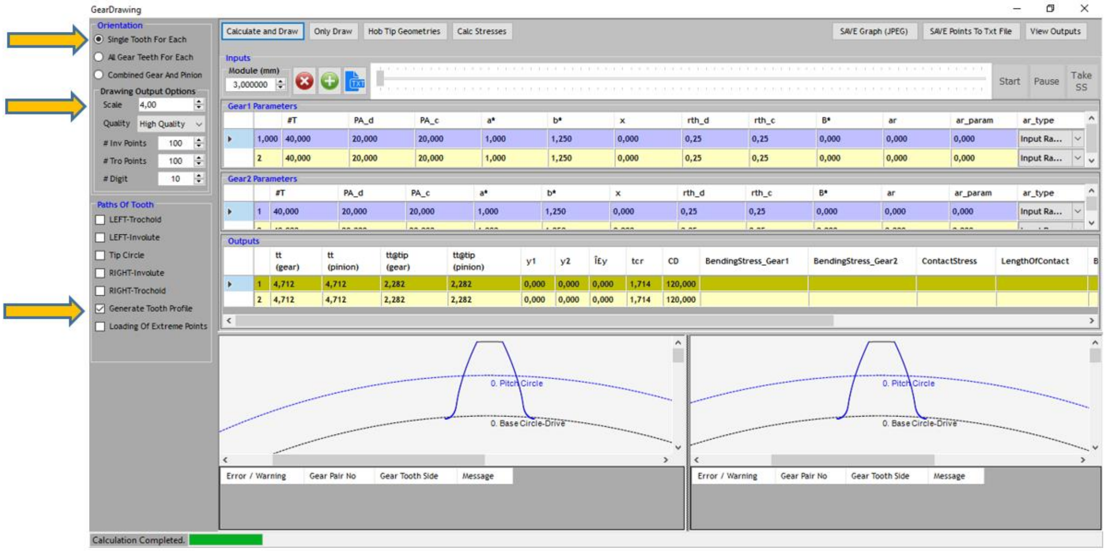


Figure 3.34 Single tooth macro geometry for pinion and gear

Figure 3.35 Loading gear teeth at highest point of single tooth contact (HPSTC)

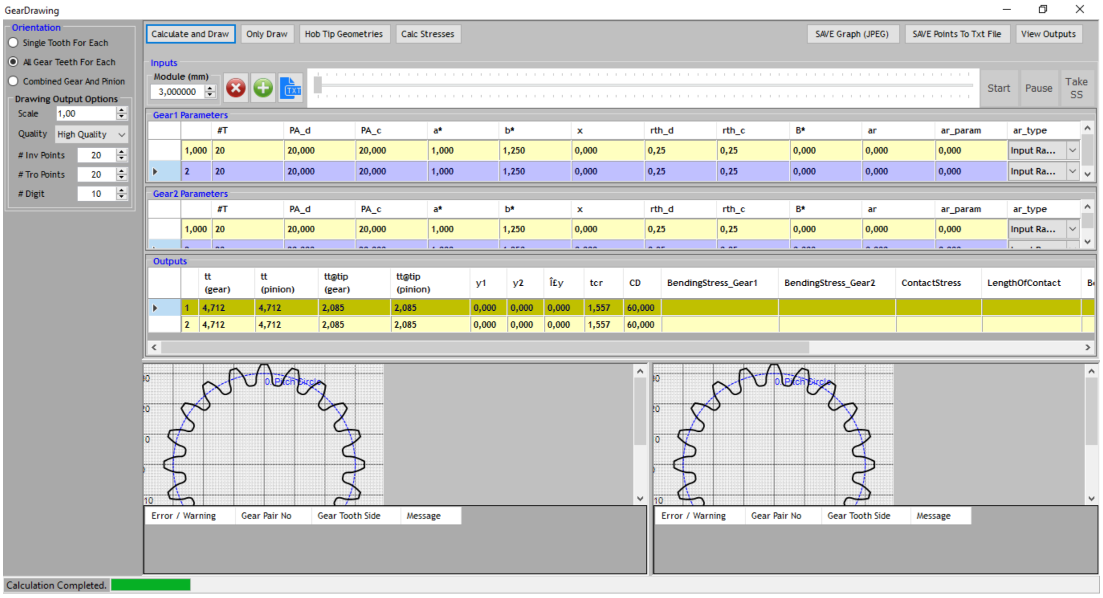


Figure 3.36 Whole gear macro geometries

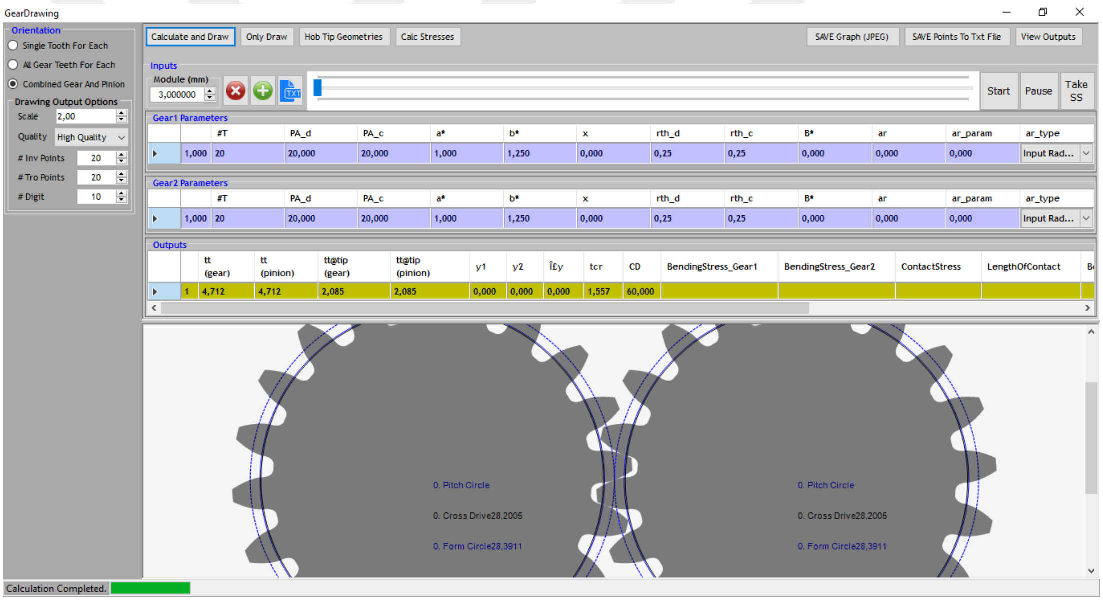


Figure 3.37 Combined pinion and gear (meshing)

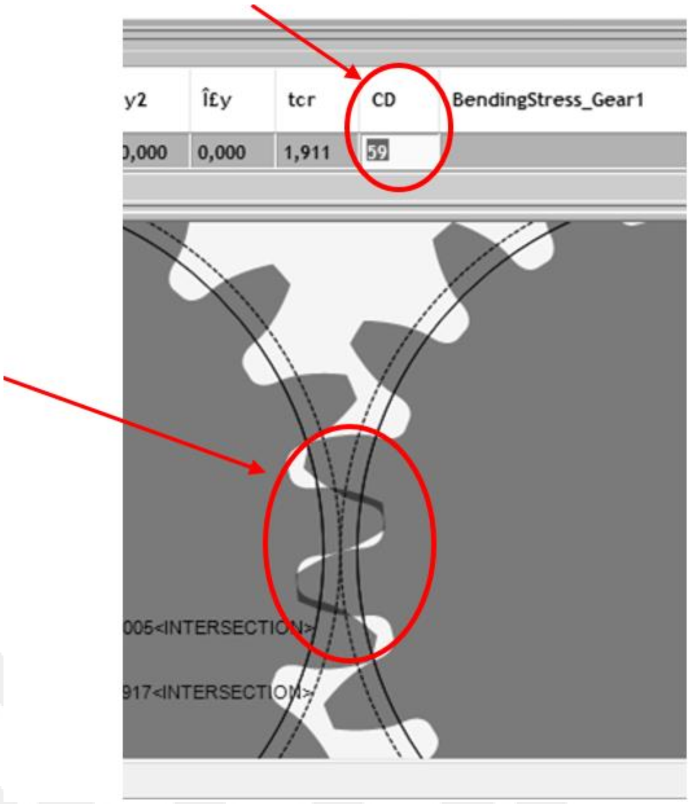


Figure 3.38 Interference because of center distance change

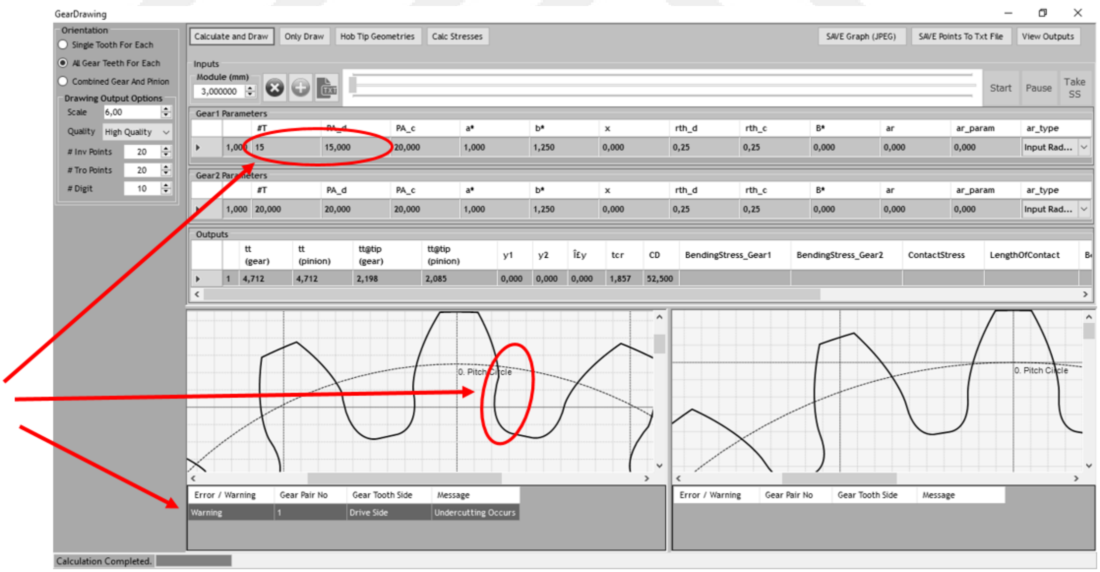


Figure 3.39 Undercutting occurrence and warning

3.5 Case Studies Of Symmetric Tooth Spur Gears

The main aim of this chapter is to generate asymmetric tooth profile spur gears for graphical check of its correctness and accuracy. Generated macro geometry of symmetric or asymmetric spur gears can also be used for any finite element analyses to determine stress or deflection values. Before generating of asymmetric spur gear profiles, macro geometry of symmetric tooth profile spur gears are generated and compared to obtained geometry by KissSoft [7] in this section.

DXXCYY is an abbreviation used for case studies of symmetric and asymmetric tooth spur gears. D refers to drive side of tooth profile whereas C represents coast side flank. XX and YY are used to define pressure angle values of both tooth flanks. For instance, D20C20 symbolize a symmetric tooth spur gear with pressure angle of 20degree for both side of tooth profile. In contrast, D30C20 refers to an asymmetric tooth spur gear with the pressure angle values of 30degree and 20degree for drive and coast sides respectively.

3.5.1 Case S1

Gear parameters for macro geometry generation of symmetric spur gear pair are given in Table 3.2. Symmetric spur gear tooth profile was generated by using developed in-house software and KissSoft [7]. Gear tooth profiles are compared for both tools in Figure 3.40. Blue curve refers to geometry generated by in-house software whereas red curve belongs to KissSoft. Solid black and red lines obtained by in-house software are given for drive side and coast side profiles respectively. Blue and yellow points refer to drive and coast side flanks constructed by KissSoft as can be seen in Figure 3.40. These curves and corresponding points overlap without any difference or deviation from each other.

Table 3.2 Symmetric tooth spur gear pair parameters (Case S1)

Parameter	Value	Unit
Module	3	mm
Teeth number	40	-
Pressure angle	20	degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.25	-

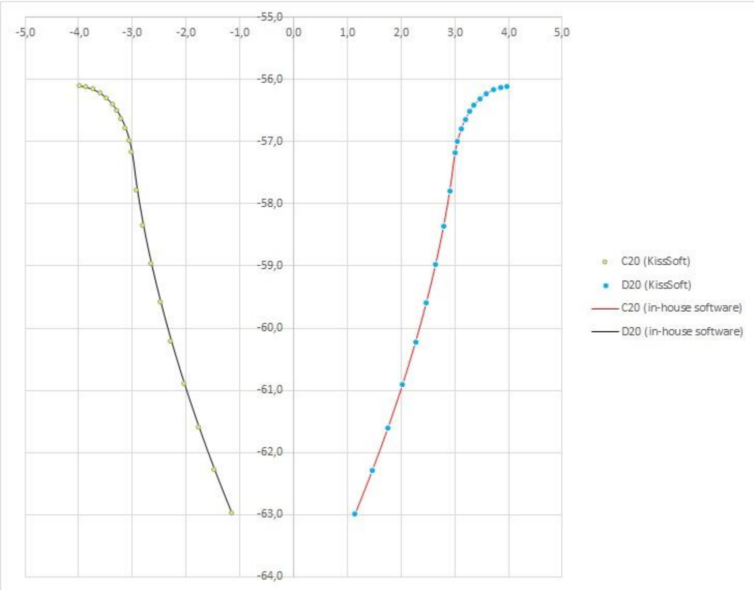


Figure 3.40 Tooth profile for Case S1
(obtained from KissSoft and in-house software)

3.5.2 Case S2

Macro geometry of a symmetric tooth profile spur gear pair with 20degree of pressure angle similar to Case S1 but different module and teeth number values was constructed by using in-house software and KissSoft. Gear parameters are given in Table 3.3. Drive and coast side profiles are presented for different line types and colors. Whereas black and red lines refer to profiles constructed by using in-house software, blue and yellow circles are belong to curves obtained from KissSoft for drive and coast flanks.

Table 3.3 Symmetric tooth spur gear pair parameters (Case S2)

Parameter	Value	Unit
Module	1	mm
Teeth number	20	-
Pressure angle	20	degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.47	-

3.5.3 Case S3

Gear pair with same gear parameters except pressure angle was constructed. Pressure angle of 30degree for drive and coast sides of gear tooth was preferred to 20degree. This gear pair macro geometry construction was carried out by both tools; in-house software and KissSoft. Curves are presented in Figure 3.42. Types and colors of curves and points for Case S3 are identical to those presented for Case S2. Curves and points coincide; there is no deviation for profiles generated by both tools (Figure 3.42).

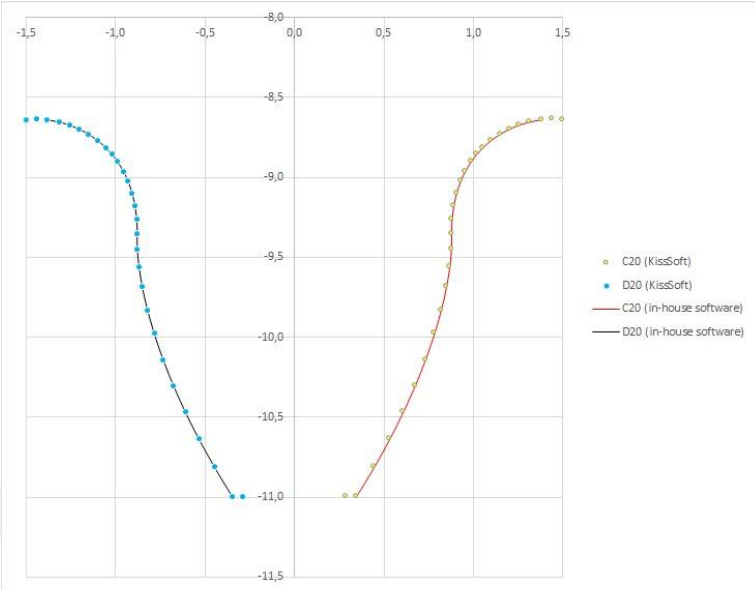


Figure 3.41 Tooth profile for Case S2
(obtained from KissSoft and in-house software)

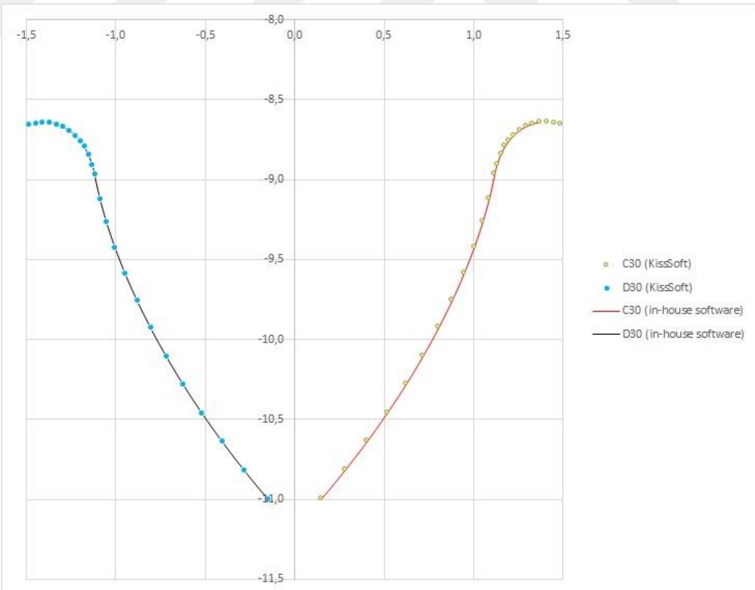


Figure 3.42 Tooth profile for Case S3
(obtained from KissSoft and in-house software)

3.6 Case Studies Of Asymmetric Tooth Spur Gears

Cases studies for macro geometry generation of asymmetric tooth spur gears are conducted in the scope of this thesis. During investigation, no commercial tool is encountered for generation of macro geometry for asymmetric spur gears. Therefore, constructed asymmetric tooth profile is compared to the symmetric tooth profiles obtained by KissSoft for drive side (modeled as one symmetric profile) and coast side (modeled as another symmetric profile). In case of need to explain this comparison method in detail, two symmetric tooth spur gears with pressure angles of 20degree (D20C20) and 30degree (D30C30) are generated by using KissSoft to compare an asymmetric tooth spur gear (D30C20) constructed by in-house software.

3.6.1 Case A1

Asymmetric tooth profile with pressure angle of 30degree and 20degree for drive and coast sides respectively is generated by using in-house software. Gear parameters used for macro geometry generation are given in Table 3.4.

This profile is required to compare but no commercial software which constructs macro geometry of asymmetric spur gears has been encountered during survey. So an asymmetric tooth is considered as two representative symmetric tooth profiles. For instance, an asymmetric tooth with pressure angles of 30degree and 20degree for drive and coast sides respectively are generated. These two representative symmetric teeth profiles are constructed by KissSoft for comparison. In-house software generates asymmetric gear macro geometry at one time without any need for comparable symmetric profiles.

Comparison of drive and coast side tooth profile of asymmetric gear is carried out by superimposing profiles obtained by in-house software and KissSoft as shown in Figure 3.43.

Table 3.4 Asymmetric tooth spur gear pair parameters (Case A1)

Parameter	Asymmetric gear	Representative symmetric gears		Unit
		Gear 1	Gear 2	
Module	3	3		mm
Teeth number	40	40		-
Pressure angle (drive/coast)	30/20	30/30	20/20	degree
Addendum coefficient	1.00	1.00		-
Dedendum coefficient	1.25	1.25		-
Cutter tip radius coefficient	0.1	0.1		-

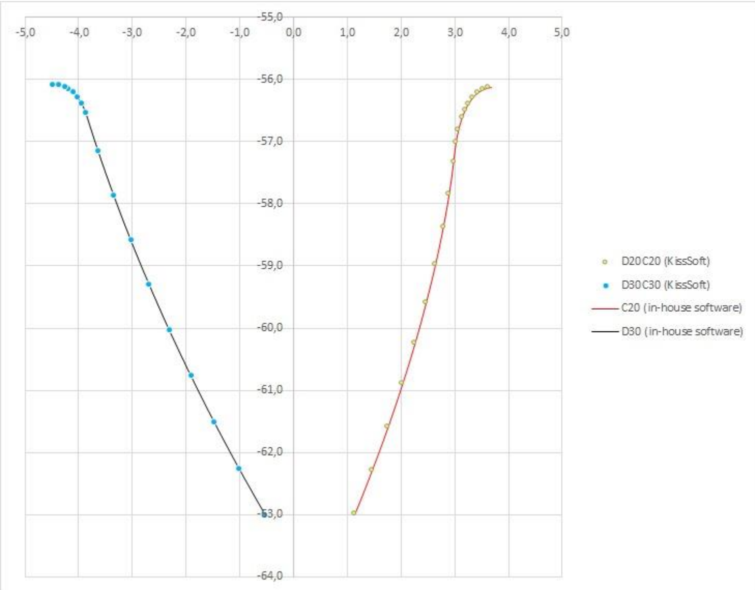


Figure 3.43 Tooth profile for Case A1
(obtained from KissSoft and in-house software)

3.6.2 Case A2

An asymmetric tooth profile with pressure angle of 30degree and 20degree for drive and coast sides (Table 3.5) respectively is constructed by using in-house software. Same gear pair is modeled as two comparable symmetric tooth profiles by using KissSoft. Drive and coast side profiles of asymmetric tooth are generated. Obtained curves from both tools are presented in Figure 3.44; it is clear that there is no difference or deviation between curves.

Table 3.5 Asymmetric tooth spur gear pair parameters (Case A2)

Parameter	Asymmetric gear	Representative symmetric gears		Unit
		Gear 1	Gear 2	
Module	1	1		mm
Teeth number	20	20		-
Pressure angle (drive/coast)	30/20	30/30	20/20	degree
Addendum coefficient	1.00	1.00		-
Dedendum coefficient	1.25	1.25		-
Cutter tip radius coefficient	0.11	0.11		-

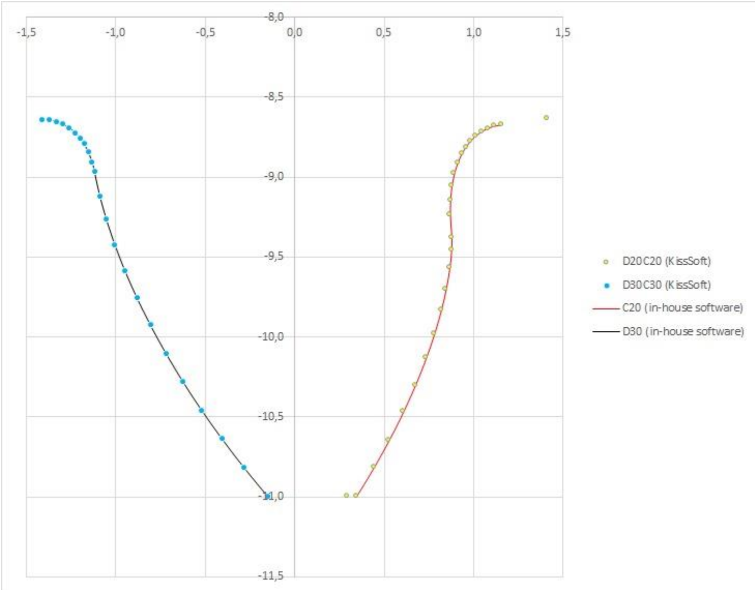


Figure 3.44 Tooth profile for Case A2
(obtained from KissSoft and in-house software)

3.6.3 Case A3

An asymmetric gear pair whose parameters are identical with gear pair given in Case A2 (Table 3.5) except pressure angle is presented here. Asymmetric tooth spur gear with pressure angle values of 25degree and 20degree for drive and coast sides respectively were constructed by in-house software and KissSoft. These tooth profiles are identical, they do not differ from each other (Figure 3.45).

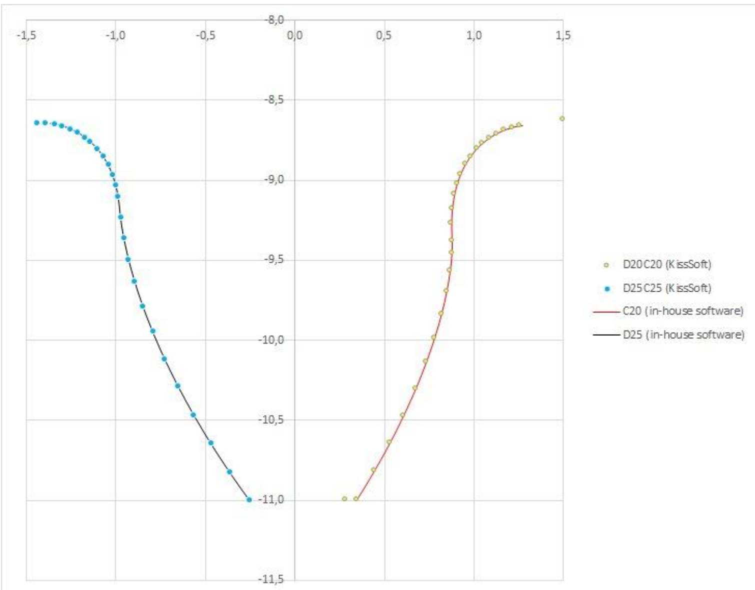


Figure 3.45 Tooth profile for Case A3
(obtained from KissSoft and in-house software)

Involute and trochoid profiles of gear tooth were constructed by using developed software and a commercial gear software, KissSoft. Generated macro geometries by these both tools do not differ from each other and there is no deviation between curves. In case of symmetric tooth profile spur gears, KissSoft is an efficient tool for macro geometry generation and stress calculation whereas no tool is encountered for asymmetric spur gears during literature surveys.

To compare macro geometry of asymmetric tooth spur gear obtained by in-house software, a different solution is discovered. Idea behind this solution is that assuming two different comparable symmetric tooth profiles for both drive and coast sides of asymmetric spur gear. Then generated comparable symmetric profiles refer to drive and coast sides of tooth respectively by KissSoft were compared to asymmetric profile constructed by in-house software.

All tooth profiles including involute tooth and trochoidal root sections coincides for both tools regardless of symmetric and asymmetric gears. These all give weight to idea of using in-house tool’ module for macro geometry generation of symmetric and asymmetric spur gears.

CHAPTER 4

TOOTH ROOT BENDING STRESS CALCULATIONS FOR SPUR GEARS

Gears are significant power transmission elements and they are used to transmit power, torque and speed between parallel and non-parallel shafts. Gear pairs are required to be designed for a safe meshing and required service life under intended design torque. Gears need to be evaluated for stresses at tooth root and surface flanks of gears. There are well accepted formulae such as Lewis and AGMA equations, gear standards namely ISO [16], AGMA [18] and DIN [17].

ISO 6336:2006 [16] is one of the international gear standards and includes different parts given in Table 4.1. Each part consists of different aspects of gears such as calculations of introduction and general influence factors, basic principles, tooth root bending stress, surface pitting, strength and quality of gear materials and service life under variable load.

Many of gears usually rotate in unidirectional for many applications without reverse motion. One of the gear flanks (drive) is highly loaded for long service life while the other side (coast) is lightly loaded for short time or unloaded. An asymmetric gear (Figure 4.1) is alternative to symmetric tooth gears for power transmission especially for unidirectional movement.

Asymmetric spur gear is presented by the two different involute profiles of different base circles in the Figure 4.2. Pressure angle of drive side of tooth differs from other side (coast) to provide stronger tooth of asymmetric spur gears.

Gears are needed to check for tooth root bending stress regardless of gear tooth profile of symmetric or asymmetric. For this aim, there is a need for tools to estimate if asymmetric gear pairs are going to be in safe or not during meshing.

Table 4.1 ISO 6336:2006 parts

Standard	Part	Scope
ISO 6336:2006	1	Basic principles, introduction and general influence factors
ISO 6336:2006	2	Calculation of surface durability (pitting)
ISO 6336:2006	3	Calculation of tooth bending strength
ISO 6336:2006	5	Strength and quality of materials
ISO 6336:2006	6	Calculation of service life under variable load
ISO 53:1998	-	Cylindrical gear for general and heavy engineering -Standard basic rack tooth profile
ISO 54:1996	-	Cylindrical gear for general and heavy engineering - Module

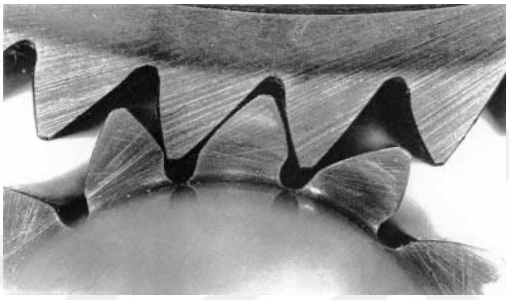


Figure 4.1 Asymmetric spur gear

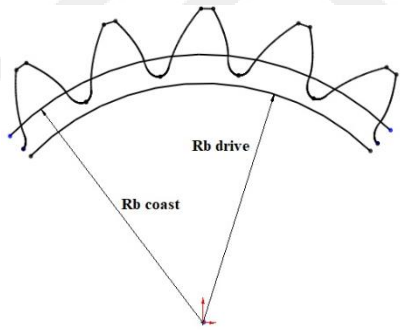


Figure 4.2 Asymmetric tooth gear with two different base circles

No standard for asymmetric spur gears is encountered during literature survey. Therefore any gear designer who wants to investigate asymmetric gear performance in terms of root stress can modify formulae given for symmetric ones, conduct finite element analyses or carry out measurements. The first one, modifying formulae given for symmetric spur gears is investigated to improve a tool for root stress calculation of asymmetric gears in this chapter.

Using gear standards or equations presented for symmetric gears directly for asymmetric gears is not possible. Because asymmetric gears differ from symmetric gears in terms of pressure angle and so tooth profiles, tooth thickness and root stress.

Design and analysis of asymmetric spur gears calls for some modifications of formulae presented in ISO 6336 Part 3 [16].

4.1 Modification Of Formulae Presented In Gear Standards

ISO 6336 Part 3 [16] presents fundamental formulae for root stress calculations of involute external or internal parallel axes gears. It recommends following formulae to determine form factor (Y_F), stress concentration factor (Y_S) and root stress of symmetric tooth spur gears given in ISO [16]:

$$\sigma_{F0} = \frac{F_t}{b\, m_n}\, Y_F\, Y_S\, Y_\beta\, Y_B\, Y_{DT} \tag{4.1}$$

$$Y_F = \frac{\frac{6h_{Fe}}{m_n} \cos \alpha_{Fen}}{\left(\frac{S_{Fn}}{m_n}\right)^2 \cos \alpha_n} \tag{4.2}$$

$$Y_s = (1.2 + 0.13L) q_s^{\left[\frac{1}{1.21+\frac{2.3}{L}}\right]} \tag{4.3}$$

$$L = \frac{s_{Fn}}{h_{Fe}} \tag{4.4}$$

$$q_s = \frac{s_{Fn}}{2\rho_F} \tag{4.5}$$

Abbreviations for these equations are given in Table 4.2.

Some modifications on formulae of stress modifying factors namely form and stress concentration factors are needed for asymmetric spur gears. Tooth root thickness and bending moment arm are very significant parameters for calculation of form factor and stress concentration factor so tooth root bending stress.

In this chapter, tooth root critical section thickness for asymmetric gear are determined by modelling asymmetric spur gear as two different symmetric spur gears (one for drive side and other one for coast side) as shown in Figure 4.4.

Table 4.2 Parameters required for equations of 4.1-4.5

Symbol	Unit	Explanation
F_t	Newton	Nominal tangential load
b	mm	Face width
m_n	mm	Normal module
Y_F	-	Form factor
Y_S	-	Stress concentration factor
Y_β	-	Helix angle factor
Y_B	-	Rim thickness factor
Y_{DT}	-	Deep tooth factor
α_{Fen}	Radian	Load direction angle, relevant to direction of application of load at the outer point of single pair tooth contact of virtual spur gears
α_n	Degree	Normal pressure angle
h_{Fe}	mm	Bending moment arm for tooth root stress relevant to load application at the outer point of single pair tooth contact
s_{Fn}	mm	Tooth root chord at the critical section
ρ_F	mm	Radius of root fillet
q_s	-	Notch parameter; $q_s = s_{Fn} / 2\rho_F$

Two non-identical root thicknesses for gear tooth models of drive and coast flanks (Figure 4.4) are summed up and then divided into two. This calculated thickness is accepted as tooth thickness at root of asymmetric spur gear. Length of bending moment arm is determined by taking this new thickness into account. Two significant stress modifying factors; form factor (Y_F) and stress concentration factor (Y_S) are evaluated by depending on tooth root thickness and bending moment arm of asymmetric gear calculated by this method. Tooth root stress is calculated based on all these parameters and factors.

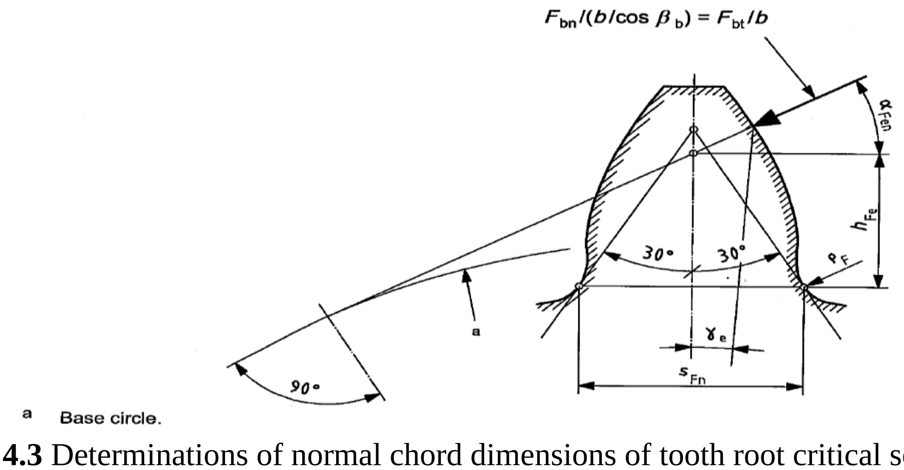


Figure 4.3 Determinations of normal chord dimensions of tooth root critical section [16]

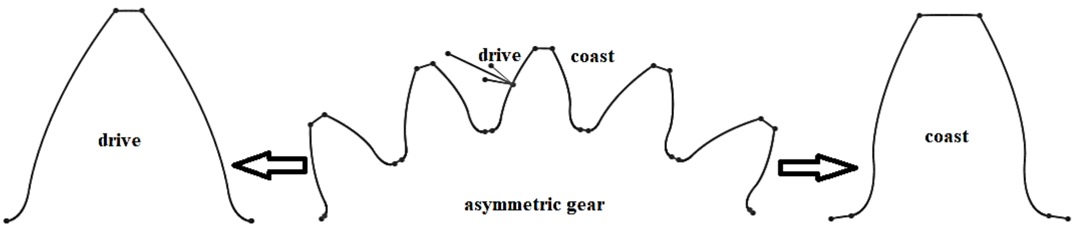


Figure 4.4 Modelling asymmetric gear as two symmetric gears

4.2 Root Stress Calculation Software For Asymmetric Spur Gears

Owing to their high load carrying capacity, asymmetric spur gears are in general use as an alternative to spur gears with symmetric tooth profile. With their increasing use, it is required to carry out more and more studies to evaluate asymmetric gears performance under load. A software which can achieve evaluation of root stress in addition to macro geometry generation process (presented in Chapter 3) has been developed and presented in this chapter.

A module of software to determine root bending stress of asymmetric spur gear is presented in Figure 4.5. Module, teeth numbers, pressure angles of both flanks, coefficients of addendum, dedendum, profile shift and root fillet radii can entered as inputs as shown in Figure 4.6-4.9. Tooth root thickness, length of bending moment arm, form factor (Y_F), stress concentration factor (Y_S) and other related parameters and tooth root bending stress are calculated and presented in Figure 4.10.

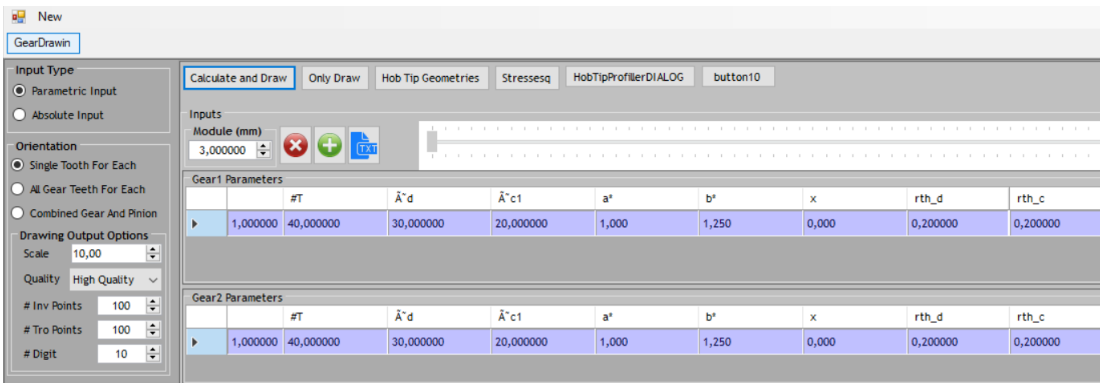


Figure 4.5 Tooth root bending stress calculation of asymmetric spur gear (in-house software)

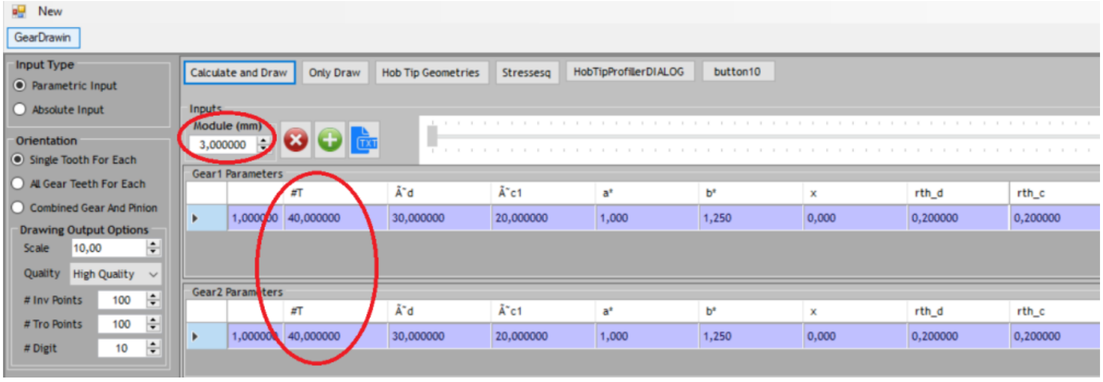


Figure 4.6 Selections of module and teeth numbers (in-house software)

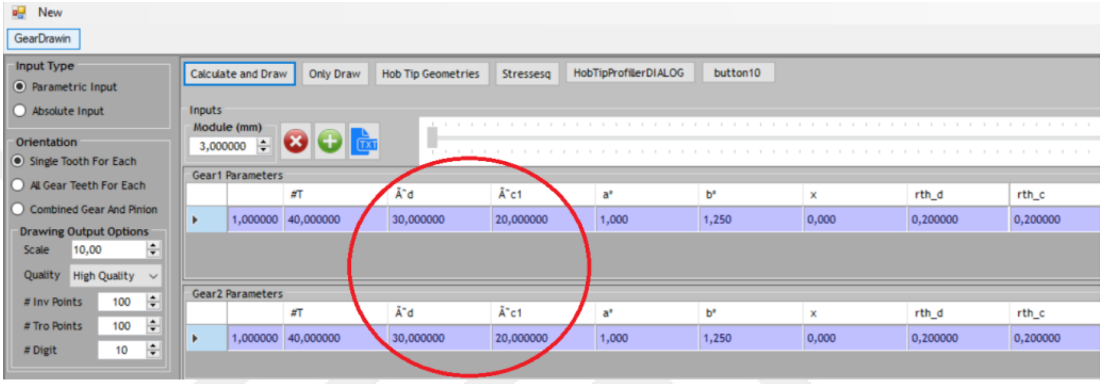


Figure 4.7 Selections of pressure angles of drive and coast sides (in-house software)

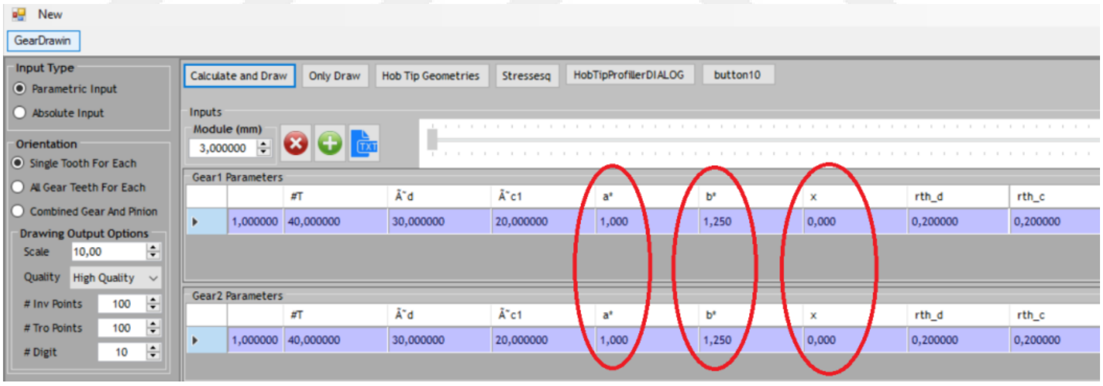


Figure 4.8 Selections of coefficients of addendum, dedendum and profile shift (in-house software)

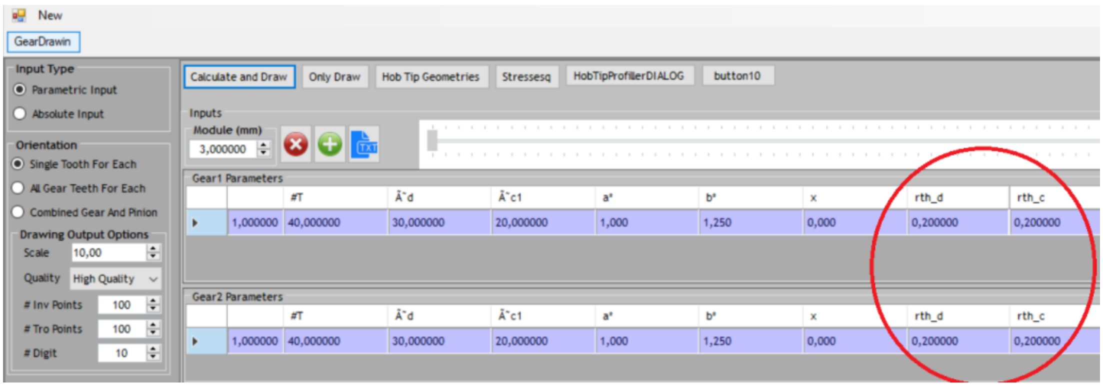


Figure 4.9 Selection of coefficients of root fillet radii (in-house software)

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	1,206023	1,206023	mm
Radius of Root Fillet (Coast Side)	1,229115	1,229115	mm
AB1 (Drive Side)	0,866025	0,866025	-
AAA1 (Drive Side)	35,623027	35,623027	-
AAA2 (Drive Side)	326,483883	326,483883	-
AAA3 (Drive Side)	0,009446	0,009446	-
AAA4 (Drive Side)	3,083955	3,083955	-
AAA5 (Drive Side)	2699,999955	2699,999955	-
den (Drive Side)	122,617962	122,617962	mm
E (Drive Side)	-0,155279	0,571182	mm
G(Drive Side)	-1,050000	-1,050000	mm
H (Drive Side)	-0,966070	-0,978177	mm
Ang (Drive Side)	0,147263	0,136618	mm
Theta (Drive Side)	0,899935	0,910579	mm
Gama E (Drive Side)	0,026279	0,026279	Radian
Form Factor Pressure Angle in Radian (Drive Side)	0,559472	0,559472	Radian
Form Factor Pressure Angle in Degree (Drive Side)	32,055405	32,055405	Degree
Involute Function of Normal Pressure Angle (Drive Side)	0,053751	0,053751	Radian
Involute Function of Form Factor Pressure Angle (Drive Side)	0,066742	0,066742	Radian
Bending Moment Arm for Tooth Root Stress (Drive Side)	3,819953	3,819953	mm
Bending Moment Arm for Tooth Root Stress (Coast Side)	3,071520	3,071520	mm
Tooth Root Normal Chord 1	7,792081	7,792081	mm
Tooth Root Normal Chord 2	6,407793	6,407793	mm
Tooth Root Normal Chord	7,099937	7,099937	mm
Tooth Root Normal Chord (Hypotenuse)	7,139276	7,139276	mm
AAB4(Drive Side)	4,850618	4,850618	-
AAB5(Drive Side)	6,579398	6,579398	-
Form Factor (Drive Side)	1,356404	1,356404	-
L (Drive Side)	1,858645	1,858645	-
AAB6(Drive Side)	0,408587	0,408587	-
Notch Parameter (Drive Side)	2,943534	2,943534	-
Stress Correction Factor (Drive Side)	2,240916	2,240916	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,794331	0,794331	-
YF*YS*YEps (Drive Side)	2,414438	2,414438	-
Pitchline Velocity	15,707963		m/sec
TangentialLoad	2673,803044		Newton
Face Width	20,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	135,454330	135,454330	MPa

Figure 4.10 Tooth root bending stress parameters' results (in-house software)

4.3 Case Studies Of Tooth Root Bending Stress

Some case studies for root bending stress evaluation of spur gears with symmetrical and asymmetrical tooth shapes are presented in this section. While root stress results of in-house software and KissSoft are compared for symmetric spur gears, there is no commercial tool for asymmetric gears. Therefore, stress values evaluated by in-house software for asymmetric gears are compared to papers in literature.

4.3.1 Case Studies Of Symmetric Spur Gears

Case studies for tooth root bending stress of symmetric spur gears are presented here. Results of some papers in literature are initially compared to in-house software's results by analyzing same symmetric gear pairs for root stress.

4.3.1.1 Case S1

Spitas et al [48] investigated root bending stress of a gear pair whose parameters are given in Table 4.3. Initially same gear pair was analyzed by using developed software for different alternatives of dedendum and clearance. Root stress values obtained from in-house software, KissSoft and reference study are presented in Table 4.4 and 4.5. Whereas the difference between stress values between in-house software and Ref [48] is 12.8% (Table 4.4) there is no difference of in-house software and KissSoft values (Table 4.5).

Table 4.3 Gear pair parameters [48]

Parameter	Value	Unit
Module	1	mm
Pressure angle	20	Degree
Teeth number	20	-
Addendum coefficient	1.0	-

Table 4.4 Tooth root stress of in-house software and reference study [48]

Standard	Profile	Cutter tip radius coefficient	Dedendum coefficient	Root stress (MPa)	
				Ref [48]	in-house software
ISO 53	A	0.38	1.25	2.51	2.86
ISO 53	B	0.30	1.25	2.65	3.02
ISO 53	C	0.25	1.25	2.73	3.13
Optimum	-	0.47	1.12	2.24	2.55

Table 4.5 Tooth root stress of in-house software and KissSoft [7]

Standard	Profile	Cutter tip radius coefficient	Dedendum coefficient	Root stress (MPa)	
				in-house software	KissSoft
ISO 53	A	0.38	1.25	2.86	2.86
ISO 53	B	0.30	1.25	3.02	3.02
ISO 53	C	0.25	1.25	3.13	3.14
Optimum	-	0.47	1.12	2.55	2.55

4.3.1.2 Case S2

Root bending stress was studied here for a gear pair whose parameters are presented in Table 4.6. KissSoft and in-house software results are presented in Table 4.7

Table 4.6 Gear pair parameters - Case S2

Parameter	Value	Unit
Module	3	mm
Pressure angle	20	Degree
Teeth number	40/40	-
Torque	160.43	Nm

Table 4.7 Tooth root stress of in-house software and KissSoft for symmetric spur gears

Gear Pair	Coefficients of cutter tip radius	Tooth Root Stress (MPa)	
		in-house software	KissSoft [13]
1	0.10	147.50	147.18
2	0.15	140.55	140.50
3	0.20	134.39	134.35
4	0.25	128.68	128.65
5	0.30	123.13	123.33
6	0.35	118.19	118.35
7	0.38	115.50	115.50
8	0.40	113.53	113.65
9	0.45	109.12	109.21
10	0.47	107.40	107.50

Under this section (Case study of S2), many case studies were performed and presented actually for different values of hob tip radius coefficients of symmetric tooth profile spur gears. Stress calculations were done by using in-house software and KissSoft. It is possible to say that these two tools give same results with a highest difference of 0.2% regardless of cutter tip radius coefficients. These all show that developed software calculates bending stress at tooth root correctly.

4.3.1.3 Case Study - S3

Sekar et al [23] studied on symmetric and asymmetric spur gears for tooth root bending stress. In this section, symmetric case studies are compared to software results. Gear pair parameters are presented in Table 4.8 and results are given in Table 4.9. Root stress versus contact point along length of contact is presented in Figure 4.11 in reference paper. Maximum bending stress is 26.3 MPa in this curve. Some screen shots from in-house software which shows input and outputs are presented in Figure 4.12-4.15. Results of both studies are very close to each other.

Table 4.8 General gear parameters [23]

Parameter	Gear Pair 1	Gear Pair 2	Unit
Module	1.0		mm
Teeth number (driver gear)	20		-
Teeth number (driven gear)	20		-
Pressure angle	20	30	degree
Addendum coefficient	1.00		-
Dedendum coefficient	1.25		-
Cutter tip radius coefficient (fully rounded)	0.47	0.11	-

Table 4.9 Tooth root thickness, bending moment arm and root bending stress [23]

Gear Pair	Ref [23]			In-house software		
	Root thickness (mm)	Moment arm (mm)	Root stress (MPa)	Root thickness (mm)	Moment arm (mm)	Root stress (MPa)
1	1.96	1.05	26.30	1.96	1.05	26.83
2	2.36	1.39	26.70	2.36	1.39	27.56

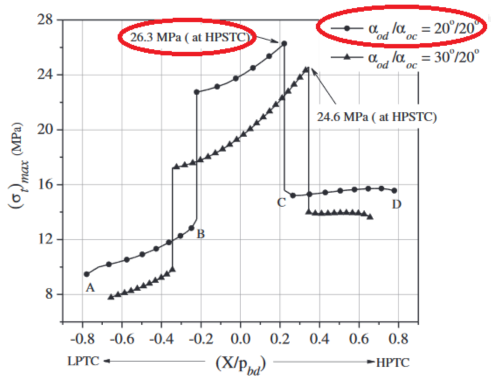


Figure 4.11 Tooth root stress for Gear Pair 1 - Case S3 [23]

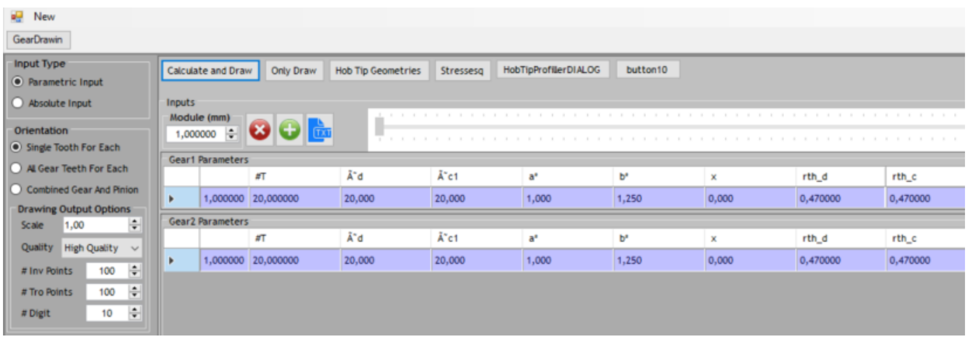


Figure 4.12 Gear parameters of Case S3 - Gear Pair 1 (in-house software)

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	0,628801	0,628801	mm
Radius of Root Fillet (Coast Side)	0,628801	0,628801	mm
Tooth Root Normal Chord	1,956641	1,956641	mm
Tooth Root Normal Chord (Hypotenuse)	1,956641	1,956641	mm
AAB4(Drive Side)	3,597562	3,597562	-
AAB5(Drive Side)	5,945487	5,945487	-
Form Factor (Drive Side)	1,652643	1,652643	-
L (Drive Side)	1,861455	1,861455	-
AAB6(Drive Side)	0,408899	0,408899	-
Notch Parameter (Drive Side)	1,555852	1,555852	-
Stress Correction Factor (Drive Side)	1,727657	1,727657	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,731746	0,731746	-
YF*YS*YEps (Drive Side)	2,089281	2,089281	-
Pitchline Velocity	2,617994		m/sec
TangentialLoad	9,396926		Newton
Face Width	1,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	26,830118	26,830118	MPa

Figure 4.13 Tooth root stress parameters of Case S3 - Gear Pair 1 (in-house software)

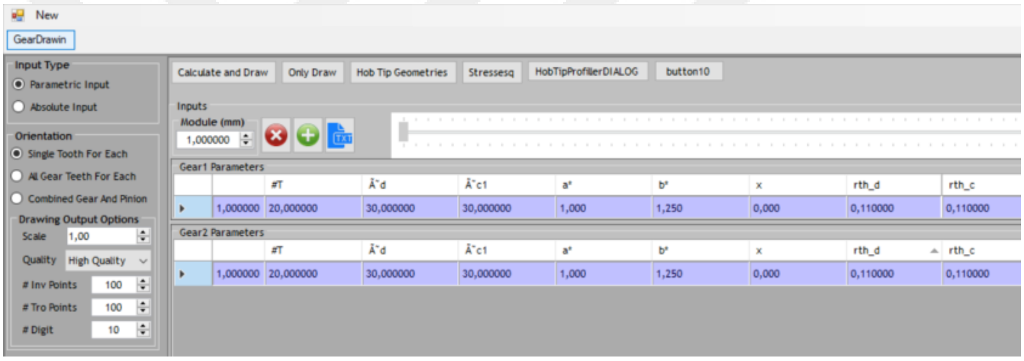


Figure 4.14 Gear parameters of Case S3 - Gear Pair 2 (in-house software)

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	0,403469	0,403469	mm
Radius of Root Fillet (Coast Side)	0,403469	0,403469	mm
Tooth Root Normal Chord	2,360625	2,360625	mm
Tooth Root Normal Chord (Hypotenuse)	2,360625	2,360625	mm
AAB4(Drive Side)	4,825968	4,825968	-
AAB5(Drive Side)	7,122786	7,122786	-
Form Factor (Drive Side)	1,475929	1,475929	-
L (Drive Side)	1,691699	1,691699	-
AAB6(Drive Side)	0,389169	0,389169	-
Notch Parameter (Drive Side)	2,925411	2,925411	-
Stress Correction Factor (Drive Side)	2,156202	2,156202	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,822432	0,822432	-
YF*YS*YEps (Drive Side)	2,617307	2,617307	-
Pitchline Velocity	2,617994		m/sec
TangentialLoad	8,660254		Newton
Face Width	1,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	27,560401	27,560401	MPa

Figure 4.15 Tooth root stress parameters of Case S3 - Gear Pair 2 (Software)

4.3.2 Case Studies Of Asymmetric Spur Gears

Several case studies for root stress calculations of asymmetric tooth spur gears are presented based on only literature works in this section because lack of a tool as software or any other means.

4.3.2.1 Case A1 and A2

Sekar et al [23] studied on symmetric and asymmetric tooth spur gears for bending tooth root bending stress. This reference study consists of numerical works of finite element analyses and analytical calculations based on ISO [16]. Common and different parameters of gear pairs of Case A1 and A2 are given in Table 4.10 and 4.11.

These parameters are entered to developed software and some figures are shown for Case A1 and A2 (Figure 4.16-4.19). Stress versus contact position along contact path constructed in reference paper for Case A1 is presented in Figure 4.20 whereas stress and related parameters taken from in-house software are given for Case A1 and A2 in Figure 4.21 and 4.22 respectively. Tooth root thickness, bending moment arm and root bending stress values are presented in Table 4.12.

Table 4.10 General gear parameters of Case Studies of A1 and A2 [23]

Parameter	Value	Unit
Module	1	mm
Teeth number	20/20	-
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	Fully rounded	-
Center distance	20	mm

Table 4.11 Pressure angle values of Case Studies of A1 and A2 [23]

Gear Pair	Pressure Angle		Unit
	Drive Side	Coast Side	
A1	30	20	Degree
A2	25	20	Degree

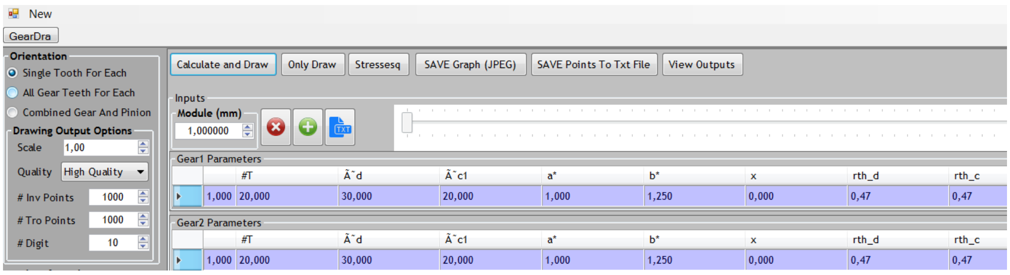


Figure 4.16 Input parameters - Case A1 (in-house software)

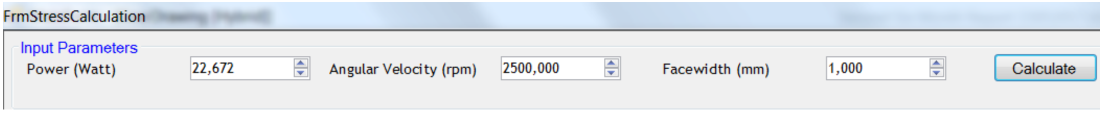


Figure 4.17 Loading parameters - Case A1 (in-house software)

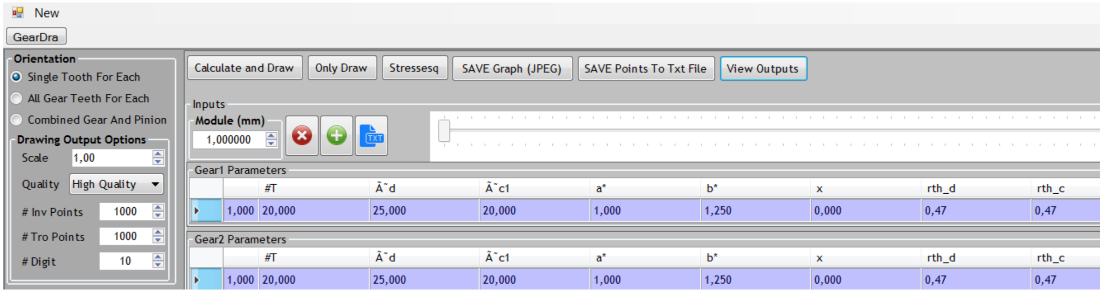


Figure 4.18 Input parameters - Case A2 (in-house software)

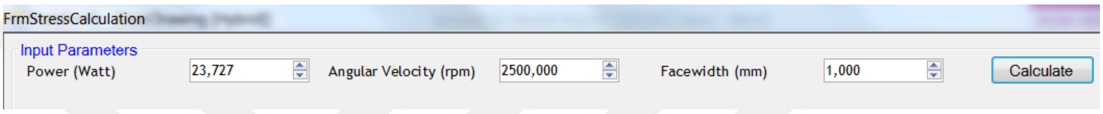


Figure 4.19 Loading parameters - Case A2 (in-house software)

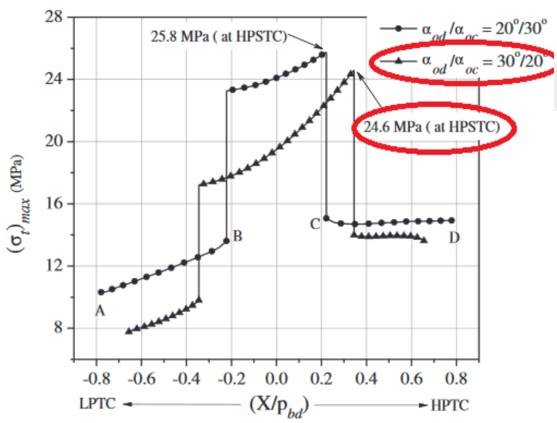
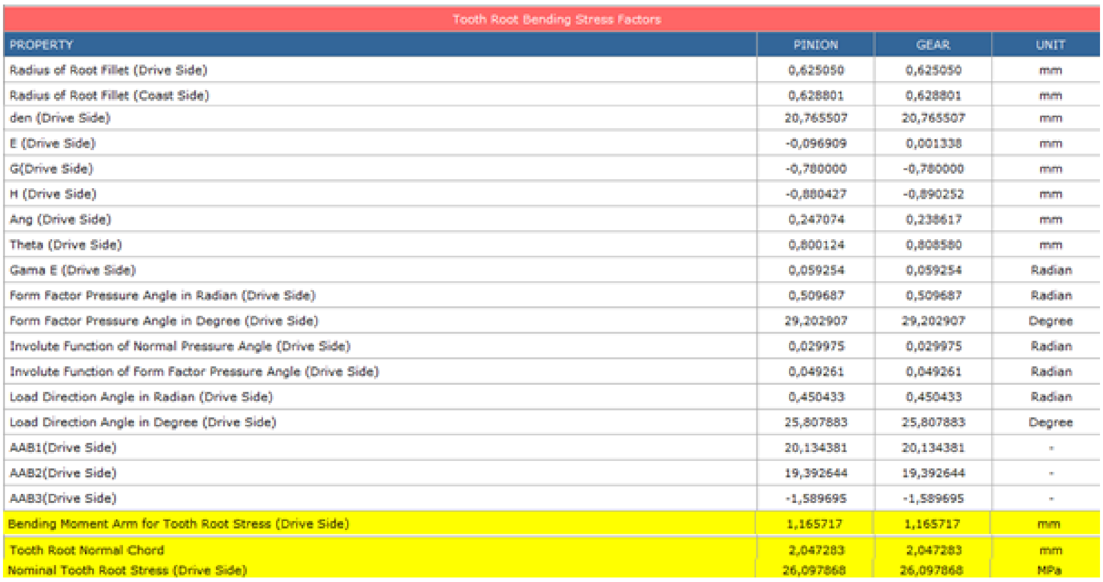


Figure 4.20 Tooth root stress – Case A1 [23]

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	0,620985	0,620985	mm
Radius of Root Fillet (Coast Side)	0,628801	0,628801	mm
den (Drive Side)	21,001355	21,001355	mm
E (Drive Side)	-0,207644	0,001338	mm
G(Drive Side)	-0,780000	-0,780000	mm
H (Drive Side)	-0,869353	-0,890252	mm
Ang (Drive Side)	0,256630	0,238617	mm
Theta (Drive Side)	0,790568	0,808580	mm
Gama E (Drive Side)	0,047654	0,047654	Radian
Form Factor Pressure Angle in Radian (Drive Side)	0,601067	0,601067	Radian
Form Factor Pressure Angle in Degree (Drive Side)	34,438581	34,438581	Degree
Involute Function of Normal Pressure Angle (Drive Side)	0,053751	0,053751	Radian
Involute Function of Form Factor Pressure Angle (Drive Side)	0,084637	0,084637	Radian
Load Direction Angle in Radian (Drive Side)	0,553413	0,553413	Radian
Load Direction Angle in Degree (Drive Side)	31,708201	31,708201	Degree
AAB1(Drive Side)	20,359442	20,359442	-
AAB2(Drive Side)	19,345018	19,345018	-
AAB3(Drive Side)	-1,578834	-1,578834	-
Bending Moment Arm for Tooth Root Stress (Drive Side)	1,296629	1,296629	mm
Tooth Root Normal Chord	2,149233	2,149233	mm
Nominal Tooth Root Stress (Drive Side)	25,048913	25,048913	MPa

Figure 4.21 Output parameters - Case A1 (in-house software)



Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	0,625050	0,625050	mm
Radius of Root Fillet (Coast Side)	0,628801	0,628801	mm
den (Drive Side)	20,765507	20,765507	mm
E (Drive Side)	-0,096909	0,001338	mm
G(Drive Side)	-0,780000	-0,780000	mm
H (Drive Side)	-0,880427	-0,890252	mm
Ang (Drive Side)	0,247074	0,238617	mm
Theta (Drive Side)	0,800124	0,808580	mm
Gama E (Drive Side)	0,059254	0,059254	Radian
Form Factor Pressure Angle in Radian (Drive Side)	0,509687	0,509687	Radian
Form Factor Pressure Angle in Degree (Drive Side)	29,202907	29,202907	Degree
Involute Function of Normal Pressure Angle (Drive Side)	0,029975	0,029975	Radian
Involute Function of Form Factor Pressure Angle (Drive Side)	0,049261	0,049261	Radian
Load Direction Angle in Radian (Drive Side)	0,450433	0,450433	Radian
Load Direction Angle in Degree (Drive Side)	25,807883	25,807883	Degree
AAB1(Drive Side)	20,134381	20,134381	-
AAB2(Drive Side)	19,392644	19,392644	-
AAB3(Drive Side)	-1,589695	-1,589695	-
Bending Moment Arm for Tooth Root Stress (Drive Side)	1,165717	1,165717	mm
Tooth Root Normal Chord	2,047283	2,047283	mm
Nominal Tooth Root Stress (Drive Side)	26,097868	26,097868	MPa

Figure 4.22 Output parameters - Case A2 (in-house software)

Table 4.12 Tooth root thickness, bending moment arm and root bending stress [23]

Case	**Ref [23]**						**In-house software**		
	Root thickness (mm)		Moment arm (mm)		Root stress (MPa)		Root thickness (mm)	Moment arm (mm)	Root stress (MPa)
	ISO	FEA	ISO	FEA	ISO	FEA	ISO	ISO	ISO
A1	2.19	2.31	1.29	1.36	23.7	24.6	2.15	1.30	25.1
A2	2.07	2.15	1.17	1.23	24.5	25.3	2.05	1.17	26.1

4.3.2.2 Case A3

Kapelevich and Shekhtman [20] used a method for modeling an asymmetric spur gear as comparable symmetric gear teeth. Pressure angle values of drive and coast flanks of asymmetric tooth are summed up and divided in two. This is used as pressure angle of comparable symmetric tooth profile spur gears. Tooth root stress was determined based on comparable symmetric tooth spur gear by conducting finite element analyses. A conversion coefficient was determined to calculate root stress of an asymmetric gear by modelling it as comparable symmetric gears [20].

While gear pair parameters for asymmetric and comparable symmetric spur gears are presented in Table 4.13 and 4.14, root fillet radius coefficient value is missing (not given in reference paper). It is stated that root optimisation is performed to minimize root stress by authors. In this thesis, highest usable cutting tool tip radius is used for stronger tooth root less root stress. Some screen shots are presented for inputs and outputs (in-house software) in Figure 4.23-4.25. Tooth root stress values of Ref [20], current study and KissSoft are given for comparable symmetric gears in Table 4.15. Owing to lack of commercial tool to calculate root stress of asymmetric gears, stress values are presented for only reference paper and current study in same table.

Table 4.13 General gear parameters - Case A3

Parameter	Value	Unit
Module	5	mm
Teeth number (pinion/gear)	20/49	-
Torque (pinion)	900	Nm
Rotational speed (pinion)	1000	rpm
Center distance	172.5	mm

Table 4.14 Gear pair parameters - Case A3

Parameters	Asymmetric gear		Comparable symmetric gear		Unit
Pressure Angle (drive/coast)	35/20		27.5		Degree
Addendum coefficient	0.921	1.080	0.951		-
Dedendum coefficient	1.151	1.081	1.127		-
Profile shift coefficient	0.06	-0.06	0.06	-0.06	-
Root radius coefficient	-		0.327		-

Table 4.15 Comparison of tooth root stress with Ref [20]

Case A3 – Gear pairs	Tooth Root Stress (MPa)		
	Ref [20]	in-house software	KissSoft
Asymmetric	295	353	-
Comparable symmetric	309	336	336

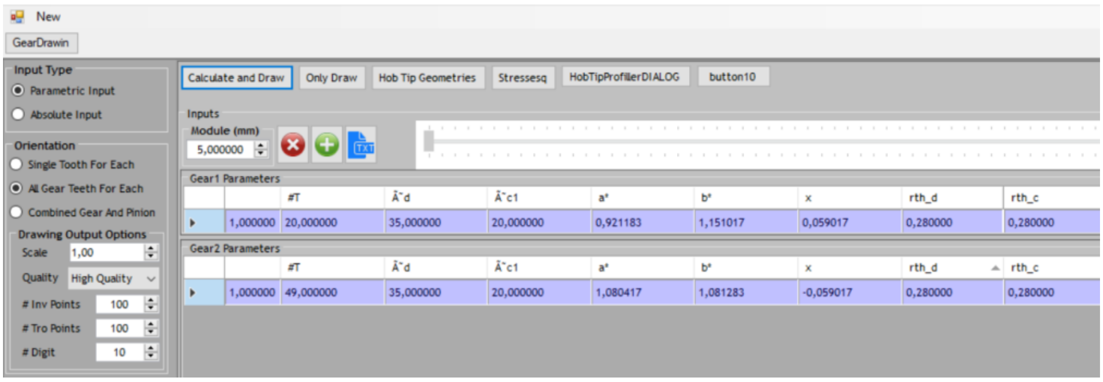


Figure 4.23 Gear pair parameters – Case A3 (in-house software)

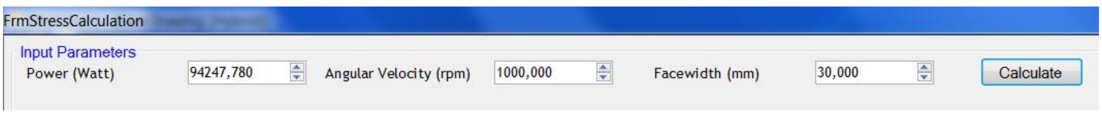


Figure 4.24 Loading parameters for Case A3 (in-house software)

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	2,215337	2,046280	mm
Radius of Root Fillet (Coast Side)	2,284423	2,075717	mm
Bending Moment Arm for Tooth Root Stress (Drive Side)	6,022198	6,831404	mm
Bending Moment Arm for Tooth Root Stress (Coast Side)	4,173818	4,979159	mm
Tooth Root Normal Chord 1	12,877452	13,482254	mm
Tooth Root Normal Chord 2	9,775553	10,454103	mm
Tooth Root Normal Chord	11,326502	11,968179	mm
Tooth Root Normal Chord (Hypotenuse)	11,476330	12,110661	mm
AAB4(Drive Side)	4,203549	4,693325	-
AAB5(Drive Side)	5,825743	6,623045	-
Form Factor (Drive Side)	1,385911	1,411163	-
L (Drive Side)	1,880792	1,751935	-
AAB6(Drive Side)	0,411034	0,396380	-
Notch Parameter (Drive Side)	2,556383	2,924374	-
Stress Correction Factor (Drive Side)	2,124548	2,184634	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,836732	0,836732	-
YF*YS*YEps (Drive Side)	2,463701	2,579537	-
Pitchline Velocity	5,235988		m/sec
TangentialLoad	18000,000000		Newton
Face Width	30,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	353,332037	369,944783	MPa

Figure 4.25 Output parameters - Case A3 (in-house software)

4.3.2.3 Case A4-A5-A6-A7

Cases of A4-A5-A6-A7 were based on finite element analyses and calculations of ISO [16] based calculations of asymmetric tooth spur gears. Gear pair parameters for all these cases are given in Table 4.16. Based on given gear parameters, non-identical pressure angle values for drive and coast sides of tooth profile were used. Combinations of 20, 25 and 30degree such as D20C25, D20C30, D25C20 and D30C20 were analyzed. Results of both studies are given in Table 4.17. Some screen shots for root stress and related parameters from software can be seen in Figure 4.26-4.29.

Table 4.16 Gear pair parameters for Cases of A4-A5-A6-A7

Gear pair parameter	Value	Unit
Module	3	mm
Teeth number	40	-
Gear ratio	1	-
Addendum factor	1.00	-
Dedendum factor	1.25	-
Face width	20	mm
Rack cutter tip radius coefficient	0.1	-
Torque	160.428	Nm

Table 4.17 Tooth root stress values for Cases of A4-A5-A6-A7 (in-house software and FEA)

Case No	Gear Pair	Tooth root stress (MPa)	
		in-house software	FEA
A4	D20C25	138.61	128.5
A5	D20C30	130.43	121.6
A6	D25C20	147.38	129.7
A7	D30C20	146.94	121.3

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	1,039317	1,039317	mm
Radius of Root Fillet (Coast Side)	1,025811	1,025811	mm
Tooth Root Normal Chord 1	6,414059	6,414059	mm
Tooth Root Normal Chord 2	7,107077	7,107077	mm
Tooth Root Normal Chord	6,760568	6,760568	mm
Tooth Root Normal Chord (Hypotenuse)	6,769981	6,769981	mm
AAB4(Drive Side)	4,772102	4,772102	-
AAB5(Drive Side)	6,008110	6,008110	-
Form Factor (Drive Side)	1,259007	1,259007	-
L (Drive Side)	2,127371	2,127371	-
AAB6(Drive Side)	0,436463	0,436463	-
Notch Parameter (Drive Side)	3,252408	3,252408	-
Stress Correction Factor (Drive Side)	2,470637	2,470637	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,687692	0,687692	-
YF*YS*YEps (Drive Side)	2,139100	2,139100	-
Pitchline Velocity	15,707963		m/sec
TangentialLoad	2673,803044		Newton
Face Width	20,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	138,616621	138,616621	MPa

Figure 4.26 Tooth root stress calculated by software for Case A4

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	1,039317	1,039317	mm
Radius of Root Fillet (Coast Side)	1,011460	1,011460	mm
Tooth Root Normal Chord 1	6,414059	6,414059	mm
Tooth Root Normal Chord 2	7,863566	7,863566	mm
Tooth Root Normal Chord	7,138812	7,138812	mm
Tooth Root Normal Chord (Hypotenuse)	7,178039	7,178039	mm
AAB4(Drive Side)	5,321024	5,321024	-
AAB5(Drive Side)	6,008110	6,008110	-
Form Factor (Drive Side)	1,129127	1,129127	-
L (Drive Side)	2,246394	2,246394	-
AAB6(Drive Side)	0,447655	0,447655	-
Notch Parameter (Drive Side)	3,434376	3,434376	-
Stress Correction Factor (Drive Side)	2,592107	2,592107	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,687692	0,687692	-
YF*YS*YEps (Drive Side)	2,012748	2,012748	-
Pitchline Velocity	15,707963		m/sec
TangentialLoad	2673,803044		Newton
Face Width	20,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	130,428856	130,428856	MPa

Figure 4.27 Tooth root stress calculated by software for Case A5

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	1,025811	1,025811	mm
Radius of Root Fillet (Coast Side)	1,039317	1,039317	mm
Tooth Root Normal Chord 1	7,107077	7,107077	mm
Tooth Root Normal Chord 2	6,414059	6,414059	mm
Tooth Root Normal Chord	6,760568	6,760568	mm
Tooth Root Normal Chord (Hypotenuse)	6,769981	6,769981	mm
AAB4(Drive Side)	4,602561	4,602561	-
AAB5(Drive Side)	6,409829	6,409829	-
Form Factor (Drive Side)	1,392666	1,392666	-
L (Drive Side)	1,912590	1,912590	-
AAB6(Drive Side)	0,414498	0,414498	-
Notch Parameter (Drive Side)	3,295230	3,295230	-
Stress Correction Factor (Drive Side)	2,374773	2,374773	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,746261	0,746261	-
YF*YS*YEps (Drive Side)	2,468084	2,468084	-
Pitchline Velocity	15,707963		m/sec
TangentialLoad	2673,803044		Newton
Face Width	20,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	147,382898	147,382898	MPa

Figure 4.28 Tooth root stress calculated by software for Case A6

Tooth Root Bending Stress Factors			
PROPERTY	PINION	GEAR	UNIT
Radius of Root Fillet (Drive Side)	1,011460	1,011460	mm
Radius of Root Fillet (Coast Side)	1,039317	1,039317	mm
Tooth Root Normal Chord 1	7,863566	7,863566	mm
Tooth Root Normal Chord 2	6,414059	6,414059	mm
Tooth Root Normal Chord	7,138812	7,138812	mm
Tooth Root Normal Chord (Hypotenuse)	7,178039	7,178039	mm
AAB4(Drive Side)	4,903882	4,903882	-
AAB5(Drive Side)	6,764282	6,764282	-
Form Factor (Drive Side)	1,379373	1,379373	-
L (Drive Side)	1,817743	1,817743	-
AAB6(Drive Side)	0,403991	0,403991	-
Notch Parameter (Drive Side)	3,528963	3,528963	-
Stress Correction Factor (Drive Side)	2,390514	2,390514	-
Contact Ratio Factor for Tooth Root Stress(Drive Side)	0,794331	0,794331	-
YF*YS*YEps (Drive Side)	2,619235	2,619235	-
Pitchline Velocity	15,707963		m/sec
TangentialLoad	2673,803044		Newton
Face Width	20,000000		mm
Helix Angle Factor	1,000000		-
Rim Thickness Factor	1,000000		-
Deep Tooth Factor	1,000000		-
Nominal Tooth Root Stress (Drive Side)	146,943783	146,943783	MPa

Figure 4.29 Tooth root stress calculated by software for Case A7

4.4 Conclusion

In this chapter, a software has been developed for root stress evaluation of spur gears with symmetrical and asymmetrical tooth profile. Tooth root stress for spur gears with both profiles can be determined via same software based on ISO standard [16] for symmetric spur gears and some modified formulae of Ref [16] for asymmetric spur gears.

Several case studies have been conducted to verify results of in-house software by comparing with those of paper presented in literature for symmetric and asymmetric spur gears and KissSoft for symmetric spur gears

By keeping practical applications in mind, calculations and comparison have been done for constant torque transmission unless otherwise stated in reference studies. Variation of loading position (Highest Point of Single Tooth Contact - HPSTC) is taken into account with the increase or decrease of drive side pressure angle of tooth profile for all calculations.

In this study, all calculations are based on an already manufactured gearbox whose operating conditions and geometrical limitations predetermined and not prone to be changed. Therefore, any gear pair, symmetric or asymmetric, for a constant center distance and input-output values of power and speed can be used in these analyses.

Increase in pressure angle of drive side flank yields reduction in base circle radius. Therefore tooth load and bending moment arm starts to increase with increase of root thickness. Tooth root thickness produces lower stress whereas increase of load and moment arm results in higher bending stress. To evaluate effect of pressure angle on different gear parameters such as base radius, loading point position (HPSTC), root thickness, bending moment arm, tooth load and so on root bending stress, several case studies are carried out.

When looking at all case studies, in-house software guesses stress values higher than literature works regardless of symmetric or asymmetric spur gear pairs. Tooth root bending stress values obtained from in-house software and KissSoft are almost same.

CHAPTER 5

TRANSMISSION ERROR THEORY AND MICRO GEOMETRY DESIGN

Gears are critical power transmission elements and their design requires to be given importance to obtain a safe operation during meshing. Gear design necessitates taking static and dynamic conditions into account at same time. Because dynamic performance is as important as performance under static loading conditions. It is directly related to gear vibration level. Vibration is an outcome of non-uniform distribution of load and yields noise during operation of gear pairs.

Transmission error (TE) has been accepted as main source of vibration and noise of gear pairs during meshing. It is unwanted owing to its severe effect on gear performance. One of the main aims of this thesis is to minimize quasi static transmission error and obtain smooth transmission error curves under loading. Tip relieving is applied to fulfill the requirements given above.

Transmission error (TE) is defined as the difference between the effective and the ideal position of the output shaft with reference to the input shaft. The ideal position represents a condition of perfect gear box, without geometrical errors and deflections. TE can be expressed either by an angular displacement or, more conveniently, as a linear displacement measured along a line of action at base circle [33].

In other words, (TE) is defined as the difference between the actual position of the output gear and the position it would occupy if the gears were perfectly conjugate. It can be defined in angular units, or can be expressed in linear units along the line of action. The TE is generally converted to linear motion along the line of action which is tangential to the base circles of the mating gears. The mathematical formulation of TE in linear units is given in Eq. 5.1:

$$\mathrm{TE} = r_{b2} * (\theta_2 - \frac{T_1}{T_2} * \theta_1) \tag{5.1}$$

θ_1: Angular rotation of the input gear

θ_2: Angular rotation of the output gear

T_1: Tooth number of the driving gear

T_2: Tooth number of the driven gear

r_{b2}: Base radius of the output gear

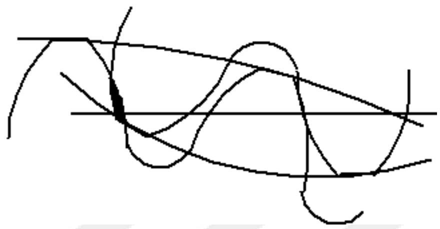


Figure 5.1 Corner contact [33]

The quasi static transmission error is directly related to the kinematic accuracy of gears, dynamic tooth load and noise. It can be defined as the difference between the theoretical position of the output gear with perfect accuracy and the actual output position [33].

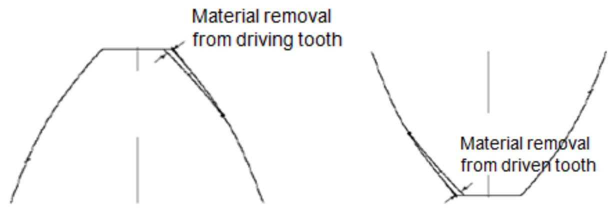
Generating gear pair with minimum geometrical error is very significant to decrease transmission error. However, transmission error occurs under loading owing to tooth flexibility regardless of gear profile with and without error. There are two impactful parameters which yield corner contact and transmission error (Figure 5.1); tooth deflection under load and adjacent pitch error.

Corner contact has a significant effect on instantaneous variation in transmission error curve with damage on tooth surface during heavily loaded operation. To avoid corner contact and obtain smooth transmission error curves under loads, intentional profile relief is applied (Figure 5.2).

Linear tip relieving is defined by two relief parameters of amount of relief and extent of relief (Figure 5.3). Amount of relief is the material thickness removed from tooth tip whereas extent of relief refers distance of how far the relief extends down the tip (Figure 5.3).

Extent of relief can be defined in linear or angular units (Eq. 5.2):

$$\text{Extent of relief} = r_b * (\text{roll angle}_{\text{end}} - \text{roll angle}_{\text{start}}) \quad (5.2)$$



(a) Driving gear (b) Driven gear

Figure 5.2 Application of tip relief for driving and driven teeth to avoid corner contact

Construction of loaded and unloaded transmission error curves for gear pairs with profile relief is shown in Figure 5.3. Gear pair conjugates with no adjacent pitch error, constant stiffness and load. The vertical axis is transmission error axis whereas horizontal axis refers to contact position along line of contact. When double tooth pair is in contact, it is called as double contact (DC) and total load on tooth is shared by these pairs. But if it is in single contact (this means only one gear pair meshes), tooth load is carried by single gear tooth pair. Relieving can be applied to both of tooth tips of driving and driven gear pair or tip of one tooth and root of other tooth. The first one is preferred in this thesis. Properly designed and applied tip relief is important because of its effects on construction of transmission error curves under design or off-design load by depending on application.

Static transmission error curves are constructed for a constant load and constant stiffness with a linear profile relief regardless of no load and loaded situations (Figure 5.4).

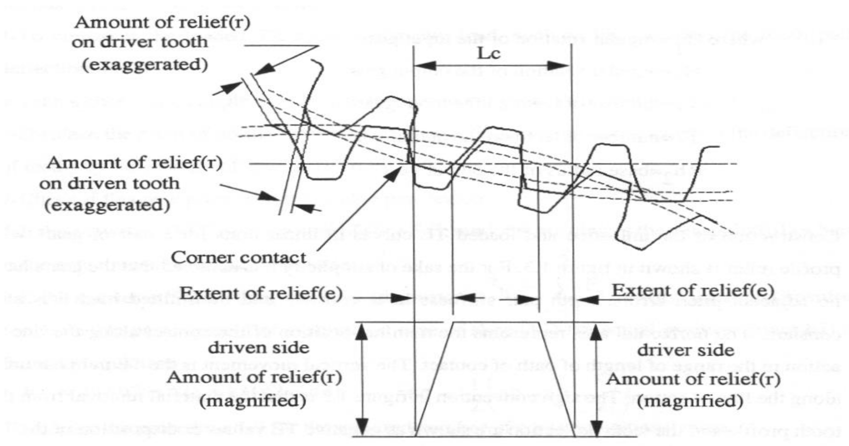


Figure 5.3 Geometry of the tooth contact and profile relief

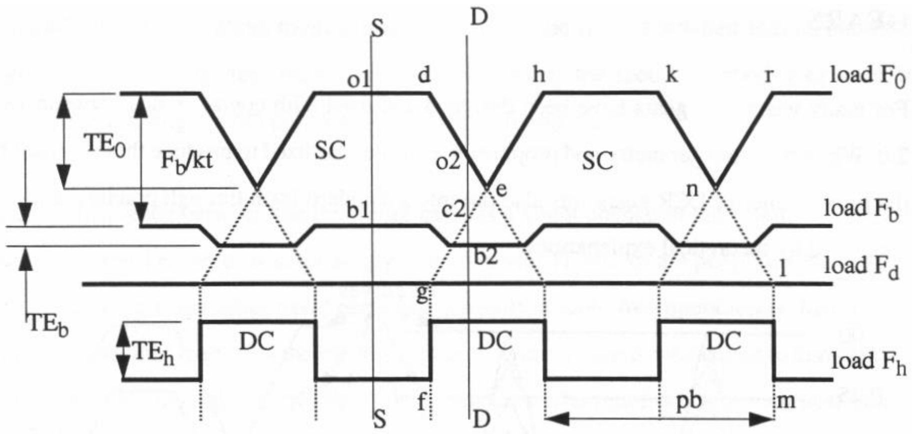


Figure 5.4 Constructions of transmission error curves for spur gear pairs

The procedure for constructing loaded or no load transmission error curves are presented as follows (Figure 5.4):

1) Construct the boundary profiles of the mating tooth pairs such as curves f-g-h-k-l-m.
2) Displace them by the base pitch of the gears along the horizontal axis.
3) Construct the no load TE curve by following the top boundary borders of the each tooth boundary profile, curve d-e-h-k-n-r.
4) To construct the loaded TE curve under any load, such as design load F_b, calculate the tooth pair deflection, $\frac{F_b}{k_t}$.
5) Take a slice, for example S-S, downwards representing the deflection of teeth) by the amount of teeth deflection, $\frac{F_b}{k_t}$.
6) Check if this new point is in the double pair region.
7) If answer is NO as in the case of slice S-S, mark the position of the new point b_1 which is the actual position of the output gear.
8) If answer is YES then use the load constraint $F_b = \sum_{i=1}^{2} F_i$ to find the actual position in the double pair region. For instance, in case of slice D-D the actual position (point b_2) is founded by equations:
   $F_b = k_t(o_2 - b_2) + k_t(c_2 - b_2)$
9) Repeat the procedure 5-8 for the required number of contact positions (slices).
10) Connect the new marked points ($b_1, b_2, etc.$) to construct the loaded TE curve.

Long, short and intermediate relief types are illustrated in Figure 5.5. Their useful and applicable limits are defined in terms of corner contact, loading and TE. The amount of relief is identical for short and long relief whereas extent of relief differs for both.

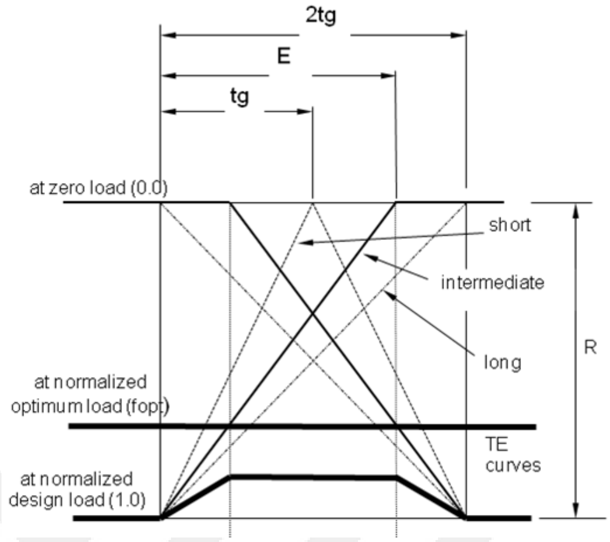


Figure 5.5 Profile relief geometry

5.1 Thin Slice Theory For Gear Tooth Pair Stiffness

When designing micro profile of spur gear pair, design load and tooth pair stiffness are needed in addition to macro geometry parameters. Amount of relief is determined based on design load and tooth pair stiffness. Therefore, evaluation of tooth pair stiffness has a significant influence on micro design of gear pair. Although approaches based on finite element analyses for evaluation of tooth pair stiffness are available in literature, there are no presented results for different tooth profile gears such as symmetric and asymmetric involute tooth profile spur gears.

There are some theoretical and experimental method to evaluate tooth stiffness [49-63]. Theoretical methods bases on analytical calculations [55-62], numerical methods of finite element analyses [51, 57-60, 62] and thin slice theory [50, 53, 54, 61].

Experimental methods and finite element analyses require long time periods and prone to big difference in result owing to any small mistake. Even in the case of fault detection, time consuming occurs for correction.

Analytical calculations are not commonly preferred but still in use. Analytical methods have been developed for definite and constant cross section geometries being not complex [63]. In this method, elastic material or part under load is modeled mathematically by using macro geometry. Then strain generated on material/part is determined based on mechanics and elastic deformation is determined according to Hooke's Law by assuming loading in elastic region. But generated analytical equations cannot be used for gears because of variation in their geometries from root to tip. Numerical calculation methods based on finite element analyses are more suitable to analyze these kinds of complex and varying cross section geometries which cannot be analyzed by analytical methods.

Finite element methods which have increasing usage in academic and other researches is one of the mostly used method for tooth root bending stress and tooth pair stiffness [51, 57-60, 62]. But these methods require commercial software and long analysis time duration by depending on element type, mesh size, number of element, load and boundary conditions. Different results can be obtained from these analyses owing to variation of input parameters.

There is another method, thin slice theory which can be used for stiffness evaluations. This theory is based on theoretical and analytical calculations. Anyone who intends to determine gear tooth pair stiffness can generate his own codes based on numerical method of this slice theory.

In applying thin slice method, part to be analyzed is divided into thin slices with constant cross sections instead of dividing the part into too many, very small pieces (as in the case of finite element analyses). Approximate total deformations are calculated for each slice easily by using analytical equations. There some studies [50, 53-54, 61] which uses thin slice method for gears. Elastic deflection of each slice is firstly calculated and then total deflection is determined by assuming gear as combination of thin slices.

In this study, thin slice method is preferred because of its mathematical ease of use with simple applicability to software of micro geometry design and static transmission error analysis in addition to obtaining results quickly compared to experimental and finite element studies.

A tool of tooth pair stiffness calculation has been developed by using positive contributions of studies based on thin slice theory and integrating these all to transmission error analysis software. Tooth pair stiffness calculations are developed and stiffness curves are constructed under different loads for gear pairs with different values of module, teeth number, root fillet radius, tip and root diameters, pressure angle, etc. by using this tool. Accurate calculations can be carried out to minimize transmission error by using these stiffness values and curves obtained by using this tool for micro geometry design.

In literature, gear tooth under load is considered as a cantilever beam with one end fixed and elastic deformation and stiffness calculations are carried out based on this fixed end. Tooth load is normal to the tooth flank so vertical component yields bending down and shear whereas horizontal component causes bending up and compression of tooth. In addition, there are two another elastic deformations; tooth root rotation under the effect of bending and elastic contact deformation on tooth flank. Loading point, bending, shear and tooth root rotations deformations are shown in Figure 5.6.

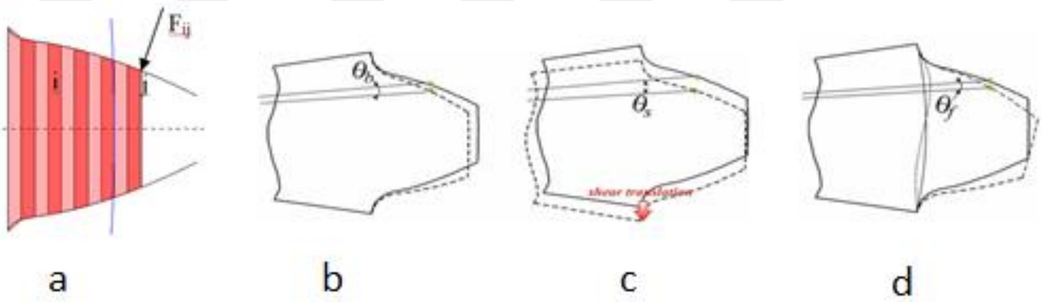


Figure 5.6 Loading point (a), bending (b), shear (c) and tooth root rotation (d) deformations [55]

Cantilever beams have uniform cross sections and constant height and analytical equations are developed for cantilever beams based on these geometrical features. Involute or any other gear tooth profile has variable cross sections and height. There is no analytical equation for tooth stiffness calculations and some finite element analyses are conducted for this aim. Thin slice method is also a numerical method (similar to finite element), based on analytical calculations and do not require a specific commercial software. Because of all these reasons abovementioned, thin slice method has a widespread use area in mechanics.

In this theory, a gear tooth is assumed as too many thin slices bonded to each other like cantilever beams with a width of dx ($\int y * dx$) and height of y. Each slice is accepted it is bonded to next and previous slices as shown in Figure 5.7.

By using analytical equations, elastic and bending deflections are calculated for each slice with uniform cross section and height. Then, total deflection is calculated by evaluating these elastic and bending deflections of each slice. A tooth which is divided into thin slices and loaded by Fij is given in Figure 5.8. Loading and geometry of slice at point of i is also presented in same figure. Deflections of each thin slice having constant height under different loads can be calculated by using analytical equations. Deflection owing to contact pressure at meshing point for one tooth regardless of slices is determined by using Eq. 5.3 and 5.4 [54].

Part of tooth inside of gear body has fractional effect on total tooth deflection in addition to the effect of teeth integral part to gear body (at outside of gear body). This deflection is called tooth foundation effect and rotation of root section at inside of gear body is modeled [54]. There are some similar equation are presented in different literature works [50, 54]. Tooth foundation effects are included in models developed in this study similar to Ref [50, 53].

Equation for elastic deflection $(q_{fe})_j$ of wide tooth including tooth root deflection is presented in Eq. 5.5 and tooth geometry is shown in Figure 5.9.

$$\delta_{Hj} = \frac{1{,}37}{2 * E_{12e}^{0{,}9} * b_e * F_{ij}^{0{,}1}} \tag{5.3}$$

$$E_{12e} = 2 * \frac{E_{1e} * E_{2e}}{(E_{1e} + E_{2e})} \tag{5.4}$$

$$(q_{fe})_j = \frac{W_j \cos^2 B_j}{E_e F}(1 - v^2)\left\{\begin{matrix} \frac{16.67}{\pi}\left[\frac{l_f}{h_f}\right]^2 \\ +2\left(\frac{1 - v - 2v^2}{1 - v^2}\right)\left[\frac{l_f}{h_f}\right] \\ +1.534\left(1 + \frac{\tan^2 \beta_j}{2.4(1 + v)}\right) \end{matrix}\right\} \tag{5.5}$$

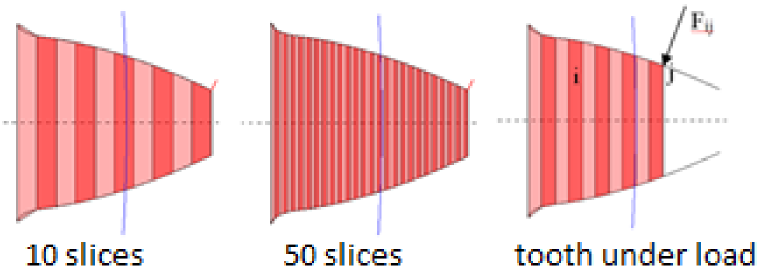


Figure 5.7 Thin sliced teeth and tooth under load

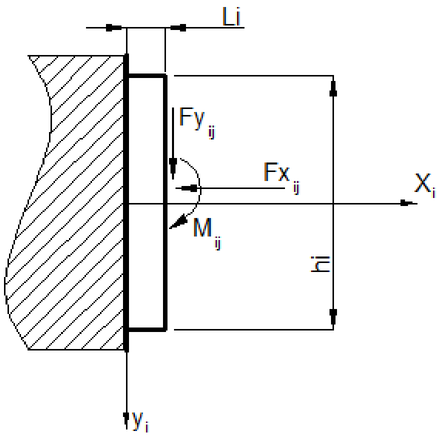


Figure 5.8 Loading at a slice of tooth [54]

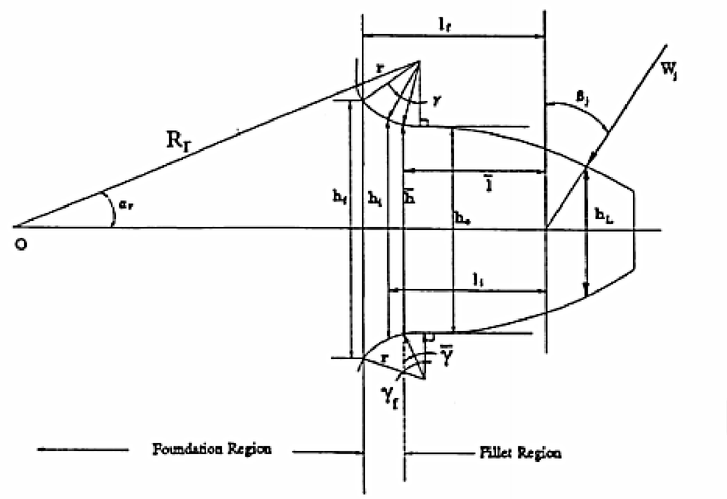


Figure 5.9 Tooth geometry used for deflection $(q_{fe})_j$ owing to tooth foundation effect [50]

Total deflection (δ_j) is calculated based on Eq.5 by adding contact deflection regardless of slices and elastic deflection of tooth root to deflections of all slices at point j and converting it to component along load [54].

$$\delta_j = \delta_{tj} + \delta_{Hj} + q_{fej} \tag{5.6}$$

By using equations given above, a software is developed for calculations of tooth deflection and tooth pair stiffness.

5.2 Software For Transmission Error Calculation

A user friendly software has been developed for micro profile design/analysis of symmetric and asymmetric tooth profile spur gears. Enterance screen of software is illustrated in Figure 5.10. This software includes four main sections as shown in Figure

5.11. These are four main sections; input, micro profile design and design loads, output values, and graphical screens.

All inputs such as module, teeth number, pressure angle for drive and coast side flanks, coefficients of addendum, dedendum, profile shift and hob tip radius, face width, load, modulus of elasticity, Poisson's ratio and angular velocity can be entered by input section of software (Figure 5.11).

The section of software for entering parameters of micro geometry design and loads to software is presented in Figure 5.12. Optimum relief can be applied by user in this section for short, long and intermediate relief types or relief parameters (amount of relief and extent of relief) can be entered by user.

Peak to peak transmission error value under the selected load is calculated and presented as in Figure 5.13.

Constructed curves are transmission error (TE), tooth pair stiffness, mesh stiffness, contact stress, sliding velocity, PxV (contact stress x sliding velocity) and load sharing. These curves are constructed under design load and off-design loads (as different percentage of design load) as shown in Figure 5.14. Different colors can be selected for each curve under a specific load to prevent confusion.

Tooth pair stiffness calculation procedure is adapted to software. This software enables tooth stiffness calculations for symmetric and asymmetric tooth spur gears. Stiffness curves are obtained by two different ways; minimum and maximum stiffness values can be entered by user (Figure 5.15) or stiffness values are determined based on thin slice theory (Figure 5.16).

Tooth pair stiffness and mesh stiffness curves are presented in enlarged scale in Figure 5.17 and 5.18 respectively.
Selection of different load values, curves of transmission error, load diagrams, contact stress, sliding velocity and PxV are presented in Figure 5.19, 5.20, 5.21, 5.22, 5.23 and 5.24 respectively.

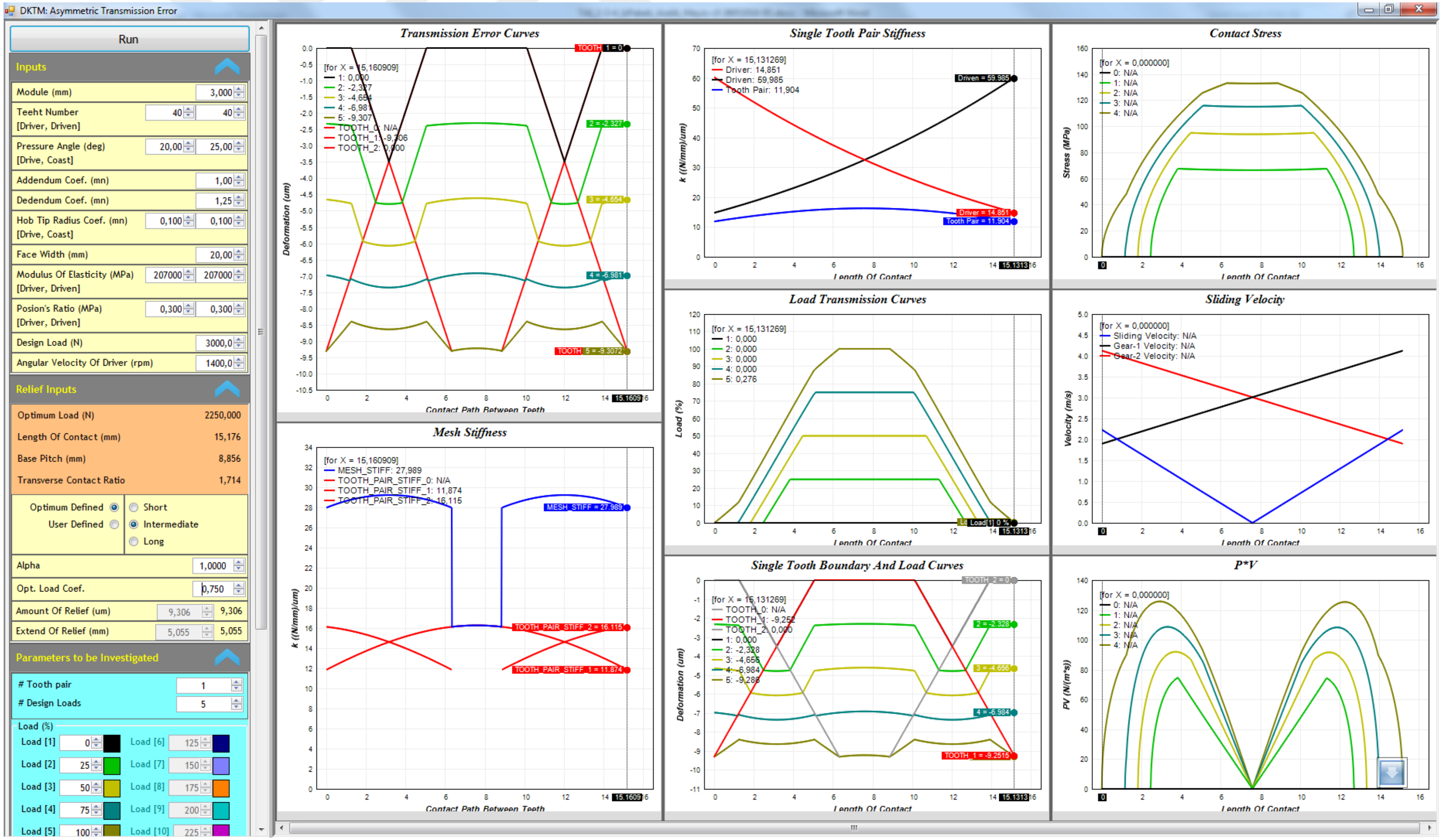


Figure 5.10 Main page of in-house software

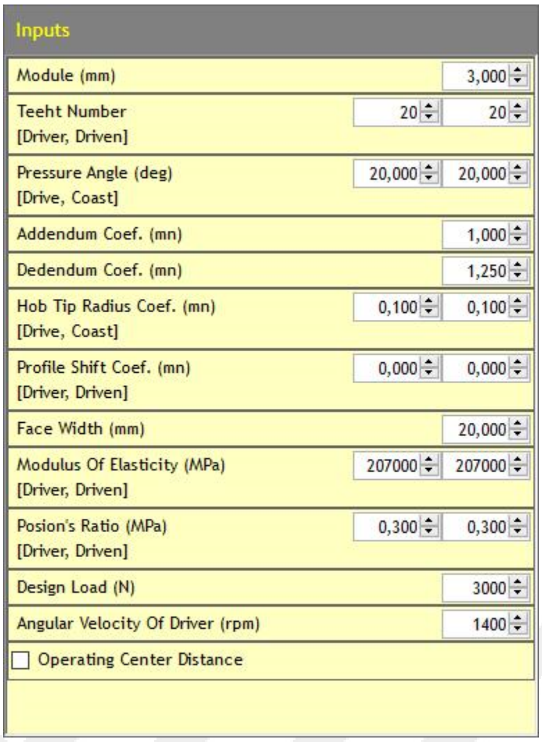


Figure 5.11 Input parameters

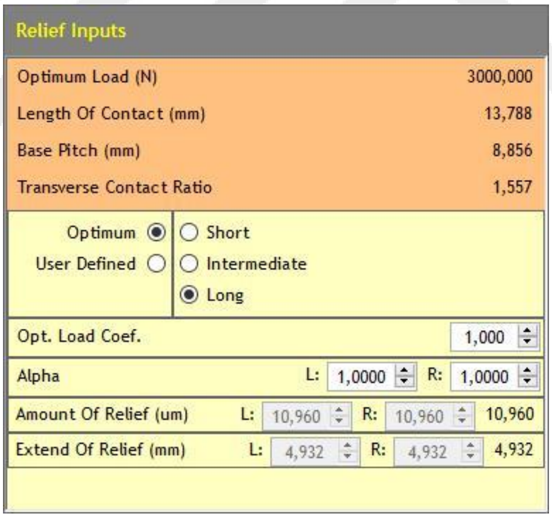


Figure 5.12 Software interface for micro profile design parameters

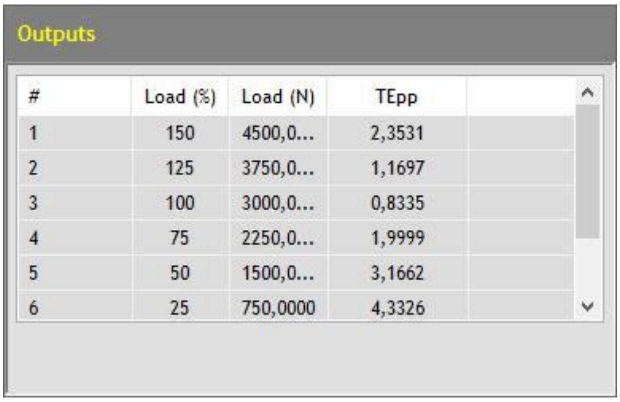


Figure 5.13 Output Values of PPTE

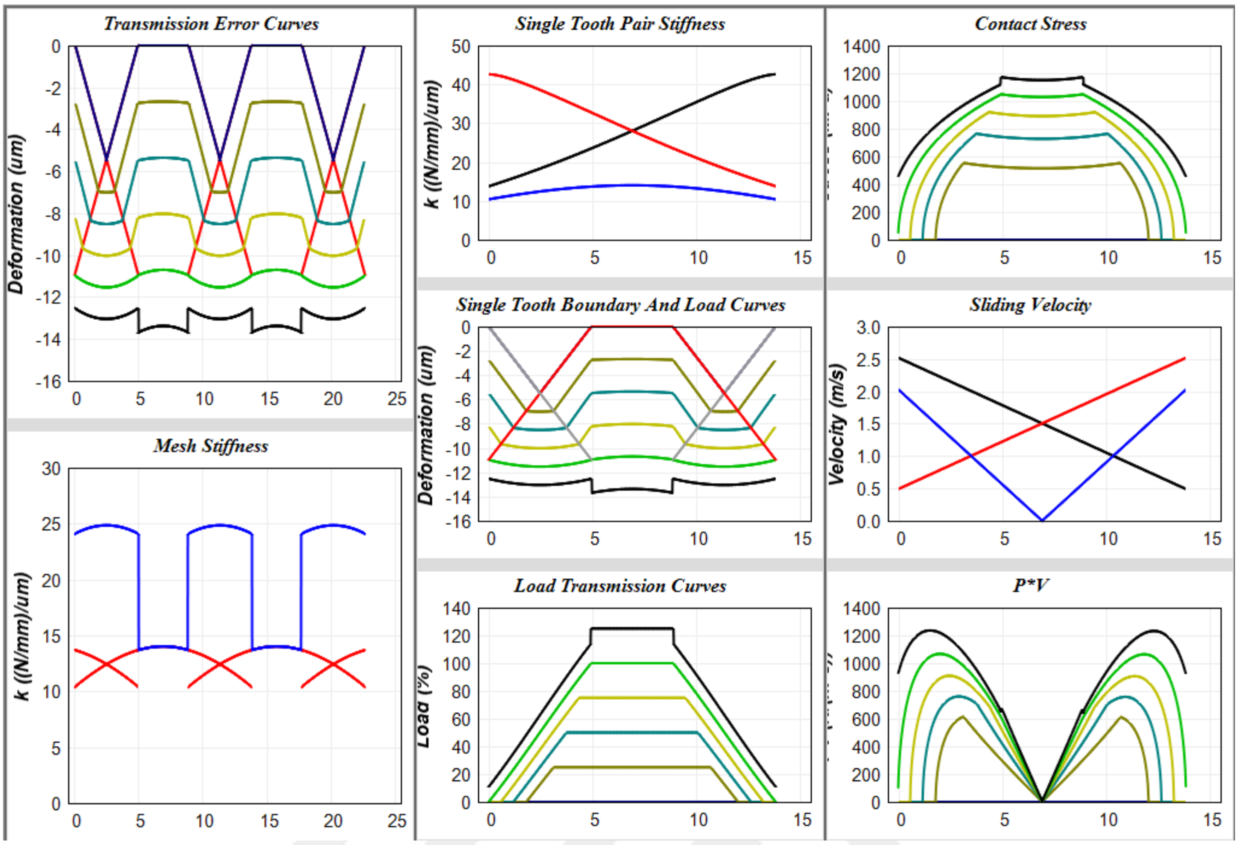


Figure 5.14 Curves of TE, stiffness, contact stress, sliding velocity and load

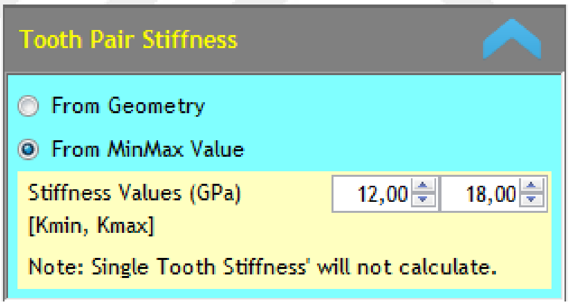


Figure 5.15 Entering maximum and minimum stiffness values by user

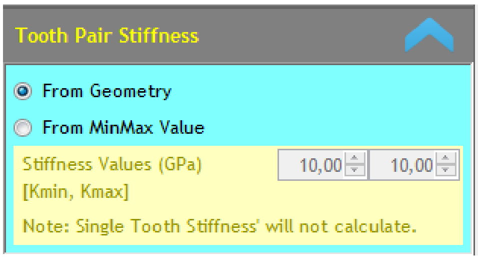


Figure 5.16 Determination of maximum and minimum stiffness values based on thin slice theory

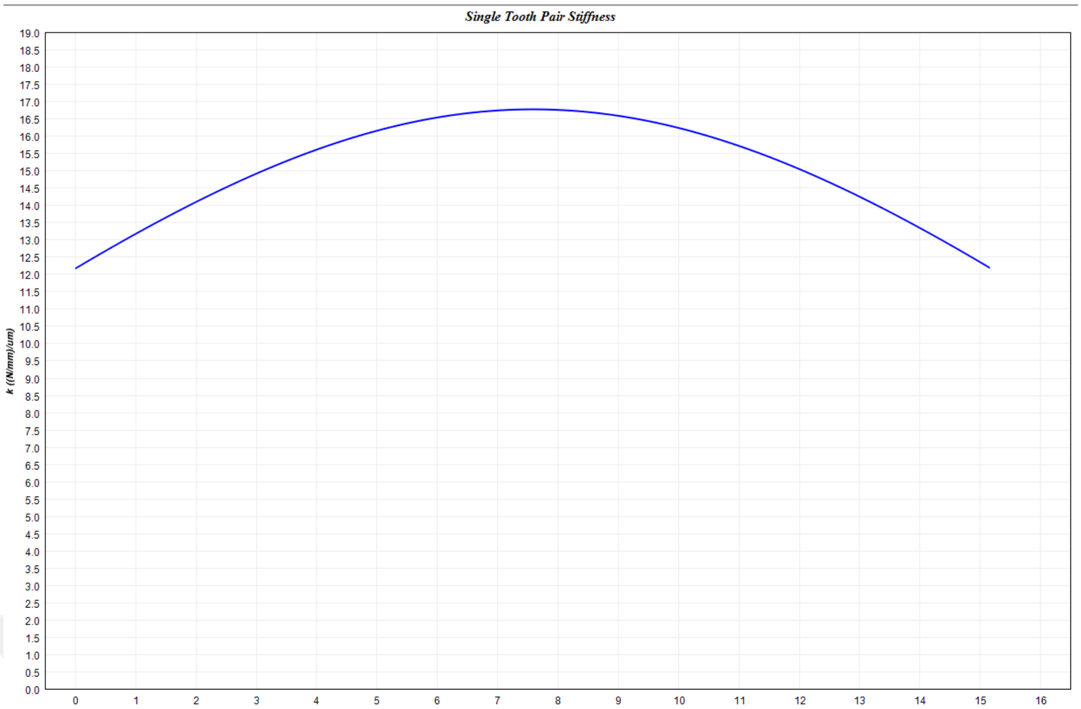


Figure 5.17 Tooth pair stiffness curve

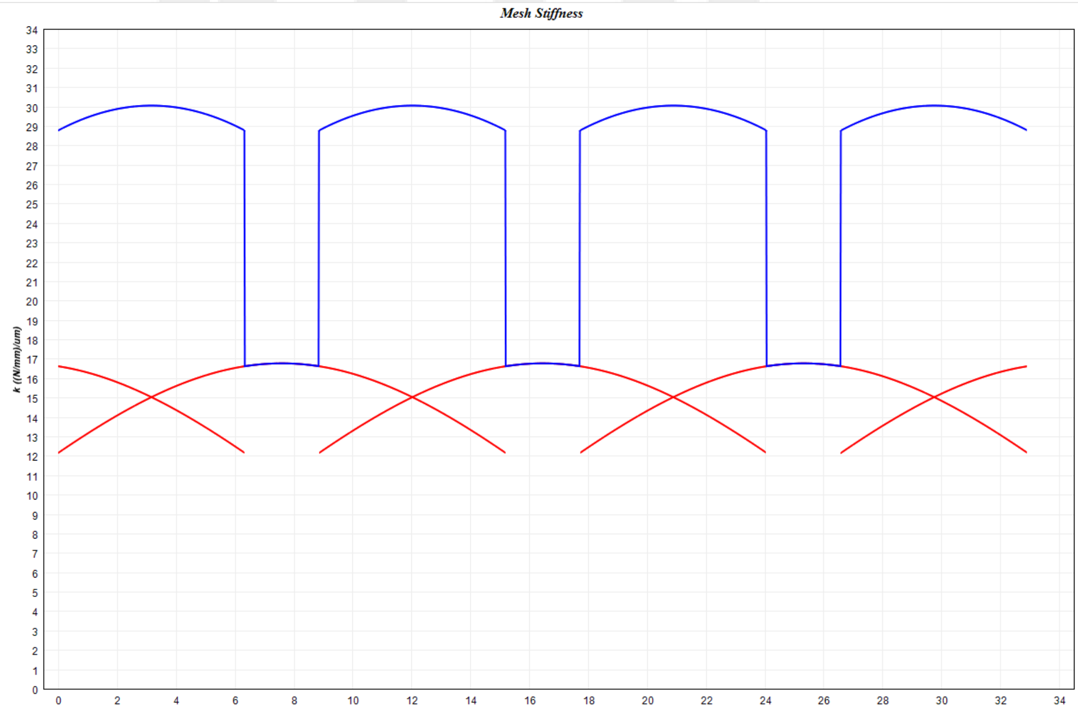


Figure 5.18 Mesh stiffness curve

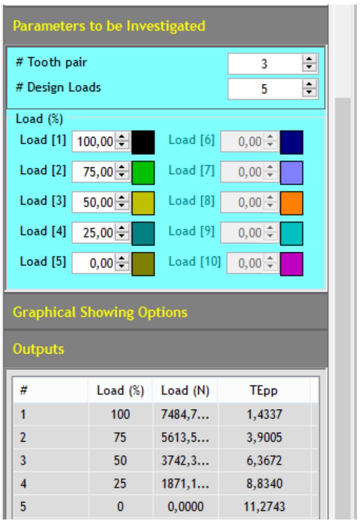


Figure 5.19 Software interface for design load percentage and PPTE values for these loads

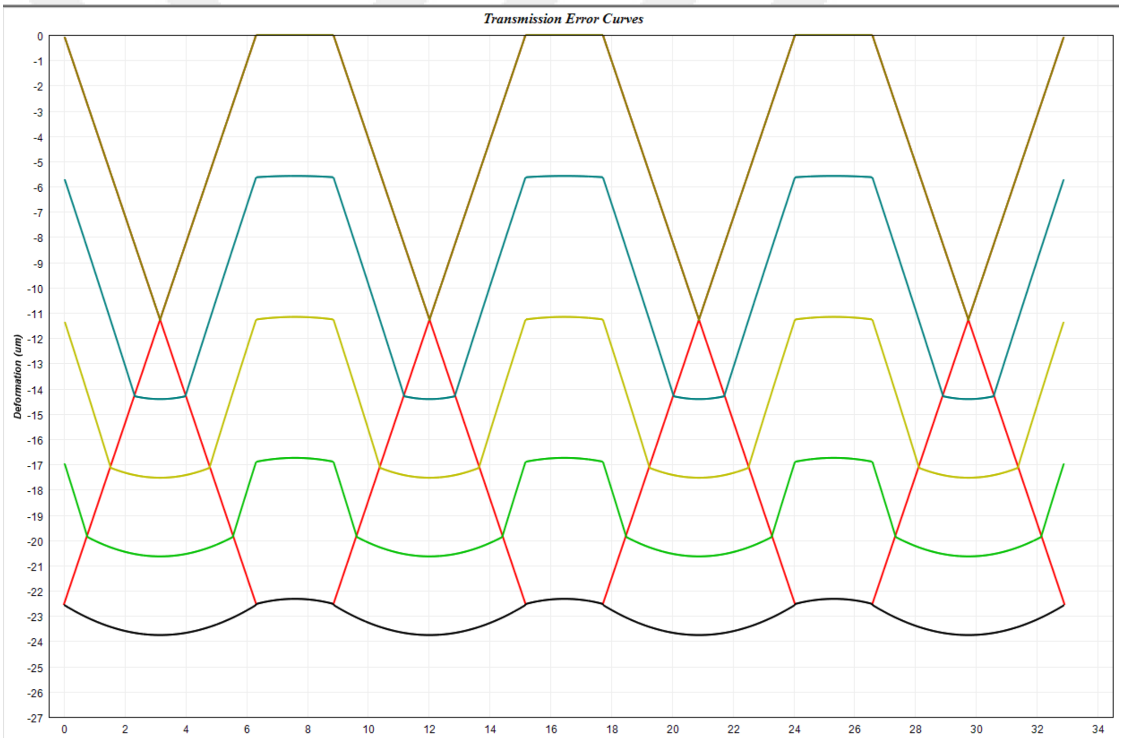


Figure 5.20 Transmission error curves for different load values (in-house software)

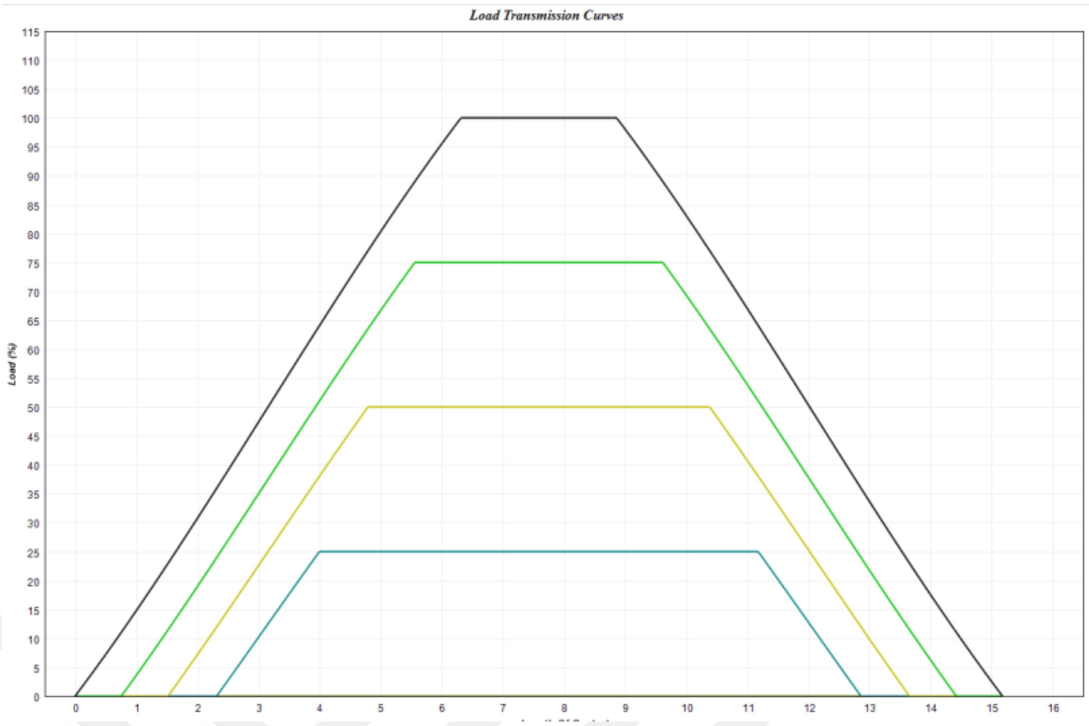


Figure 5.21 Load diagrams (in-house software)

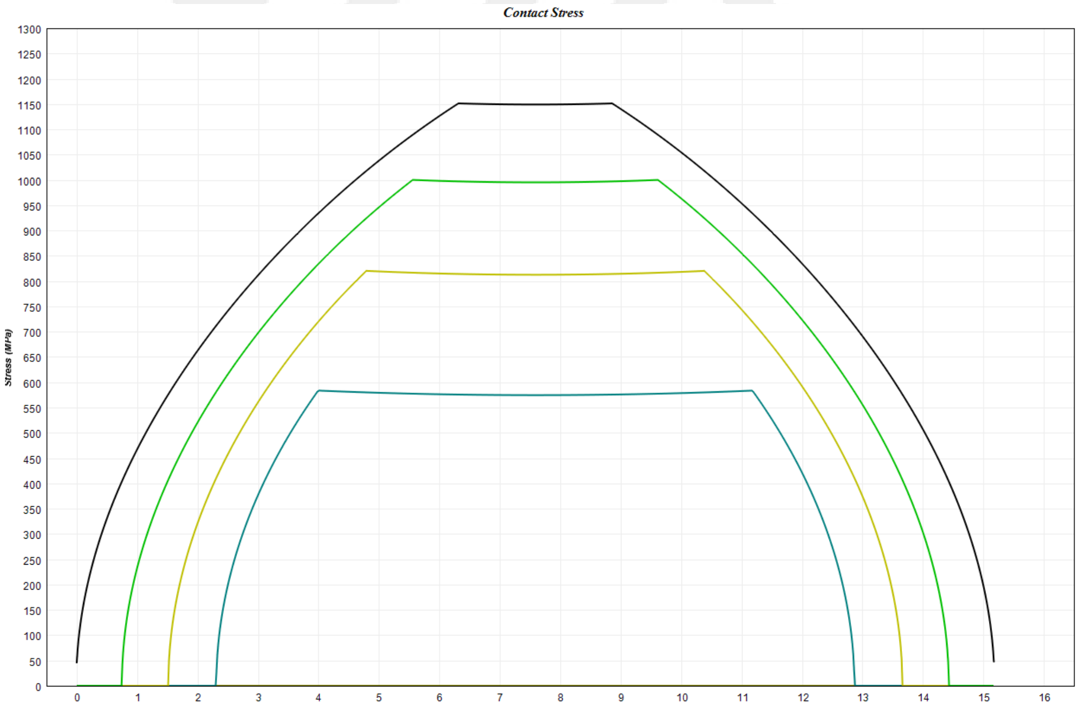


Figure 5.22 Contact stress (in-house software)

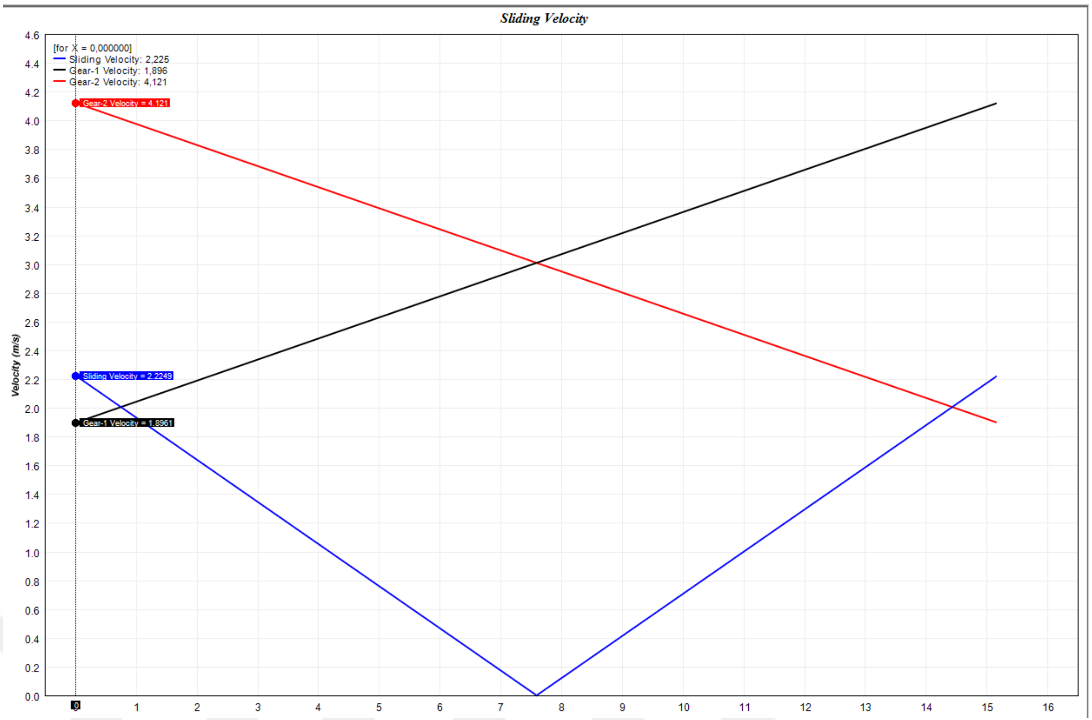


Figure 5.23 Sliding velocity curves (in-house software)

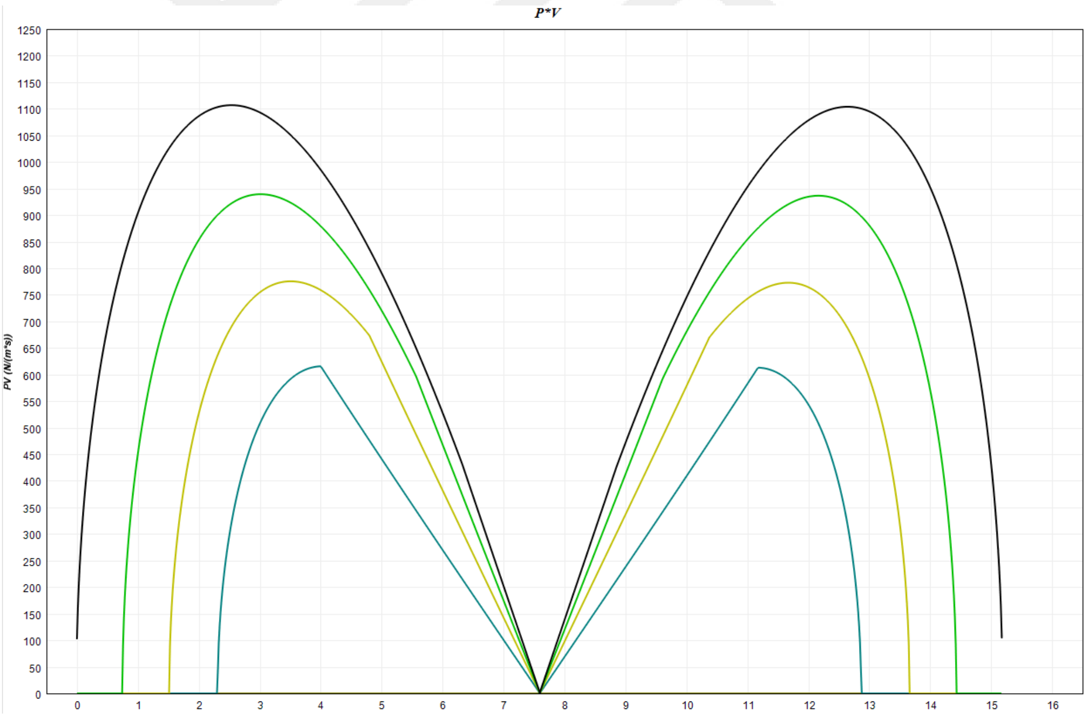


Figure 5.24 PxV curves (in-house software)

5.3 Case Studies

Some case studies for stiffness calculations and micro profile design of symmetric and asymmetric spur gears are conducted and presented in this section. Case studies were conducted based on gear pair macro and micro parameters given by literature works. Values of tooth pair stiffness and mesh stiffness were determined with constructing curves belong these values by using software for some cases. In other case studies, peak to peak transmission error values were evaluated and transmission error curves for different loads were constructed.

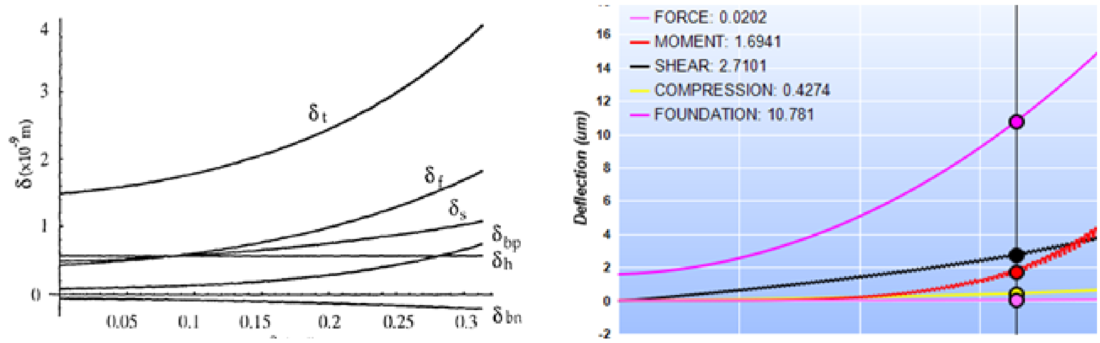
5.3.1 Case Studies Of Stiffness

Tooth pair stiffness, mesh stiffness and transmission error values for symmetric and asymmetric tooth profile external spur gears were calculated by developed software. Stiffness and transmission error calculations are based on thin slice theory and Harris map respectively. After calculations, curves for each parameter was constructed and presented for design load and off-design loads.

5.3.1.1 Case S1 (Stiffness calculation)

Symmetric tooth spur gear pair with module of 6mm, teeth number of 23 and gear ratio of unity is studied for tooth deflection under load. Tooth deflection with different components such as force, moment, shear, axial and tooth root foundation was determined by loaded at different loading points on tooth flank surface along length of contact (Figure 5.25).

Deflections of force, moment, shear, compression and tooth foundation were calculated by using software. Constructed curves are presented for reference study [49] and software in Figure 5.26. Their behaviors under load and relative magnitudes are compared and it is seen that they are in compliance.



a) b)

Figure 5.25 Tooth loading at different locations along path of contact

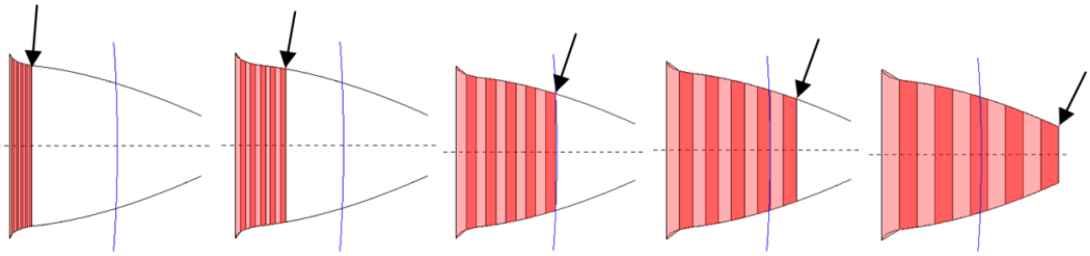


Figure 5.26 Tooth deflection components

5.3.1.2 Case S2 (Stiffness calculation)

Gear pair analyzed in reference study [57] was studied here for stiffness calculations. Finite element analyses were conducted in reference paper whereas calculations were performed by using software based on thin slice theory. A symmetric spur gear pair with 2mm of module, 20degree of pressure angle, 20 of teeth number, gear ratio of unity was studied for 3325N. Curve of 1/keq= (E*δ) / (F/L) was presented without unit in reference paper where keq, E, δ, F and L are equivalent tooth stiffness, elastic modulus, tooth deflection, force and width respectively. This value (1/keq) is converted into tooth pair stiffness (N/mm/micron=GPa) by taking reciprocal of it and multiplying with modulus of elasticity (E=207GPa).

Results from different sources are presented in same curve. Midpoint on horizontal axes refers to pitch point of gear and edge points refer to start and end of contact. These values of 17 and 23 are evaluated for pitch point edge points respectively. In this case, stiffness for pitch point and edge points (start and end of contact) can be calculated as follows ((1/17.0)*207=) 12.17N/mm/micron and ((1/23.0)*207=) 9.00N/mm/micron respectively. Tooth pair stiffness curve constructed by software is presented in Figure 5.27. Values for pitch point and edge points can be read from this figure as 11.70N/mm/micron and 8.25N/mm/micron respectively. These values satisfy with results of reference study [57] with 4-8% difference.

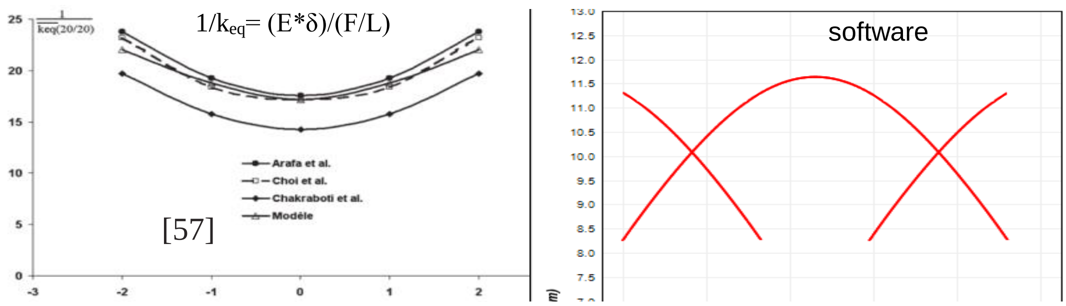


Figure 5.27 Tooth pair stiffness (software) and $1/k_{eq}$ curves along length of contact

5.3.1.3 Case S3 (Stiffness calculation)

A gear pair with normal module of 4mm, pressure angle of 20degree, face width of 20mm and teeth number of 21 and 49 for pinion and gear respectively was analyzed in this section and compared with results of Ref [60]. Tooth pair stiffness curves for both studies are presented in Figure 5.28. The stiffness curve obtained by reference study is given in N/mm whereas N/mm/micron is preferred in this thesis. So it is converted through dividing by face width for comparison. Peak and lowest value of tooth pair stiffness of curve [60] are evaluated by 13.25 N/mm/micron and 9.8 N/mm/micron respectively via dividing original values by 20mm of face width. Corresponding values can be read as 12.75 N/mm/micron and 8.5 N/mm/micron from curve constructed by software (Figure 5.28). Values are in accordance with each other (with 4-13% difference).

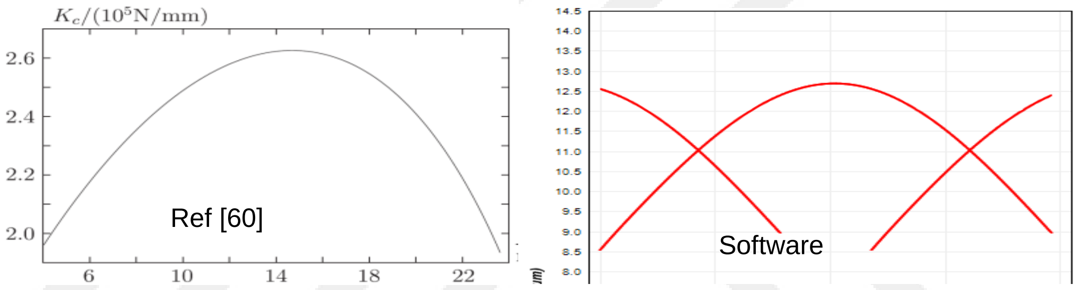


Figure 5.28 Tooth pair stiffness curves along length of contact for Ref [60] and software

5.3.1.4 Case S4 (Stiffness calculation)

Spur gear pair with normal module of 3mm, 20degree of pressure angle, 60mm of face width, teeth numbers of 65, addendum coefficient of unity and dedendum coefficient of 1.25 was studied for stiffness evaluations. Tooth pair stiffness curve including single contact and double contact regions for reference paper [62] is presented in Figure 5.29. Stiffness for single tooth contact region is 14.0 N/mm/micron for Ref [62] whereas corresponding value is 15.6 N/mm/micron for software (Figure 5.29). Value obtained by software is relatively higher than that of reference paper. Peak value of tooth pair stiffness referring to double contact region can be read as 24.5 N/mm/micron from paper [62]. When this value is divided by 2, 12.25 N/mm/micron is obtained for single tooth. Corresponding value is 13.5 N/mm/micron and higher than reference value with a difference of 10-12%.

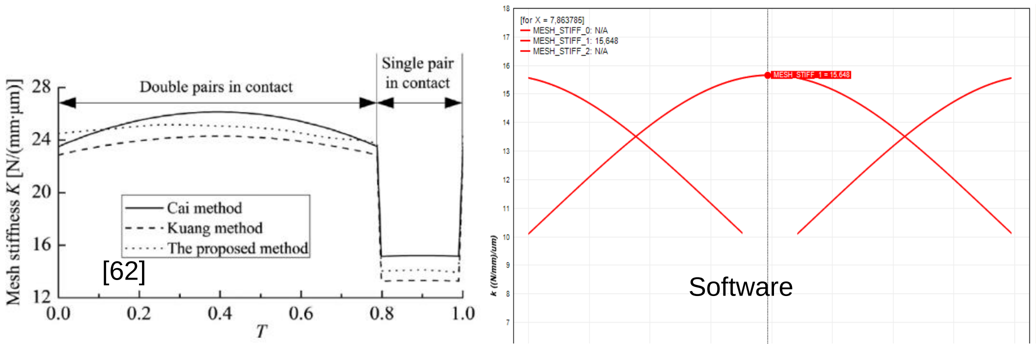


Figure 5.29 Tooth pair stiffness curves along length of contact for Ref [62] and software

5.3.1.5 Case S5 (Stiffness calculation)

A symmetric spur gear pair whose parameters are given in Table 5.1 [1] is analyzed for stiffness calculations. Parameters are entered into software as shown in Figure 5.30. Results are presented in Table 5.2. Mesh stiffness curves are presented in Figure 5.31 for both studies.

Software calculated values of maximum and minimum stiffness higher than reference study [1] for this symmetric spur gear pair case.

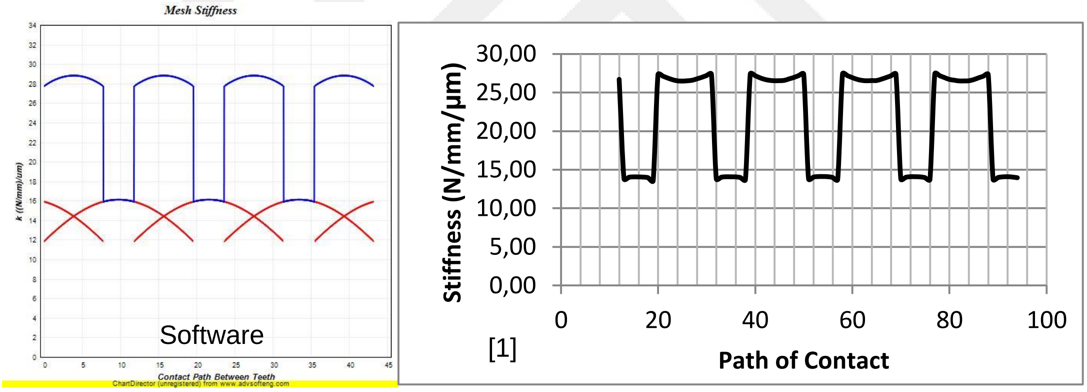
Table 5.1 General gear parameters of Case S5

Parameter	Value	Unit
Module	4	mm
Teeth number	30/30	-
Pressure angle	20	Degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.3	-
Face width	15	mm
Design load	7500	N

Table 5.2 Mesh stiffness values obtained by Ref [1] and software (Case S5)

Mesh stiffness values	FEA [1]	Software	Unit
Minimum	13.58	16.20	N/mm/micron
Maximum	27.43	28.90	N/mm/micron

Figure 5.30 Gear parameters input screen to software (Case S5)



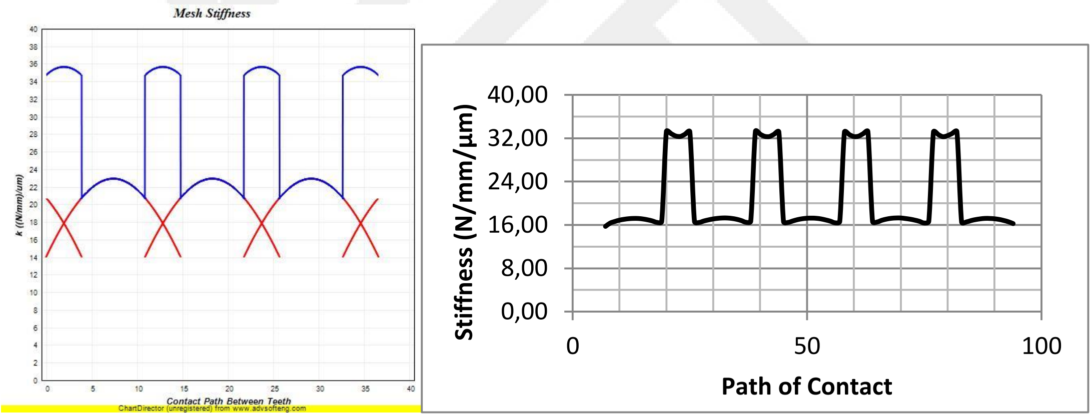
(a) (b)

Figure 5.31 Mesh stiffness curves for Ref [1] and software (Case S5)

5.3.1.6 Case A1 (Stiffness calculation)

An asymmetric spur gear pair whose general gear parameters are same with Case S5 (Table 5.1) was studied in terms of mesh stiffness. Pressure angle of drive side is 30degree whereas 20degree is used for coast side flank. General gear parameters were entered in software and stiffness calculation method was selected (Figure 5.32). Mesh stiffness values are given in Table 5.3 and curves are illustrated for Ref [1] and software in Figure 5. 33.

Figure 5.32 Gear parameters input screen to software (Case A1)



(a) (b)

Figure 5.33 Mesh stiffness curves for Ref [1] and software (Case A1)

Table 5.3 Stiffness values obtained by Ref [1] and software (Case A1)

Stiffness Values	FEA [1]	Software	Unit
Minimum	16.25	21.00	N/mm/micron
Maximum	33.26	35.50	N/mm/micron

5.3.1.7 Case A2 (Stiffness calculation)

Asymmetric tooth spur gear pair whose parameters are given in Table 5.4 was analyzed for stiffness. General gear parameters were entered in software as in Figure 5.34. Tooth pair stiffness and mesh stiffness curves are given in blue and red colors respectively in

Figure 5.35 for software. Only tooth pair stiffness [1] is presented in same figure for reference study. Tooth pair stiffness values are presented in Table 5.5.

Table 5.4 General gear parameters of Case A2

Parameter	Value	Unit
Module	1	mm
Teeth number	20/20	-
Pressure angle (drive/coast)	20/25	Degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.398	-
Face width	1	mm
Design load	10	N

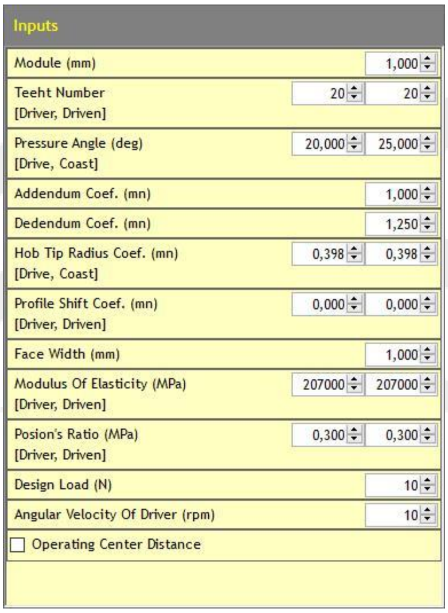


Figure 5.34 Gear parameters input screen to software (Case A2)

Table 5.5 Stiffness values obtained by Ref [1] and software (Case A2)

Stiffness Values	FEA [1]	Software	Unit
Minimum	9.58	10.62	N/mm/micron
Maximum	15.15	13.74	N/mm/micron

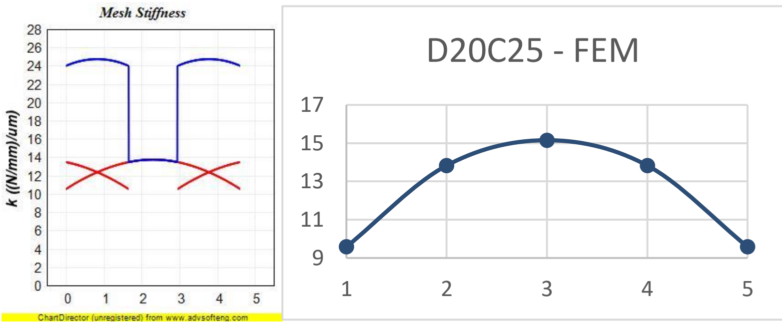


Figure 5.35 Tooth pair stiffness curves for Ref [1] and software (Case A2)

5.3.1.8 Case A3 (Stiffness calculation)

An asymmetric spur gear pair with greater pressure angle of drive side was studied in this case for stiffness evaluation. General gear parameters are presented in Table 5.6. Tooth pair stiffness (blue color) and mesh stiffness (red color) are presented for software whereas tooth pair stiffness curve is illustrated for Ref [1] in Figure 5.36. Tooth pair stiffness values for both studies are given in Table 5.7.

Table 5.6 General gear parameters of Case A3

Parameter	Value	Unit
Module	1	mm
Teeth number	20/20	-
Pressure angle (drive/coast)	20/30	Degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.308	-
Face width	1	mm
Design load	10	N

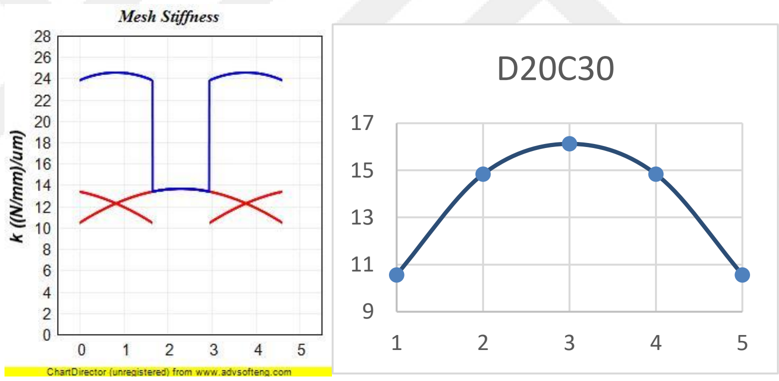


Figure 5.36 Tooth pair stiffness curves for Ref [1] and software (Case A3)

Table 5.7 Stiffness values obtained by Ref [1] and software (Case A3)

Stiffness Values	FEA [1]	Software	Unit
Minimum	10.56	10.47	N/mm/micron
Maximum	16.13	13.65	N/mm/micron

5.3.1.9 Case A4 (Stiffness calculation)

Asymmetric tooth spur gear pair with same general parameters presented in Case A3 except pressure angle was analyzed for tooth pair stiffness values. Pressure angles of 30degree and 20degree were preferred for drive and coast side flanks of gear tooth profile. Stiffness values and curves are presented in Table 5.8 and Figure 5.37 respectively.

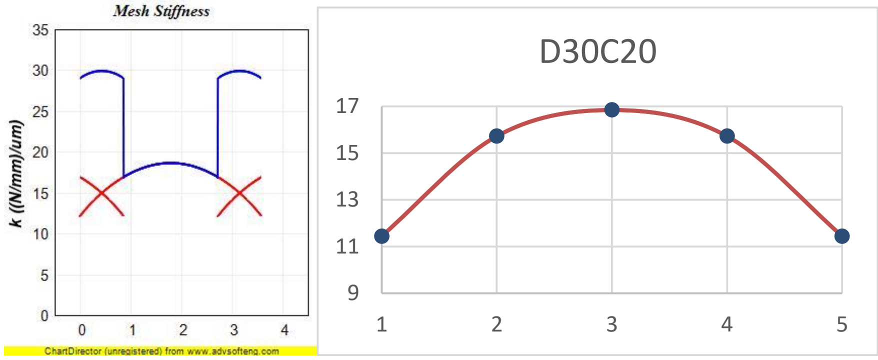


Figure 5.37 Tooth pair stiffness curves for Ref [1] and software (Case A4)

Table 5.8 Stiffness values obtained by Ref [1] and software (Case A4)

Stiffness Values	FEA [1]	Software	Unit
Minimum	11.44	12.19	N/mm/micron
Maximum	16.84	18.62	N/mm/micron

5.3.1.10 Case A5 (Stiffness calculation)

An asymmetric spur gear pair with gear parameters given in Table 5.9 was studied in this case. Stiffness curves and values were obtained and presented in Figure 5.38 and Table 5.10 respectively.

Table 5.9 General gear parameters of Case A5

Parameter	Value	Unit
Module	1	mm
Teeth number	20/20	-
Pressure angle (drive/coast)	25/20	Degree
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Cutter tip radius coefficient	0.398	-
Face width	1	mm
Design load	10	N

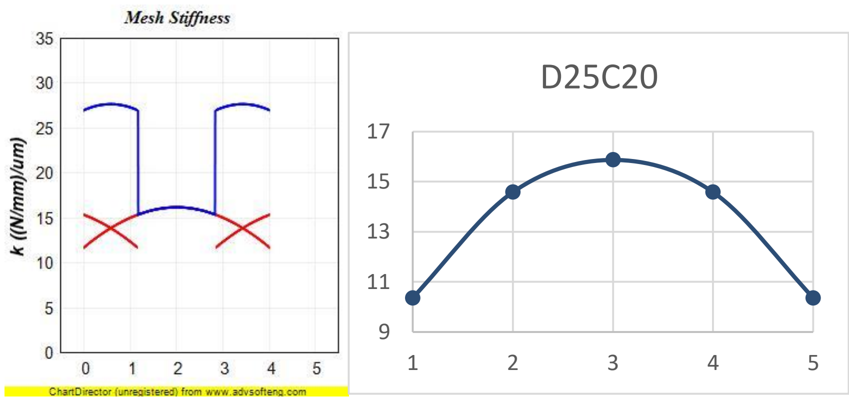


Figure 5.38 Tooth pair stiffness curves for Ref [1] and software (Case A5)

Table 5.10 Stiffness values obtained by Ref [1] and software (Case A5)

Stiffness Values	FEA [1]	Software	Unit
Minimum	10.35	11.65	N/mm/micron
Maximum	15.87	16.12	N/mm/micron

In-house software developed in this thesis gives generally higher or lightly lower values when results were compared to reference studies. Especially when software results were compared with those of finite element analyses, it is obvious to say that software yields higher stiffness values.

5.3.2 Case Studies Of Transmission Error

5.3.2.1 Case S6 (Transmission error calculation)

Results of reference study [33] and analytical prediction based on Harris map (Software) were compared. Gear parameters of reference study are presented in Table 5.11. Peak to peak transmission error (PPTE) value was presented as 1.00 µm in reference paper whereas 1.05 µm has been calculated by software. Transmission error (TE) curves for different loads and PPTE values are given in Figure 5.39 and 5.40 for current and reference study respectively. PPTE values in Ref [33] (measured and predicted values) and predicted by software (based on Harris map) are close to each other(with a difference of 0.05 µm).

Table 5.11 General gear parameters of Case S6

Parameter	Value	Unit
Pressure Angle (drive/coast)	20/20	Degree
Normal module	6.35	mm
Pinion tooth number	32	-
Gear tooth number	32	-
Design load	300	N/mm
Face width	20.0	mm
Relief type	Long	-

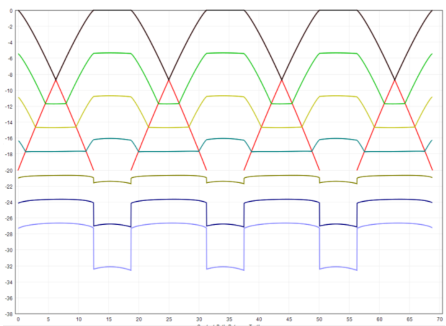


Figure 5.39 TE curves and PPTE values for Case S6 (software)

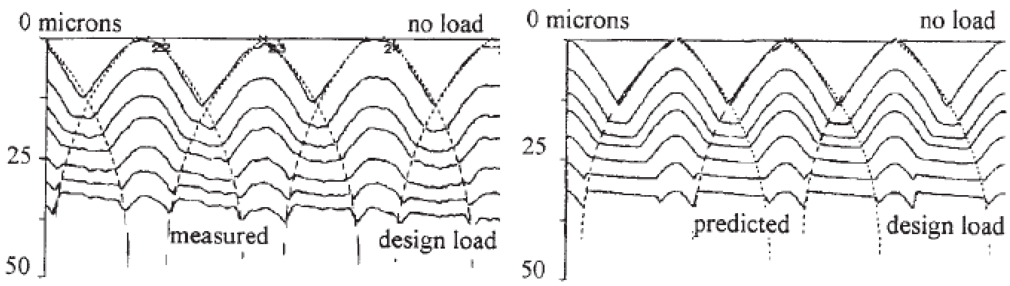


Figure 5.40 Measured and predicted TE Curves of Case S6 [33]

5.3.2.2 Case S7 and S8 (Transmission Error Calculation)

Gear pairs with long relief were studied for quasi static transmission error analysis. Gear pair parameters are presented in Table 5.12 and 5.13. Peak to peak transmission error values are given in Table 5.14-5.15 while static transmission error curves under different loads are illustrated in Figures of 5.41-5.44. It can be concluded that the difference of PPTE values evaluated by software and Ref [1] are very small for Case S7 and S8.

Table 5.12 General gear parameters of Case S7 and S8

Parameter	Value	Unit
Normal module	3	mm
Teeth number of pinion	40	-
Gear ratio	1	-
Design torque	160.43	Nm
Face width	20	mm

Table 5.13 Pressure angle and relief parameters of Case S7 and S8

Parameter	Value		Unit
	S7	S8	
Pressure angle (drive/coast)	20/20	25/25	degree
Amount of relief	9.128	7.917	µm

Table 5.14 PPTE values of Case S7 (D20C20)

PPTE (µm)		
Load (%)	In-house software	FEA
25	3.588	3.200
50	2.586	2.000
100	0.584	0.700

Table 5.15 PPTE values of Case S8 (D25C25)

PPTE (µm)		
Load (%)	**In-house software**	**FEA**
25	3.197	3.500
50	2.419	2.500
100	0.864	0.900

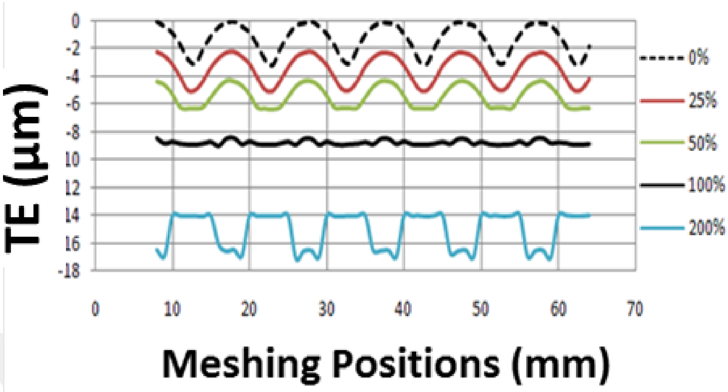


Figure 5.41 TE curves of Case S7 (D20C20) [1]

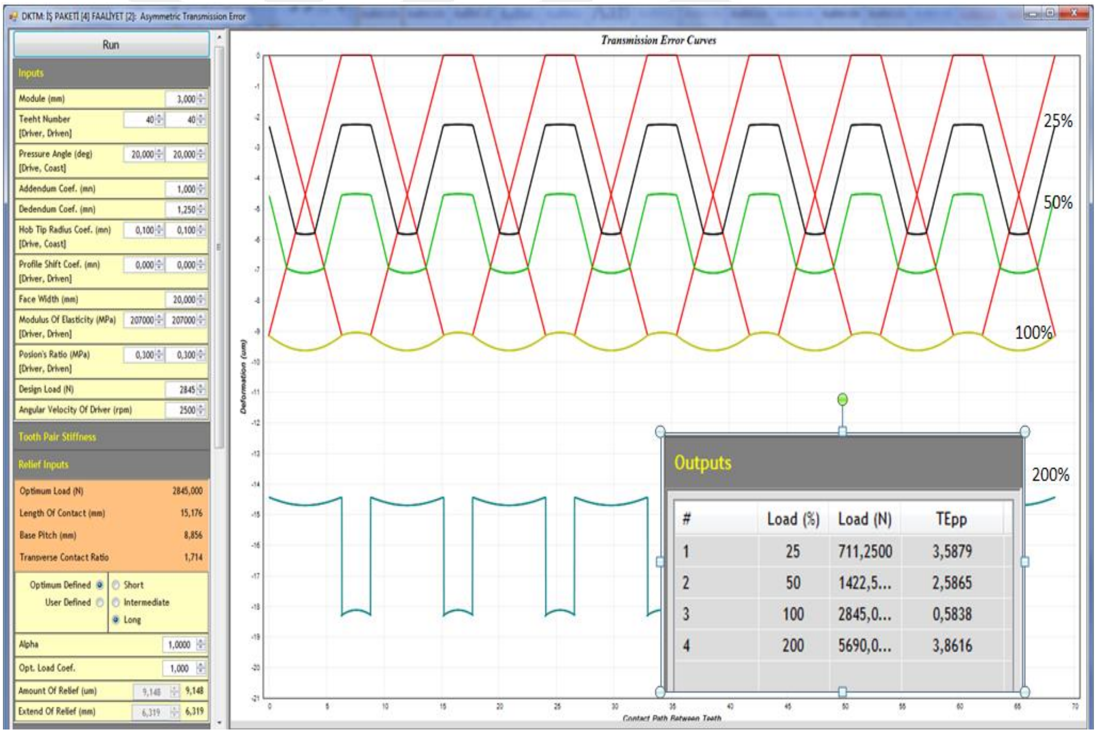


Figure 5.42 TE curves of Case S7 (D20C20) (software)

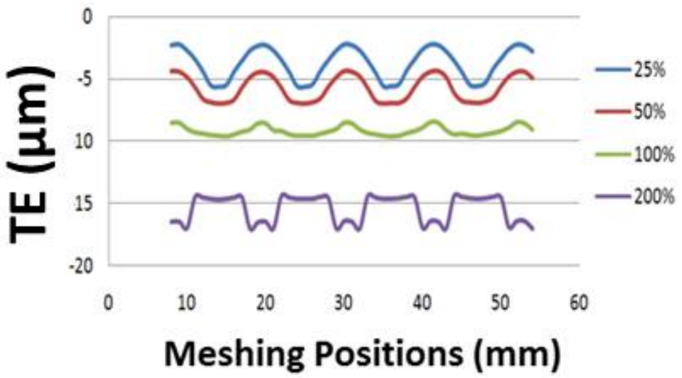


Figure 5.43 TE curves of Case S8 (D25C25) [1]

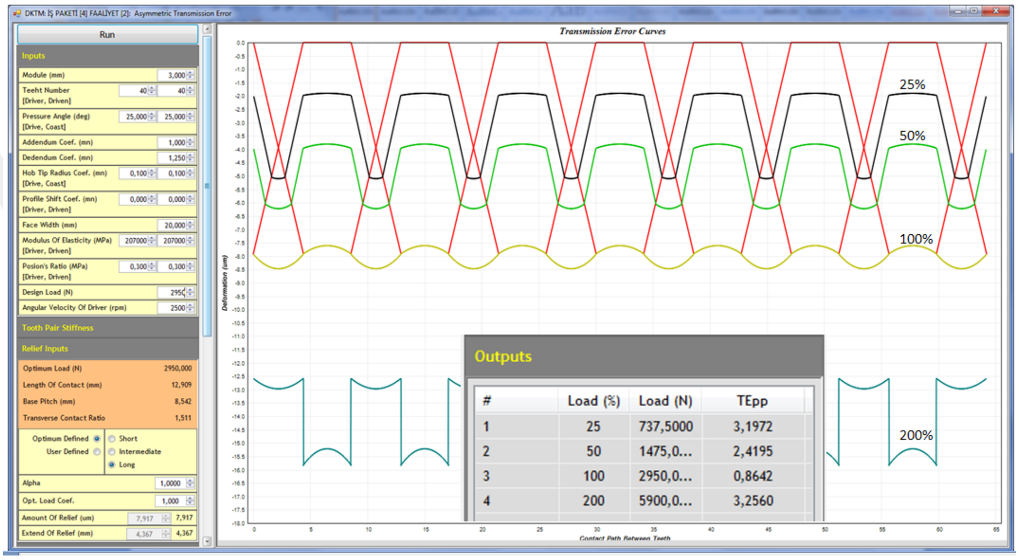


Figure 5.44 TE curves of Case S8 (D25C25) (software)

5.3.2.3 Case S9 and S10 (Transmission Error Calculation)

Gear pairs with long relief (S9) and short relief (S10) were analyzed for transmission error. General gear parameters are given in Table 5.16 and 5.17. TE curves for several loads are illustrated in Figures of 5.45-5.48. PPTE values are given in Table 5.18 and 5.19.

Table 5.16 General gear parameters of Case S9-S10

Parameter	Value	Unit
Normal module	4	mm
Teeth number of pinion	30	-
Pressure angle	20	Degree
Gear ratio	1	-
Face width	15	mm

Table 5.17 Relief parameters of Case S9-S10

Parameter	Value		Unit
	S9	S10	
Relief type	Long	Short	-
Amount of relief	31.46	31.46	µm
Extent of relief	7.17	3.89	mm

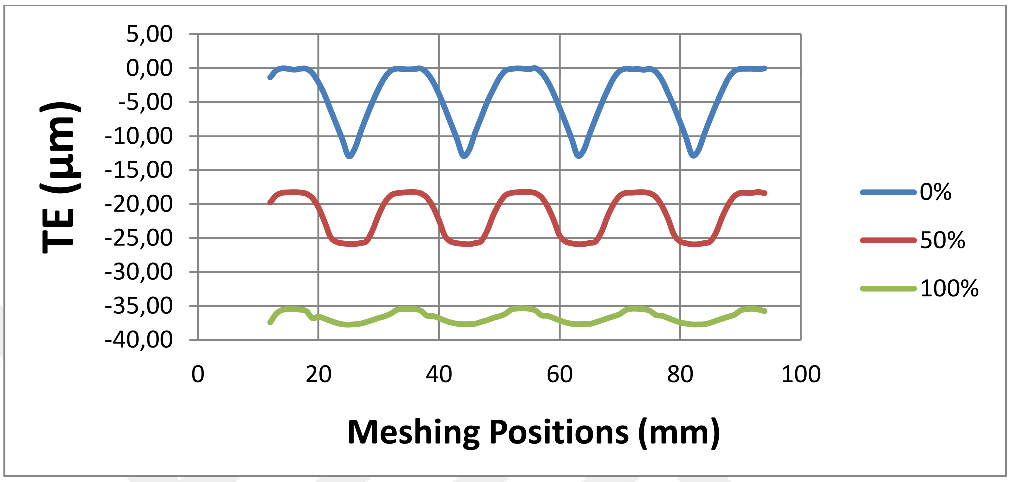


Figure 5.45 TE curves of Case S9 (D20C20 - Long relief) Ref [1]

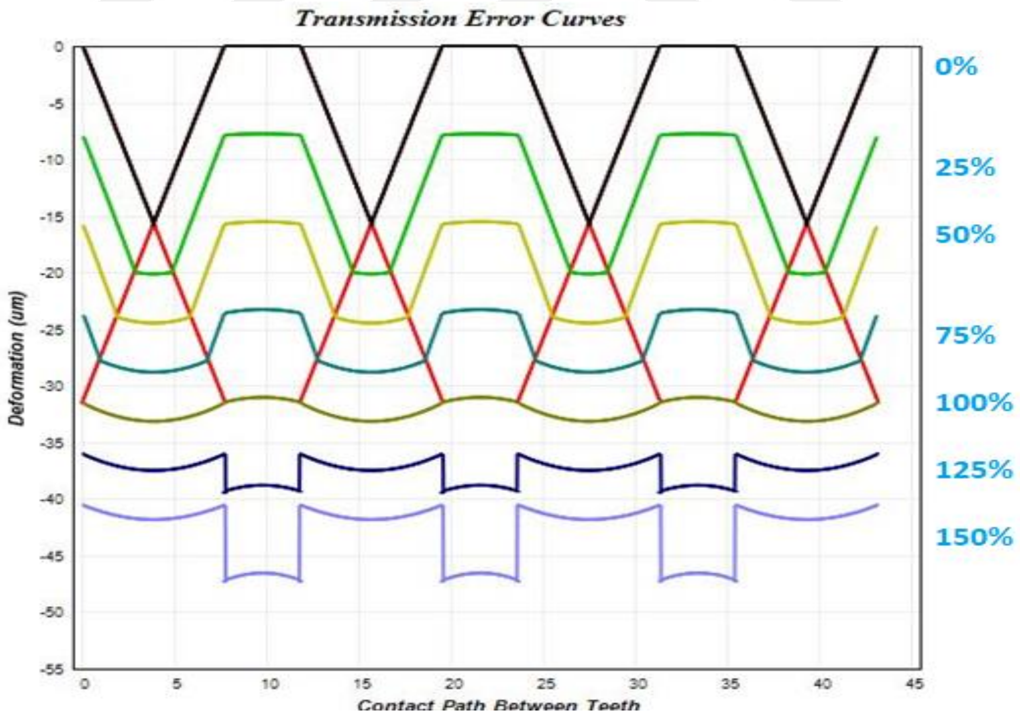


Figure 5.46 TE curves of Case S9 (D20C20 - Long relief) (software)

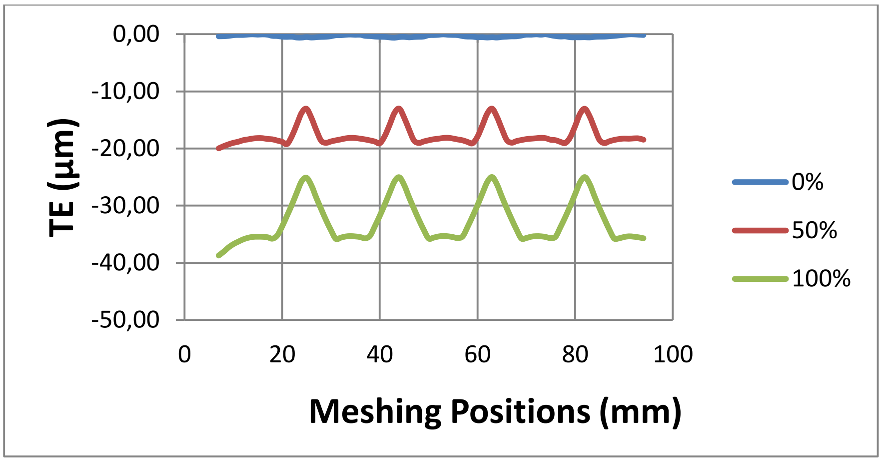


Figure 5.47 TE curves of Case S10 (D20C20 - Short relief) Ref [1]

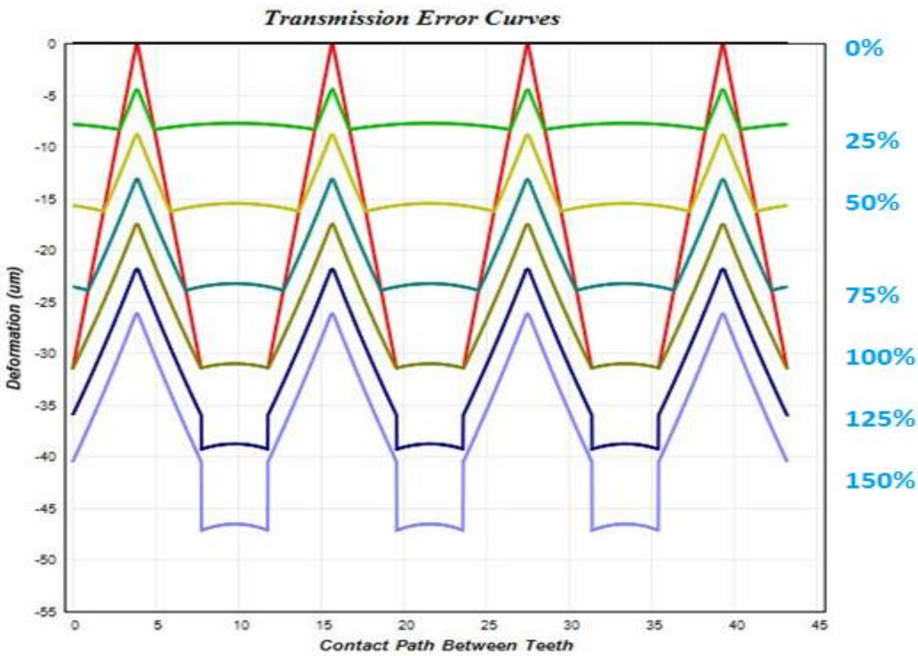


Figure 5.48 TE curves of Case S10 (D20C20 - Short relief) (software)

Table 5.18 PPTE values of Case S9 (D20C20 - Long relief)

PPTE (µm)		
Load (%)	**In-house software**	**FEA [1]**
0	15.760	12.950
50	8.956	7.730
100	2.114	2.390

Table 5.19 PPTE values of Case S10 (D20C20 - Short relief)

PPTE (µm)		
Load (%)	**In-house software**	**FEA [1]**
0	0	0.6
50	7.500	6.190
100	13.980	10.900

5.3.2.4 Case A6, A7 and A8

Asymmetric tooth spur gears with different pressure angle and relief parameters were studied to minimize transmission error here. General gear parameters and relief parameters for this gear pairs are given in Table 5.20 and 5.21. Transmission error curves of these gears by software and finite element analyses of Ref [1] are presented in Figure 5.49-5.50, 5.51-5.52, 5.53-5.54. These figures' regimes are similar and do not differ from each other drastically. PPTE values are given in Table 5.22, 5.23 and 5.24 for Cases of A6, A7 and A8 respectively.

Table 5.20 General gear parameters of Case A6-A7-A8

Parameter	Value	Unit
Normal module	3	mm
Teeth number of pinion	40	-
Gear ratio	1	-
Design torque	160.43	Nm
Face width	20	mm

Table 5.21 Gear parameters of Case A6-A7-A8

Parameter	Cases			Unit
	A6	A7	A8	
Pressure angle (drive/coast)	20/25	20/30	25/20	Degree
Relief type	Long	Long	Long	-
Design load	2845.26	2845.26	2950.06	N
Length of contact	15.18	15.18	12.91	mm
Amount of relief	8.833	8.533	8.149	µm

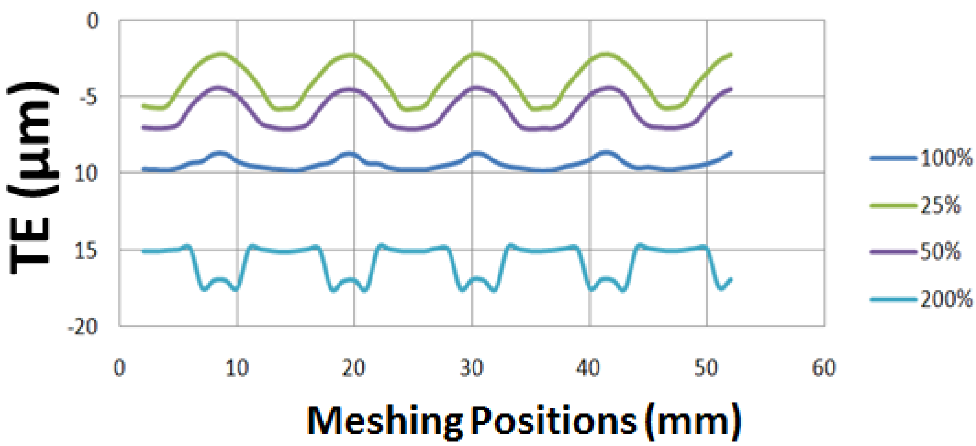


Figure 5.49 TE curves of Case A6 (D20C25-Long relief) Ref [1]

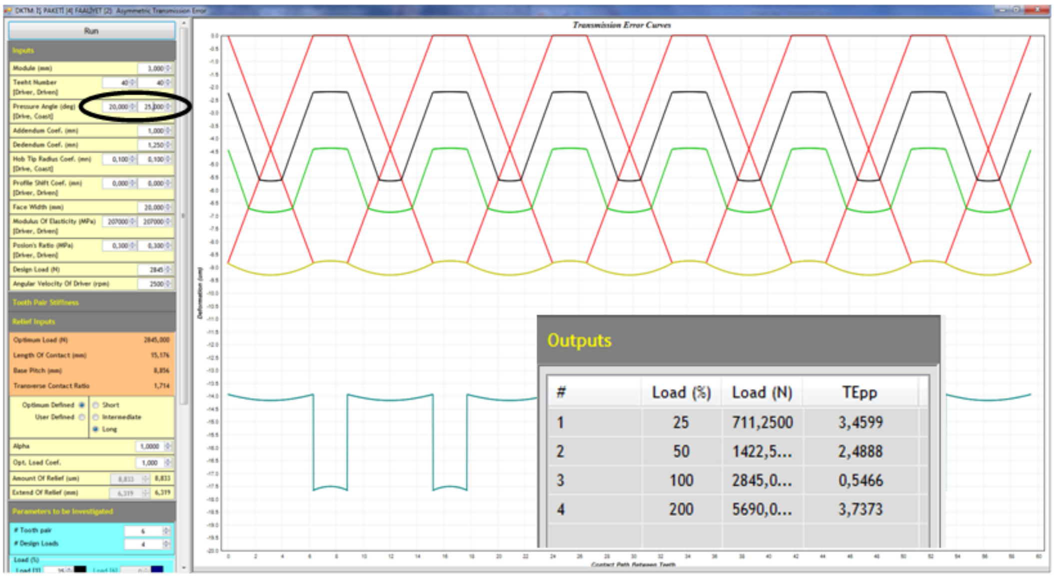


Figure 5.50 TE curves of Case A6 (D20C25-Long relief) (in-house software)

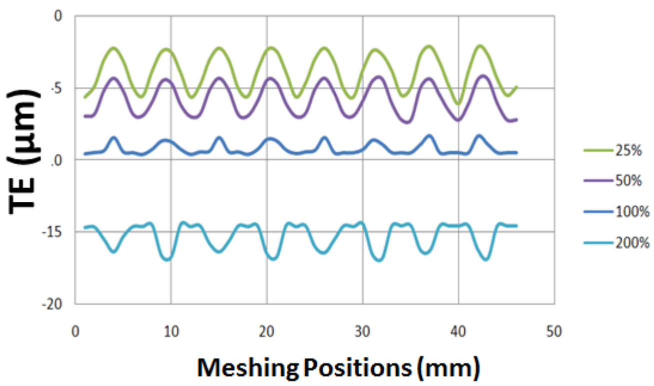


Figure 5.51 TE curves of Case A7 (D20C30-Long relief) Ref [1]

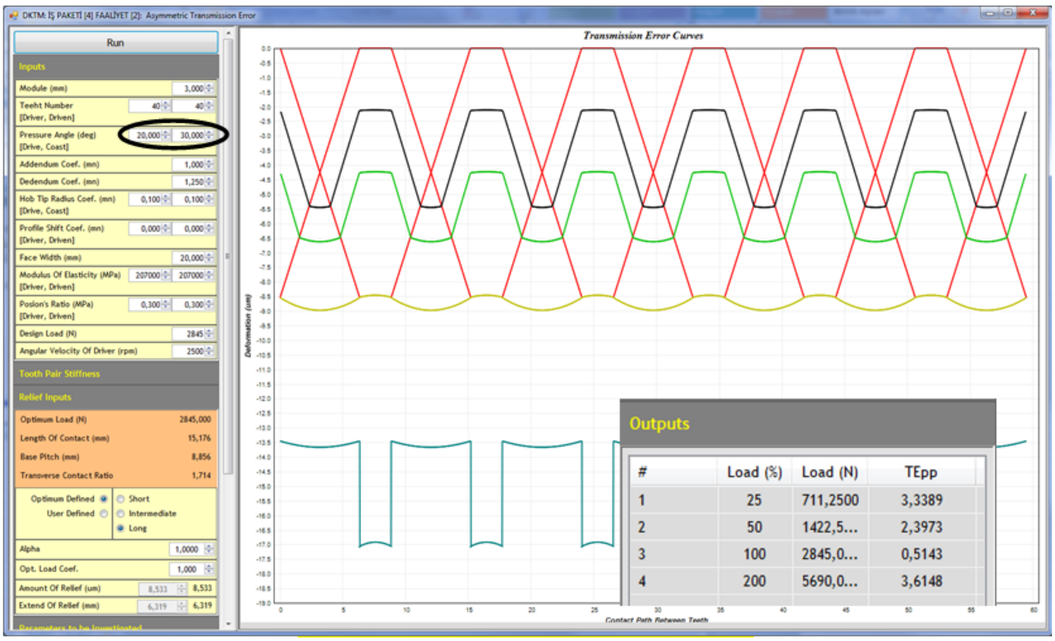


Figure 5.52 TE curves of Case A7 (D20C30-Long relief) (in-house software)

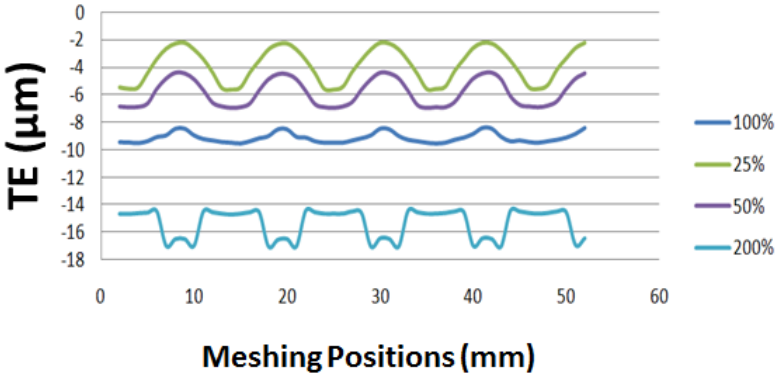


Figure 5.53 TE curves of Case A8 (D25C20-Long relief) Ref [1]

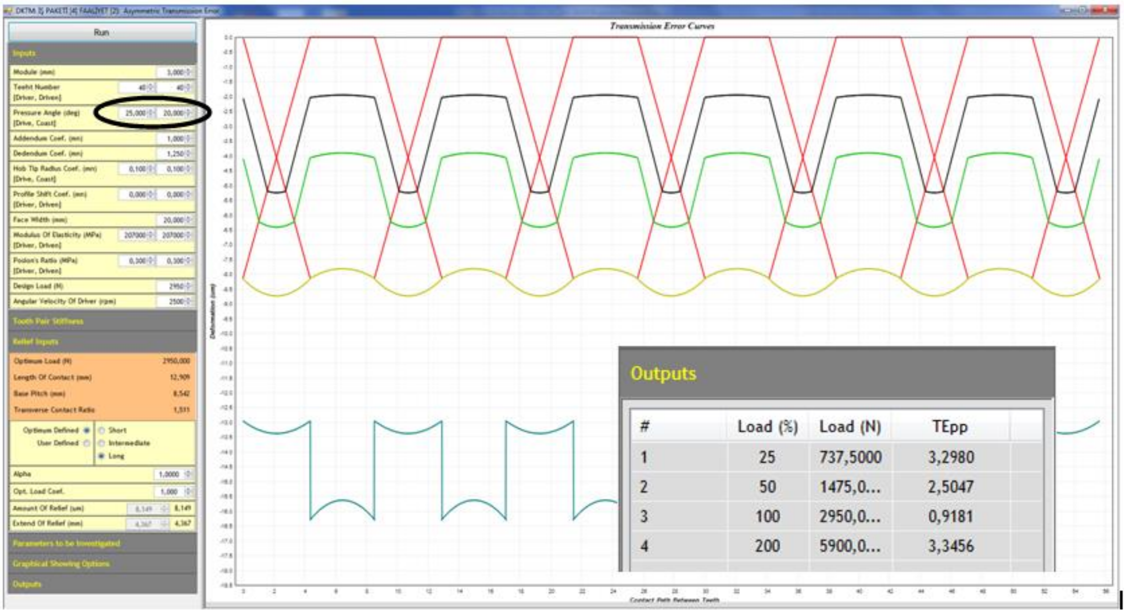


Figure 5.54 TE curves of Case A8 (D25C20-Long relief) (in-house software)

Table 5.22 PPTE values of Case A6 (D20C25)

PPTE (μm)		
Load (%)	In-house software	FEA
25	3.460	3.500
50	2.490	2.500
100	0.547	0.600

Table 5.23 PPTE values of Case A7 (D20C30)

PPTE (μm)		
Load (%)	**In-house software**	**FEA**
25	3.390	3.900
50	2.397	2.500
100	0.514	0.500

Table 5.24 PPTE values of Case A8 (D25C20)

PPTE (μm)		
Load (%)	**In-house software**	**FEA**
25	3.298	3.800
50	2.505	2.800
100	0.918	1.000

5.3.2.5 Case A9 (No Relief), A10 (Long Relief) and A11 (Short Relief)

Asymmetric profile spur gears with long and short relief were studied for minimization of transmission error. Gear and relief parameters for cases A9, A10 and A11 are given in Table 5.25 and 5.26. Transmission error curves of asymmetric spur gears with no relief (Case A9), long relief (Case A10) and short relief (Case A11) constructed by software and finite element analyses of Ref [1] are illustrated in Figure 5.55, 5.56 and 5.57. PPTE values are given in Table 5.27, 5.28 and 5.29 for Cases of A9, A10 and A11 respectively.

PPTE (Table 5.27-5.29) values differ from each other for both works but TE curves constructed by software and FEA [1] are very similar in terms of regime. Difference is higher for design load of Case A9 and A11 while higher difference is obtained for no load condition of Case A10.

Table 5.25 General gear parameters of Case A9-A10-A11

Parameter	Value	Unit
Normal module	4	mm
Teeth number of pinion	30	-
Pressure Angle (drive/coast)	30/20	Degree
Gear ratio	1	-
Addendum coefficient	1.00	-
Dedendum coefficient	1.25	-
Hob tip radius coefficient	0.3	-
Design torque	422	Nm
Face width	15	mm

Table 5.26 Relief parameters of Case A9-A10-A11

Parameter	Cases			Unit
	A9	**A10**	**A11**	
Pressure angle (drive/coast)	30/20	30/20	30/20	Degree
Relief type	No	Long	Short	-
Amount of relief	0	25.514	25.514	μm
Extent of relief	0	3.843	1.922	mm

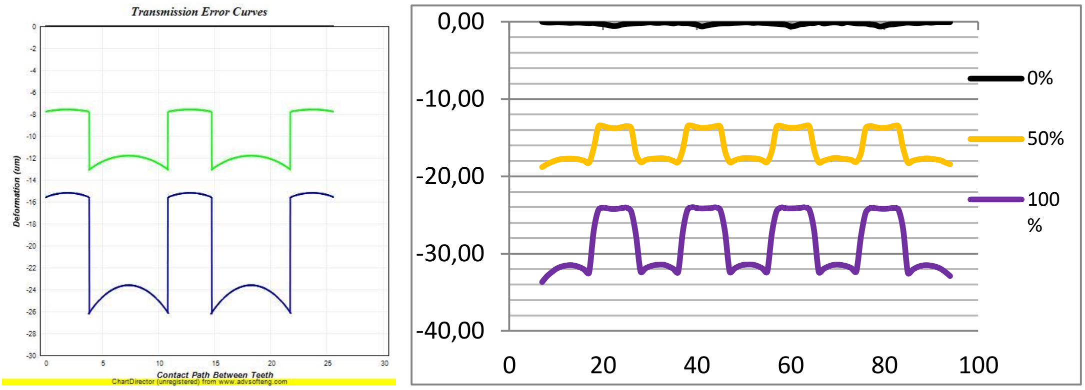


Figure 5.55 TE curves of Case A9 (D30C20-No relief) - software and Ref [1]

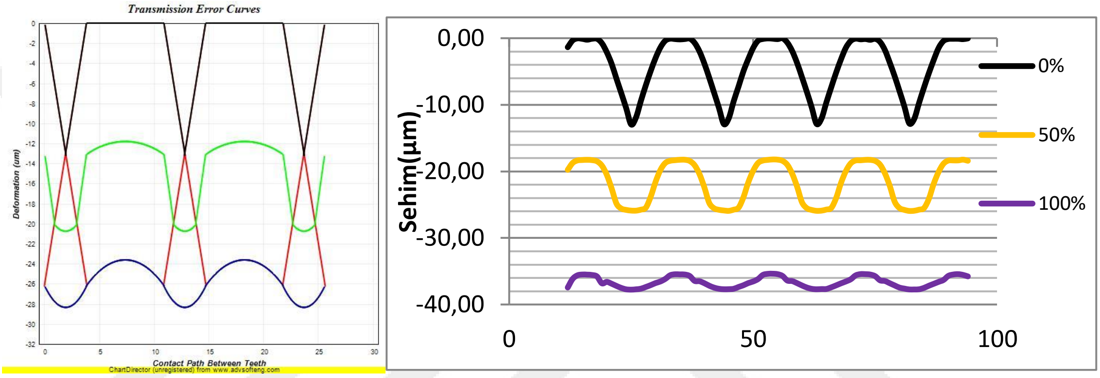


Figure 5.56 TE curves of Case A10 (D30C20-Long relief) - software and Ref [1]

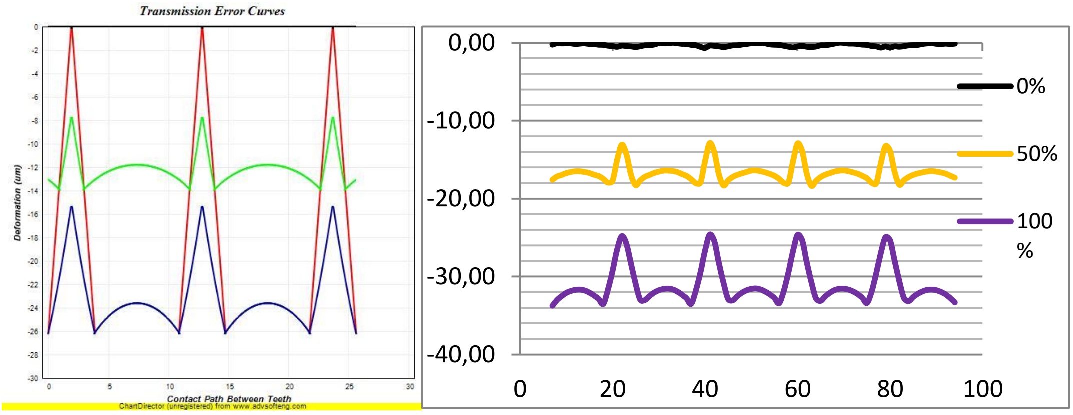


Figure 5.57 TE curves of Case A11 (D30C20-Short relief) - software and Ref [1]

Table 5.27 PPTE values of Case A9

PPTE (μm)		
Load (%)	**In-house software**	**FEA**
0	0	0.57
50	5.07	4.97
100	10.14	8.90

Table 5.28 PPTE values of Case A10

PPTE (µm)		
Load (%)	**In-house software**	**FEA**
0	13.15	9.77
50	8.94	6.51
100	4.73	2.25

Table 5.29 PPTE values of Case A11

PPTE (µm)		
Load (%)	**In-house software**	**FEA**
0	0.12	0.67
50	6.2	5.47
100	10.87	8.90

CHAPTER 6

DISCUSSION AND CONCLUSION

6.1 Introduction

This thesis has been conducted as a part of Indigenous Helicopter Program of Rotary Wing Technology Center (DKTM) [1] which was launched in the scope of contract signed between SSM (Undersecretariat for Defence Industries) and TAI (Turkish Aerospace Industries).

This study mainly is based on macro and micro design-analysis of asymmetrical tooth spur gears which can be applied to helicopter transmission units because of their high load carrying advantages.

User friendly software which have capabilities such as macro geometry generation, root bending stress calculations based on ISO [16], stiffness calculations based on thin slice theory, and micro profile design for minimum transmission error based on Harris map has been developed for symmetric and asymmetrical profile spur gears.

Results of root stress, stiffness and transmission error calculations have been compared to those of published papers.

6.2 About Macro Geometry Generation

By using developed software, macro geometries of symmetrical and asymmetrical spur gears have been generated. Commercial software, KissSoft [7] was used for checking accuracy and correctness of generated macro profiles of symmetric spur gears. Two comparable symmetric tooth profiles (for drive and coast side flanks) were generated by using KissSoft for each asymmetric tooth profile for comparison with outcome of developed software.

Macro geometries for symmetric and asymmetric tooth spur gears generated by both tools coincide with each other. Points of both profiles of involute and root satisfy for in-house software and KissSoft.

It can be concluded that, developed software yields accurate macro geometries for symmetrical and asymmetrical external spur gears.

6.3 About Tooth Root Bending Stress Evaluation

Gear root bending stresses have been evaluated based on ISO [16] by using developed in-house software for asymmetric spur gears. Root stresses were determined by using software and compared to results of published papers and KissSoft [7].

Software estimates root stresses in almost same levels with KissSoft whereas root stresses evaluated by using software are higher than reference papers presented in literature. This give chance to design stronger gears for desired torque amount. But heavier gears than intended are obtained.

6.4 About Tooth Pair Stiffness, Micro Profile Design, Static Transmission Error

Thin slice theory and Harris map method has been explained for evaluations of tooth stiffness and transmission error respectively in this thesis. Micro profile design has been conducted to optimize quasi static transmission error based on calculated stiffness. Smooth transmission error curves have been constructed for design and off-design loads.

A module of user friendly software has been developed to calculate stiffness, transmission error, load sharing, surface contact stress (P), sliding velocity (V), (PxV) and construct corresponding curves.

6.5 Further Studies

The studies conducted for asymmetric spur gears can be applied for helical gears and internal spur gears. Asymmetry for helical gears can be searched for further reductions in root stress.

Micro profile design can also be investigated to obtain minimum transmission error and smooth transmission error curves.

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